IN TWO SECTIONS—SECTION TWO

Transactions

of the

A.S.M.E.

6
13 2
33 3

SOCIETY RECORDS—PART 1

FEBRUARY, 1941

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of The American Society of Mechanical Engineers

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Foreword

THE Transactions of The American Society of Mechanical Engineers include selected technical papers and reports delivered at meetings of the Society, its Professional Divisions, and its Local Sections, the *Journal of Applied Mechanics* (contributions of the Applied Mechanics Division), certain records of the Society of permanent value, and indexes to its publications.

In order to secure the advantages of timeliness and greater usefulness in issuing these Society Records, the material comprising them is divided into a number of parts, each one of which is mailed as a supplement to one of the regular monthly issues of the Transactions. For 1941, the first of these, the present issue, contains the personnel of the Council and committees for the year. Another, to be issued sometime later in the year, will contain memorial notices of deceased members. The indexes to miscellaneous publications, *Mechanical Engineering,* and to the Transactions themselves, must, necessarily, be issued in 1942, and will probably be mailed as a supplement to the January issue of that year.

In binding the 1941 Transactions, all of these parts of the Society Records will be assembled at the back of the volume as has been customary for several years. To aid in locating references in the bound volumes, the page numbers of the sections containing the *Journal of Applied Mechanics* and the Society Records are preceded by the letters A and RI, respectively.

THE COMMITTEE ON PUBLICATIONS

WILLIAM A. HANLEY

PRESIDENT OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
1940-1941

William A. Hanley

William Andrew Hanley, mechanical engineer and business executive, of Indianapolis, Ind., President of The American Society of Mechanical Engineers for the year 1940-1941, was born in Greencastle, Ind., in 1886. He attended St. Joseph's College, Rensselaer, Ind., for two years and then matriculated at Purdue University, where he received the degree of Bachelor of Science in mechanical engineering in 1911. Twenty-six years later, his alma mater bestowed upon him the honorary degree of Doctor of Engineering.

Prior to attending Purdue University, Mr. Hanley worked five years for the Republic Steel Corporation and the Broderick Boiler Company, both in Muncie, Ind. Immediately after graduation, he entered the employ of Eli Lilly and Company, of Indianapolis, manufacturers of medicinal products. Today he is a director of that company and head of the engineering division. This division designs and supervises all engineering projects, construction, power, maintenance, etc., for the corporation, its branches and subsidiaries, and, in addition, operates certain highly mechanized production departments. In 1938-1939, Mr. Hanley spent much time in Basingstoke, England, building a new manufacturing plant for the British subsidiary of the Lilly company.

Mr. Hanley was elected an Associate-Member of the A.S.M.E. in 1913, promoted to full membership in 1920, and made a Fellow in 1936. In 1916, he was one of the organizers and the first secretary of the Central Indiana Section. In 1919, the local members elected him chairman of the Section. The following year saw the beginning of many years of service by him in the activities and affairs of the parent body with his acceptance of an appointment as one of the A.S.M.E. representatives on the American Engineering Council. During the period from 1922 to 1927, he served on the Committee on Local Sections and, from 1933 to 1938, on the Committee on Relations With Colleges. In 1927, he was elected to a three-year term as a Manager of the Society and, in 1930, to a two-year term as Vice-President. Other A.S.M.E. activities in which he has taken a part include the Special Committee on Junior Participation, Special Committee on Relationship of Society to Accrediting Program, and Committee on Medals.

Over a long period of years Mr. Hanley has contributed to the technical press a number of articles on both engineering and economic subjects. He is a past-president of the Indiana Engineering Council, an honorary member of Tau Beta Pi, a member of the Newcomen Society of England, and a fellow of the American Association for the Advancement of Science. He is also a trustee of Purdue University, of Park School of Indianapolis, of the Sigma Phi Epsilon Fraternity (national), and of the Associated Catholic Charities of Indianapolis.

The American Society of Mechanical Engineers

HEADQUARTERS: 29 West 39th St., New York, N. Y.
MID-WEST OFFICE: Room 1617, 205 West Wacker Drive, Chicago, Ill.

The members of the Council and of its standing and special committees given on the following pages are those in office on January 1, 1941, serving for the official year 1940–1941. The terms of office of members of other committees are not fixed by the official calendar.

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(Personnel of Special Committee, p. RI-7)

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(Personnel of Advisory Committee, p. RI-8)

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(Personnel of Professional Divisions' Executive Committees, p. RI-10)

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(Personnel of Local Sections' Executive Committees, p. RI-15)

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(Student Branches and Officers, p. RI-23)

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† Personnel of all Technical Committees, pp. RI-25-38.

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(Personnel of Boiler Code Committees, pp. RI-37-38)

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(Continued)

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(Dates in parentheses denote expiration of terms)

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A.S.M.E. REPRESENTATIVES ON OTHER ACTIVITIES

See also A.S.M.E. Representatives on Other Research Committees, etc., pages RI-26, 31, 34, 35, 38

(Dates in parentheses denote expiration of terms)

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SECTION M, ENGINEERING

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. L. Sackett

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. L. Baker (1942)
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Article B6A, Par. 16: The Standing Committee on Professional Divisions shall, under the direction of the Council, have supervision of the Professional Divisions of the Society.

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G. B. Karelitz (1942)
W. A. Carter (1943)
L. H. Fry (1944)
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Junior Advisers

A. E. Blirer (1941)
H. B. Fernald, Jr. (1942)

Aeronautic

Organized, 1920

C. H. Dolan, II, *Chairman*

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C. F. Kettering
Alex. Klemin
R. K. LeBlond
W. B. Mayo
P. B. Morgan
T. A. Morgan
S. A. Moss
E. A. Sperry
A. R. Stevenson, Jr.
J. G. Vincent
Theo. von Karman
C. J. Ward
E. P. Warner
B. M. Woods
Orville Wright

Ammunition

(Professional Group)

Organized, 1940

Committee to be appointed

Applied Mechanics

Organized, 1927

J. P. Den Hartog, *Chairman*

EXECUTIVE COMMITTEE

J. P. Den Hartog, *Chairman*
H. L. Dryden, *Secretary*
J. H. Keenan
Jesse Ormondroyd
B. M. Woods

Associates

Rupen Eksergian
J. C. Hunsaker
A. L. Kimball
F. M. Lewis
G. B. Pegram
C. R. Soderberg
E. O. Waters
H. M. Westergaard

Representative on Aviation Liaison Group

J. C. Hunsaker

Research Secretary

Jesse Ormondroyd

JOURNAL OF APPLIED MECHANICS

J. M. Lessells, *Editor*

SPONSORS

Dynamics, E. L. Thearle
Elasticity, Stephen Timoshenko
Fluid Mechanics, H. L. Dryden
Lubrication, G. B. Karelitz
Plasticity, A. Nádai
Strength of Materials, R. E. Peterson
Thermodynamics, J. A. Goff

Fuels

Organized, 1920

William G. Christy, *Chairman*

EXECUTIVE COMMITTEE

William G. Christy, *Chairman*
D. C. Weeks, *Secretary*
O. F. Campbell, *Representative on Aviation Liaison Group*
H. F. Hebley
A. R. Mumford
A. W. Thorson

Associates

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J. S. Bennett, 3rd
M. P. Cleghorn
B. J. Cross
M. D. Engle
D. S. Frank
E. R. Kaiser
T. A. Marsh
M. A. Mayers
R. L. Rowan
J. E. Tobey
D. C. Weeks

COOPERATION WITH A.I.M.E.

J. E. Tobey, *Chairman*

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J. F. Barkley, *Chairman*
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A. G. Christie
William G. Christy
T. A. Marsh
T. E. Purcell
R. A. Sherman
R. R. Tucker

PROGRAM

A. W. Thorson, *Chairman*
D. S. Frank, *Assistant Chairman*
E. R. Kaiser, *Junior Member*

REVIEW OF PAPERS

William G. Christy
M. D. Engle
A. W. Thorson
D. C. Weeks

Graphic Arts

Organized, 1922

EXECUTIVE COMMITTEE

T. E. Dalton, *Secretary*
A. E. Giegengack
F. W. Hoch
W. B. Laughton
R. G. MacDonald
W. M. Passano
B. L. Sites
B. D. Stevens
B. L. Wehmhoff

Heat Transfer

(Professional Group)

Organized, 1938

E. D. Grimison, *Chairman*
W. S. Patterson, *Group Secretary*

EXECUTIVE COMMITTEE

E. D. Grimison, *Chairman*
T. B. Drew
L. M. K. Boelter
R. A. Bowman
H. C. Hottel

Advisory Associates

C. E. Lucke
A. K. Scott
J. H. Sengstaken

Junior Representatives

J. L. Menson
R. H. Wolin

Heat Transfer
(Continued)

COORDINATION

. S. Patterson

Liaison Officer for Local Sections

K. Scott

presentative on Aviation Liaison Group

M. K. Boelter

Representatives of Other Divisions

uels, B. J. Cross
ydraulic, J. D. Scoville
on and Steel, W. Trinks
l and Gas Power, F. G. Hechler
etroleum, J. D. Peterson
ower, O. F. Campbell
ocess Industries, Arnold Weisselberg
ailroad, L. H. Fry

Research Secretary

B. Drew

Members at Large

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. H. Heilman
. L. Tuve

DIRECT-FIRED FLUID HEATERS AND BOILERS

. D. Grimison, *Chairman*
ohn Blizard
. S. Frank
. B. Schueller
/. J. Wohlenberg

NDUSTRIAL FURNACES AND KILNS

/. Trinks, *Chairman*
. C. Hottel
/. A. Ticknor

PAPERS

/. L. De Baufre, *Chairman*
. A. Bowman
. F. Kayan
. K. Scott
. A. Sherman

TESTING TECHNIQUE

I. C. Hottel, *Chairman*
. J. Cross
. H. Jackson
. H. Rushton
. K. Scott

THEORY AND FUNDAMENTAL RESEARCH

. M. K. Boelter, *Chairman*
. P. Colburn
. B. Drew
Max Jakob
. D. Streid

THERMO-PHYSICAL PROPERTIES OF MATERIALS

R. H. Norris, *Chairman*
O. Kenneth Bates
T. H. Chilton
R. C. H. Heck
Max Jakob
R. J. S. Pigott
J. F. Downie Smith

Subcommittee on Specific Heat of Gases

Max Jakob, *Chairman*
W. L. De Baufre
J. A. Goff
A. C. Gullikson
R. C. H. Heck
R. J. S. Pigott
R. L. Sweigert

UNFIRED HEAT TRANSFER EQUIPMENT

R. A. Bowman, *Chairman*
W. E. Beline
A. C. Mueller
B. E. Short
Townsend Tinker

Hydraulic

Organized, 1926

E. B. Strowger, *Chairman*

EXECUTIVE COMMITTEE

E. B. Strowger, *Chairman*
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M. P. O'Brien
J. D. Scoville
F. G. Switzer
R. V. Terry

CAVITATION

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J. M. Mousson
W. J. Rheingans
G. F. Wislicenus

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American Society for Testing Materials, F. N. Speller
Engineering Institute of Canada, Ernest Brown
Institution of Mechanical Engineers, G. S. Baker

Representative of France

A. Tenot

HYDRAULIC PRIME MOVERS

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J. P. Growden
L. F. Harza
P. L. Heslop
George Jessup
F. H. Rogers
F. Schmidt
S. O. Schomberger
S. H. Van Patter

PUMPING MACHINERY

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R. L. Daugherty, *Chairman*

WATER HAMMER

Honorary Chairman, Lorenzo Allievi, Rome, Italy

J. D. Scoville, *Sponsor*
S. Logan Kerr, *Chairman*
N. R. Gibson
Eugene Halmos
L. F. Moody
R. S. Quick
E. B. Strowger

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American Water Works Association, F. M. Dawson and L. H. Kessler

Associate Members, Representing:

Australia, George Higgins
Brazil, A. W. K. Billings and F. Knapp
Engineering Institute of Canada, R. W. Angus and F. M. Wood
France, Louis Bergeron and Charles Camichel
Germany and Verein deutscher Ingenieure, D. Thoma
Great Britain and Institution of Mechanical Engineers, E. Bruce Ball and A. H. Gibson
Italy, Gaudenzio Fantoli and Albino Pasini
Switzerland, Charles Jaeger and O. Schnyder

Machine Shop Practice

Organized, 1921

Sol Einstein, *Chairman*

EXECUTIVE COMMITTEE

Sol Einstein, *Chairman*
Warner Seely, *Secretary*
E. W. Ernest
A. M. Johnson
Erik Oberg

CUTTING METALS

Hans Ernst, *Chairman*

FOUNDRY PRACTICE

James Thomson, *Chairman*
R. E. Kennedy, *Secretary*

LUBRICATION ENGINEERING

B. G. Tang, *Chairman*
C. M. Larson
H. J. Masson

Machine Shop Practice

(Continued)

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E. O. Waters, *Chairman*
A. E. R. de Jonge
J. H. Marchant
J. Marin

WELDING

C. W. Obert, *Chairman*

Management

Organized, 1920

H. B. Bergen, *Chairman*

EXECUTIVE COMMITTEE

H. B. Bergen, *Chairman*
J. R. Bangs, Jr., *Vice-Chairman*
G. M. Varga, *Secretary*
L. A. Appley
J. M. Talbot
Archie Williams

Representatives of Local Sections

Atlanta, S. C. Hale
Kansas, A. H. Sluss
Louisville, C. D. Eldridge
Metropolitan, W. F. Carhart
Milwaukee, B. V. E. Nordberg
Philadelphia, C. S. Gotwals
Seattle, H. J. McIntyre

Junior Adviser

F. R. Kiernan

Representative on Aviation Liaison Group

R. E. Gillmor

Research Secretary

E. H. Hempel

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L. P. Alford
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C. W. Beese
F. B. Bell
Wallace Clark
H. V. Coes
K. H. Condit
Howard Coonley
N. E. Elsas
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W. D. Fuller
W. H. Gesell
Lillian M. Gilbreth
R. E. Gillmor
C. H. Hatch
E. H. Hempel
P. E. Holden
W. F. Hosford
J. M. Juran
D. S. Kimball
W. H. Kushnick
T. S. McEwan
L. C. Morrow
A. I. Peterson

GENERAL COMMITTEE
(Continued)

D. B. Porter
F. E. Raymond
J. W. Roe
E. H. Schell
E. D. Smith
L. W. Wallace
J. E. Younger

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Administration Organization, L. A. Appley
Industrial Marketing, J. R. Bangs, Jr.
Mathematical Statistics, A. I. Peterson
Works Standardization, J. M. Juran

DEPRECIATION STUDIES

H. V. Coes
P. T. Norton, Jr.

Materials Handling

Organized, 1920

G. E. Hagemann, *Chairman*

EXECUTIVE COMMITTEE

G. E. Hagemann, *Chairman*
R. B. Renner, *Vice-Chairman*
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J. A. Jackson
M. C. Maxwell

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N. W. Elmer
H. C. Keller
R. H. McLain
F. E. Moore
P. D. Oesterle
E. D. Smith
J. B. Webb

Junior Associates

A. J. Burke
Cornelius Crowley
E. Z. Gabriel
R. W. Grundman
D. D. Jones
Peter Shaw

Metals Engineering

(Formerly Iron and Steel)

Organized, 1927

Reorganized, 1940

W. R. Webster, *Chairman*

EXECUTIVE COMMITTEE

W. R. Webster, *Chairman*
R. A. North, *Secretary*
J. A. Clauss
G. L. Fisk
J. H. Hitchcock
W. Trinks

Associates

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S. M. Marshall
B. C. McFadden
J. H. Romann
M. D. Stone
R. J. Wean
S. M. Weckstein
T. H. Wickenden

Oil and Gas Power

Organized, 1921

C. W. Good, *Chairman*

EXECUTIVE COMMITTEE

C. W. Good, *Chairman*
L. N. Rowley, Jr., *Secretary*
H. E. Degler
E. S. Dennison
W. L. H. Doyle
F. G. Hechler

Associates

R. D. Campbell
G. J. Dashefsky
L. R. Ford
W. K. Gregory
E. J. Kates
L. H. Morrison
B. V. E. Nordberg
M. J. Reed
Lee Schneitter

Junior Adviser

C. K. Holland

Research Secretary

Lee Schneitter

Liaison Representatives

American Society of Naval Architects and Marine Engineers, L. R. Ford
Aviation Liaison Group, H. E. Degler
Heat Transfer Group, F. G. Hechler

EDITING

E. J. Kates
Ernest Nibbs
M. J. Reed

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L. H. Morrison
M. J. Reed

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H. C. Major, *Chairman*
H. C. Lenfest, *Secretary*
B. B. Bachman
R. P. Bolster
L. T. Brown
R. D. Campbell
E. Hale Codding
W. J. Cumming

OIL ENGINE POWER COST
(Continued)

. J. KATES
. B. MORGAN
. I. MOORE
. D. NOILES
. J. REED
. TOM SAWYER
EE SCHNEITTER
. H. SCHWEITZER
. C. THUERK
. A. TRIMMER
TANLEY WRIGHT

OIL AND GAS POWER CONFERENCES

1941 Arrangements Committee

. W. GOOD, *Chairman*
. C. BOYER, *Chairman, Kansas City Arrangements Committee*
H. E. DEGLER

1942 Location and Selection Committee

W. L. H. DOYLE, *Chairman*
C. W. GOOD
W. K. GREGORY

PUBLICITY

L. H. MORRISON

TECHNICAL PROGRAM

E. S. DENNISON, *Chairman*
G. C. BOYER
W. L. H. DOYLE
LEE SCHNEITTER

Petroleum

Organized, 1925

Reorganized, 1937

W. F. HERBERT, *Chairman*

EXECUTIVE COMMITTEE

W. F. HERBERT, *Chairman*
W. H. CARSON, *Secretary*
H. R. AUERSWALD
E. H. BARLOW
H. L. EGGLESTON

GENERAL COMMITTEE

Atlantic Group

GUSTAVUS AUER
P. E. FRANK
T. H. HAMILTON
J. D. PETERSON
V W. SMITH

Mid-Continent Group

H. A. AUERSWALD
C. J. COBERLY
W. F. HERBERT
F. J. HOLZBAUR
W. H. STUEVE

Power

Organized, 1920

G. C. EATON, *Chairman*

EXECUTIVE COMMITTEE

G. C. EATON, *Chairman*
L. M. GOLDSMITH, *Secretary*
THEODORE BAUMEISTER, JR., *Research Secretary*
O. F. CAMPBELL
E. L. HOPPING

Junior Adviser

F. ROBERT HARTIN

Process Industries

Organized, 1934

J. W. HUNTER, *Chairman*

EXECUTIVE COMMITTEE

J. W. HUNTER, *Chairman*
T. R. OLIVE, *Secretary*
THEODORE BAUMEISTER, JR.
RICHARD O'MARA
WILLIAM RAISCH
A. F. SPITZGLASS
ARNOLD WEISSELBERG
W. R. WOOLRICH
J. I. YELLOTT
F. L. YERZLEY

Liaison Officer With Standing Committee on Professional Divisions

J. H. SENGSTAKEN

Junior Adviser

G. M. BEISCHER

Research Secretary

ARNOLD WEISSELBERG

Other Liaison Representatives

Aviation Liaison Group
  Industrial Instruments and Regulators Committee, E. A. SPERRY
  Subdivision on Rubber and Plastics
    W. F. BARTOE, Plastics
    F. L. YERZLEY, Rubber
Heat Transfer Group, ARNOLD WEISSELBERG
Society of Automotive Engineers, F. L. YERZLEY

COMMITTEE CHAIRMEN

Air Conditioning, C. F. KAYAN
Drying, ARNOLD WEISSELBERG
Food Processing, G. L. MONTGOMERY
Industrial Instruments and Regulators
  E. S. SMITH, *Chairman*
  A. F. SPITZGLASS, *Secretary*
  J. C. PETERS, *Acting Secretary*
Manufactured Gas, G. M. BEISCHER
Mechanical Separation, RICHARD O'MARA
Papers, Awards, and Honors, C. E. LUCKE
Program, J. W. HUNTER
Pulp and Paper, A. D. ASBURY
Sanitation, WILLIAM RAISCH
Sugar, F. M. GIBSON
Sulphur, B. E. SHORT
Vegetable Oils, R. W. MORTON

Subdivision on Rubber and Plastics

EXECUTIVE COMMITTEE

F. L. YERZLEY, *Chairman*
J. F. D. SMITH, *Vice-Chairman*
G. M. KLINE, *Secretary*
W. F. BARTOE
S. H. HAHN
L. E. JERMY
R. A. NORTH
W. A. ZINZOW

Railroad

Organized, 1920

A. I. LIPETZ, *Chairman*

EXECUTIVE COMMITTEE

A. I. LIPETZ, *Chairman*
C. L. COMBES, *Secretary*
J. G. ADAIR
D. S. ELLIS
J. R. JACKSON
W. M. SHEEHAN

GENERAL COMMITTEE (RR2)

A. I. LIPETZ, *Chairman*
H. P. ALLSTRAND
B. S. CAIN
W. I. CANTLEY
J. E. DAVENPORT
L. B. JONES
F. G. LISTER
F. E. LYFORD
K. F. NYSTROM
A. A. RAYMOND
JOHN ROBERTS
R. W. SALISBURY
W. C. SANDERS
DENNISTOUN WOOD
E. G. YOUNG
G. A. YOUNG

ADVISORY COMMITTEE (RR3)

L. H. FRY
G. W. RINK
C. T. RIPLEY
E. C. SCHMIDT
W. H. WINTERROWD

MEETINGS AND PAPERS (RR5)

W. M. SHEEHAN, *Chairman*
W. I. CANTLEY
J. R. JACKSON
C. T. RIPLEY

SURVEY (RR6)

E. G. YOUNG, *Chairman*
B. S. CAIN
K. F. NYSTROM

MEMBERSHIP (RR8)

A. A. RAYMOND, *Chairman*
D. S. ELLIS
W. C. SANDERS
W. M. SHEEHAN
L. K. SILLCOX

Textile

Organized, 1921

A. D. Asbury, *Chairman*

EXECUTIVE COMMITTEE

A. D. Asbury, *Chairman*
F. L. Bradley, *Vice-Chairman*
W. B. Heinz, *Secretary*
R. DeVere Hope
H. H. Iler
J. D. Robertson
E. R. Stall
A. Wadsworth Stone

Associates

A. W. Benoit
W. S. Brown
Winn Chase
M. A. Golrick, Jr.
Albert Palmer

Southern Representative

S. B. Earle

PROGRAM

W. W. Starke, *Chairman and Metropolitan Representative*

Wood Industries

Organized, 1921

C. B. Norris, *Chairman*

EXECUTIVE COMMITTEE

C. B. Norris, *Chairman*
M. J. McDonald
D. R. Gray
Sern Madsen
T. D. Perry

Associates

C. L. Babcock
P. H. Bilhuber
H. B. Carpenter
F. P. Cartwright
G. E. French
A. W. Keuffel
A. S. Kurkjian
J. S. Mathewson
E. D. May
R. H. McCarthy
A. D. Smith, Jr.
H. M. Sutton
Charles White
A. C. Fegel, *Metropolitan Representative*

COMMITTEE CHAIRMEN

Dimensional Limits and Allowances, Sern Madsen
Use of Plywood as an Engineering Material, T. D. Perry
Wood Finishing, M. J. MacDonald

LOCAL SECTIONS

ARTICLE B6A, PAR. 17: The Standing Committee on Local Sections shall, under the direction of the Council, have supervision of the Local Sections of the Society.

STANDING COMMITTEE ON LOCAL SECTIONS

H. L. EGGLESTON, *Chairman* (1941)

J. N. LANDIS (1942)
F. W. MARQUIS (1944)
F. L. WILKINSON, JR. (1943)
J. A. KEETH (1945)

Junior Advisers

SIDNEY DAVIDSON (1941)
C. C. KIRBY (1942)

REGIONAL GROUP DELEGATES TO ANNUAL CONFERENCES

Terms expire October, 1941

A. R. ACHESON, *Speaker for 1940 Conference*, Group III
L. H. VON OHLSEN, *Secretary*, Group I
O. B. SCHIER, II, Group II
H. M. GANO, Group V
A. D. ASBURY, Group IV
B. V. E. NORDBERG, Group VI
W. D. TURPIN, Group VII

Terms expire October, 1942

A. R. MUMFORD, *Speaker for 1941 Conference*, Group II
A. D. HUGHES, *Secretary*, Group VII
A. D. ANDRIOLA, Group I
C. T. OERGEL, Group V
J. S. MOREHOUSE, Group III
R. A. CROSS, Group VI
F. C. SMITH, Group IV
C. W. CRAWFORD, Group VIII

AKRON-CANTON

Organized: 1920
Territory: Counties of Richland, Ashland, Medina, Summit, Portage, Wayne, Stark, Holmes, Tuscarawas, Carroll, and Coshocton in Ohio
Place of Meeting: As selected monthly
Number of Members: 130

EXECUTIVE COMMITTEE

M. R. BOWERMAN, *Chairman*
A. G. WALKER, *Vice-Chairman*
E. D. GEORGE, *Secretary-Treasurer*
V. R. CAMP
JAMES FORREST
S. H. HAHN
L. B. HOLMES
O. J. HORGER
E. H. KENDALL
A. D. MACLACHLAN
G. C. MCMULLEN
G. J. SCHOESSOW
A. W. SEEKINS
A. E. SHETLER
J. H. VANCE

ANTHRACITE-LEHIGH VALLEY

Organized: 1920, as Lehigh Valley; reorganized, 1928, as Anthracite-Lehigh Valley
Territory: Counties of Bradford, Susquehanna, Wayne, Sullivan, Wyoming, Lackawanna, Columbia, Luzerne, Monroe, Pike, Schuylkill, Carbon, Berks, Lehigh, Northampton in Pennsylvania, and Warren in New Jersey
Place of Meeting: One meeting annually at Allentown, Bethlehem, Easton, Hazleton, Pottsville, Reading, Scranton, and Wilkes-Barre
Local Organization: The Engineers' Club of Lehigh Valley
Number of Members: 200

EXECUTIVE COMMITTEE

R. H. PORTER, *Chairman*
F. C. PETERS, *Vice-Chairman*
J. W. STEINMEYER, *Vice-Chairman*
D. G. WILLIAMS, *Vice-Chairman*
C. W. MERRICK, *Secretary*
M. C. STUART, *Treasurer*
G. W. FARNHAM
J. W. GISH, JR.
C. C. HERTEL
R. E. MOYER
L. E. MYLTING
W. P. SAUNIER
WALTER TALLGREN
R. L. WILLIS

ATLANTA

Organized: 1913
Territory: Radius of sixty miles from Atlanta, Ga.
Place of Meeting: Atlanta Athletic Club
Luncheon meeting every Monday at 12:30 p.m. at Atlanta Athletic Club
Number of Members: 85

EXECUTIVE COMMITTEE

F. C. SMITH, *Chairman*
A. H. KOCH, *Vice-Chairman*
J. M. RITTLEMEYER, *Secretary*
R. N. BENJAMIN
R. S. HOWELL
W. C. KIRBY
E. W. KLEIN, JR.
R. L. SWEIGERT

BALTIMORE

Organized: 1916
Territory: Radius of thirty miles from Baltimore, Md.
Place of Meeting: Engineers Club of Baltimore
Luncheon meeting every Wednesday at 12:00 noon at Engineers Club
Number of Members: 226

EXECUTIVE COMMITTEE

G. W. KEEN, *Chairman*
S. B. SEXTON, *Secretary-Treasurer*
W. D. BOYNTON
L. F. COFFIN
R. C. DANNETTEL
SIDNEY HAUSMAN
J. W. MOUSSON
L. F. WELANETZ
S. M. WHITELEY

JUNIOR GROUP

W. A. HAZLETT, *Chairman*
G. I. CHINN, *Vice-Chairman*
W. B. ELITZ, *Secretary*
J. F. HANNA
D. F. LANE

BIRMINGHAM

Organized: 1915
Territory: Radius of sixty miles from Birmingham, Ala.
Place of Meeting: Tutwiler Hotel
Number of Members: 91

EXECUTIVE COMMITTEE

H. S. KENT, *Chairman*
J. M. GALLALEE, *Vice-Chairman*
J. B. BELL, *Secretary-Treasurer*
R. A. POLGLAZE
C. F. VON HERRMANN, JR.

BOSTON

Organized: 1909
Territory: Radius of thirty miles from Boston, Mass.
Place of Meeting: Mass. Inst. of Technology
Local Organization: Engineering Societies of New England
Number of Members: 547

BOSTON
(*Continued*)

Executive Committee

G. K. Saurwein, *Chairman*
Kerr Atkinson, *Vice-Chairman*
R. A. Spence, *Secretary-Treasurer*
H. J. Brown
T. W. Hopper
J. W. Zeller

Junior Group

R. N. Gilbert, *Chairman*
Anton Salecker, *Vice-Chairman*
Samuel Crowell, 3d, *Secretary*
R. A. Spence, *Treasurer*
E. I. Bower
F. M. Magee

BRIDGEPORT

Organized: 1917, as a Branch of Connecticut Section; reorganized as a Section, 1923
Territory: Fairfield County, Conn.
Place of Meeting: Stratfield Hotel
Local Organization: Engineers' Club of Bridgeport
Number of Members: 120

Executive Committee

C. N. Hoagland, *Chairman*
Rudolf Beck, *Vice-Chairman*
W. H. Sniffen, *Secretary*
A. W. Hagan, *Treasurer*
A. H. Beede
C. A. Buss
I. C. Jennings
R. C. Moody
O. J. Richmond
J. W. Roe
J. D. Skinner
J. H. Van Yorx, Jr.

BUFFALO

Organized: 1915
Territory: Radius of thirty miles from Buffalo, N.Y.
Place of Meeting: University Club, Deleware Ave.
Local Organization: Engineering Society of Buffalo
Number of Members: 184

Executive Committee

W. M. Kauffman, *Chairman*
N. C. Barnard, *Vice-Chairman*
M. C. Case, *Secretary*
C. E. Harrington, *Treasurer*
J. G. Benson
Paul Dubosclard
H. M. Evarts
H. F. Kerker
J. L. Yates, *Adviser for Juniors*

CENTRAL ILLINOIS

Organized: 1937
Territory: All the territory in Central Illinois between the following counties on the northern boundary: Bureau, LaSalle, Knox, Stark, Putnam, Marshall, Livingston, Peoria; counties on the southern boundary: Pike, Scott, Morgan, Sangamon, Macon, Piatt, Douglas, and Edgar
Place of Meeting: Hotel Pere Marquette or Caterpillar Show Room
Number of Members: 104

Executive Committee

F. L. Meyer, *Chairman*
R. T. Mees, *Vice-Chairman*
C. O. Smith, *Vice-Chairman*
F. H. Thomas, *Vice-Chairman*
L. E. Johnson, *Secretary-Treasurer*
R. E. McClain, *Assistant Secretary*
M. A. Clements

Junior Group

M. A. Clements

CENTRAL INDIANA

Organized: 1916
Territory: Radius of eighty miles from Indianapolis, within Indiana
Place of Meeting: Place varies
Local Organization: Indiana Engineering Society
Number of Members: 140

Executive Committee

J. A. Drogue, *Chairman*
H. A. McAninch, *Vice-Chairman*
W. J. Cope, *Secretary-Treasurer*
G. L. Fowler
P. F. Helm
H. H. Skabo
R. W. Worley

CENTRAL PENNSYLVANIA

Organized: 1921
Territory: Radius of approximately sixty miles from State College, Pa.
Place of Meeting: State College and Altoona, Pa.
Number of Members: 75.

Executive Committee

F. C. Stewart, *Chairman*
J. O. P. Hummel, *Secretary-Treasurer*
C. L. Allen
J. S. Doolittle
W. D. Garman
G. L. Guillet
A. H. Zerban

CHICAGO

Organized: 1913
Territory: Radius of fifty miles from Chicago, Ill.
Headquarters: Mid-West A.S.M.E. Office, Room 1617, 205 West Wacker Drive, Chicago, Ill.
Place of Meeting: Civic Opera Bldg., 20 N. Wacker Dr.
Luncheon Meeting every Tuesday at 12:15 p.m. at Chicago Engineers' Club
Local Organization: Western Society of Engineers
Number of Members: 805

Executive Committee

L. M. Ellison, *Chairman*
C. C. Austin, *Vice-Chairman*
H. M. Black, *Vice-Chairman*
J. R. Michel, *Vice-Chairman*
Daniel Roesch, *Vice-Chairman*
F. B. Orr, *Secretary-Treasurer*
R. H. Bacon
J. A. Folse
W. P. Holtzman
A. H. Jens
J. S. Kozacka
F. H. Lane
J. C. Marshall
T. S. McEwan
H. L. Nachman
C. W. Parsons
V. L. Peickii
H. S. Philbrick
J. C. Reid
Karl Tranzen
R. E. Turner
C. L. Wachs

Junior Group

J. C. Marshall
S. J. Tozer

CINCINNATI

Organized: 1912
Territory: Radius of thirty miles from Cincinnati, Ohio
Place of Meeting: Engineers' Club Rooms, Ninth & Race Sts.
Local Organization: Engineers' Club of Cincinnati
Number of Members: 192

Executive Committee

E. S. Saurbrunn, *Chairman*
E. H. Mitsch, *Vice-Chairman*
J. W. Bunting, *Secretary-Treasurer*
H. B. Brandt
A. G. Bruck
T. B. Morris
L. F. Nenninger
F. P. Rhame
H. P. Thompson
H. C. Uihlein

CLEVELAND

Organized: 1918
Territory: Counties of Lorain, Cuyahoga Lake, Geauga, and Ashtabula in Ohio
Place of Meeting: Case Club or Cleveland Engineering Society Rooms
Local Organization: Cleveland Engineering Society
Number of Members: 248

Executive Committee

J. P. Dearasaugh, *Chairman*
E. R. McCarthy, *Secretary*
F. A. Barnes, *Treasurer*
A. B. Einig
J. M. Main
R. R. Slaymaker
H. A. Schwartz
A. G. Trumbull

COLORADO

Organized: 1919
Territory: Entire State of Colorado
Place of Meeting: Parisienne Rotisserie Inn, Denver, Colo.
Local Organization: Colorado Engineering Council (Colorado Society of Engineers)
Number of Members: 76

Executive Committee

R. F. Throne, *Chairman*
J. C. Reed, *Secretary-Treasurer*
L. D. Crain
A. L. Hill
F. A. Lockwood
F. H. Prouty
G. A. Richter
J. T. Strate

COLUMBUS

Organized: 1920
Territory: Counties of Union, Delaware, Licking, Madison, Franklin, Fayette, Pickaway, and Ross in Ohio
Place of Meeting: Battelle Memorial Institute and The Ohio State University
Local Organization: Engineers' Club of Columbus
Luncheon Meeting third Friday of each month at 12:00 noon at Engineers' Club, Columbus
Number of Members: 80

Executive Committee

E. M. Sampson, *Chairman*
H. R. Limbacher, *Vice-Chairman*
J. G. Lowther, *Secretary-Treasurer*
H. M. Blank
A. I. Brown
J. L. Purdy
C. P. Roberts
R. N. Tucker

DAYTON

Organized: 1926
Territory: Counties of Drake, Miami, Champaign, Preble, Montgomery, Greene, and northern part of Butler and Warren in Ohio
Place of Meeting: Engineers' Club of Dayton
Local Organization: Engineers' Club of Dayton
Number of Members: 99

Executive Committee

A. R. Weber, *Chairman*
J. J. Healy, *Vice-Chairman*
G. W. Wells, *Secretary*
A. F. Poock, *Treasurer*
C. L. Bauer
R. K. Coppock
H. M. Gano
P. H. Kemmer
Burson Treadwell

DETROIT

Organized: 1916
Territory: Radius of thirty miles from Detroit, Mich.
Place of Meeting: Place varies
Local Organization: Engineering Society of Detroit
Number of Members: 435

Executive Committee

T. Jeffords, *Chairman*
A. M. Selvey, *Secretary-Treasurer*
E. J. Abbott
J. W. Armour
B. W. Beyer, Jr.
M. L. Fox
J. F. Jarnagin
D. E. McGuire
Jesse Ormondroyd
J. P. Schechter
R. K. Weldy

Junior Group

L. W. Lentz, *Chairman*

EAST TENNESSEE

Organized: 1922
Territory: All counties in Tennessee east of the west boundary of Scott, Morgan, Cumberland, White, Warren, Coffee, Moore, Franklin; Belle County in Kentucky; and Rossville, Dade, Walker, Cattasa, Whitfield, Murray, Gordon, Chattooga in Georgia
Place of Meeting: Places varies
Local Organization: Chattanooga Engineers Club and Knoxville Technical Club
Luncheon Meeting every Monday noon at Chattanooga Engineers Club
Number of Members: 89

Executive Committee

T. C. Ervin, *Chairman*
E. Torok, *1st Vice-Chairman*
P. J. Freeman, *2nd Vice-Chairman*
John Horne, *3rd Vice-Chairman*
J. Mack Tucker, *Secretary-Treasurer*
Horace Carpenter
M. B. Conviser
P. G. Jacka
R. W. Morton
W. R. Chambers, *Past-Chairman*

ERIE

Organized: 1917
Territory: Radius of thirty miles from Erie, Pa.
Place of Meeting: Auditorium of Pennsylvania Telephone Company
Number of Members: 77

Executive Committee

C. T. Oergel, *Chairman*
McDonald S. Reed, *Vice-Chairman*
E. C. Ims, *Secretary-Treasurer*
G. W. Bach
F. G. Brinig
E. H. Horstkotte
H. B. Joyce
G. F. Linde
G. I. Rainesalo

FLORIDA

Organized: 1925
Territory: State of Florida
Place of Meeting: Various Cities in State
Local Organization: Florida Engineering Society, Gainesville, Fla.
Number of Members: 82

Executive Committee

J. H. Clouse, *Chairman*
V. C. Couchman, *1st Vice-Chairman*
H. J. B. Scharnberg, *2nd Vice-Chairman*
W. E. Drew, *Secretary-Treasurer*
John Hunter
D. W. Pinkerton
R. A. Thompson

FORT WAYNE

Organized: 1939
Territory: Counties of LaGrange, Steuben, Noble, DeKalb, Whitley, Allen, Wabash, Huntington, Wells, Adams, Miami, Blackford and Jay in Indiana; Counties of Williams, Defiance, Paulding, Van Wert and Mercer in Ohio
Local Organization: Fort Wayne Engineers' Society
Number of Members: 28

Executive Committee

W. L. Kanus, *Chairman*
F. L. Ruoff, *Vice-Chairman*
W. H. Connor, *Secretary*
F. T. McInerney, Jr., *Treasurer*
W. N. Buck
H. A. Ellis
C. H. Matson

GREEN MOUNTAIN

Organized: 1923
Territory: Entire State of Vermont and neighboring and closely related communities of Claremont and Hanover, N.H.
Place of Meeting: Springfield, Windsor, Vt., and Claremont, N.H.
Local Organization: Vermont Engineering Society
Number of Members: 36

Executive Committee

M. H. Arms, *Chairman*
C. H. Adams, *Vice-Chairman*
F. H. Canary, *Secretary-Treasurer*
H. L. Daasch
C. J. Dewell
F. T. Gear
D. T. Hamilton
C. A. Renfrew

GREENVILLE

Organized: As a Branch, 1923; as a Section, 1927
Territory: Radius of sixty miles from Greenville, S.C.
Place of Meeting: Meetings held at Greenville, Clemson College, S.C., Canton, Asheville, and Enka, N.C.
Number of Members: 41

Executive Committee

R. H. Hughes, *Chairman*
J. H. Sams, *Secretary-Treasurer*
A. D. Asbury
C. D. Blackwelder
B. E. Fernow
R. B. Fuller
W. G. Wood

HARTFORD

Organized: 1917, as Branch of Conn. Section; reorganized, 1923; New Britain Section merged with Hartford Section, July 1, 1940
Territory: Hartford County except that portion served by New Britain Section
Place of Meeting: Hartford Electric Light Company
Number of Members: 158

Executive Committee

H. F. Ramm, *Chairman*
F. O. Hoagland, *Vice-Chairman*
L. C. Smith, *Vice-Chairman*
R. D. Keller, *Secretary-Treasurer*
P. W. Bauer
S. A. Brandenburg
H. Burdick
R. F. Dow
C. N. Flagg
E. P. Herrick
B. S. Lewis
W. E. Loomis
Henry Michelsen
D. K. Morgan
W. S. Paine
C. H. Richardson
C. C. Stevens
S. H. Stoner
S. J. Teller
H. B. van Zelm

INLAND EMPIRE

Organized: 1921
Territory: East of Columbia River in State of Washington, and Counties of Okanogan and Benton, and part of Northern Idaho
Place of Meeting: Davenport Hotel, Spokane
Luncheons Wednesdays at 12:00 noon, Davenport Hotel, Spokane
Local Organization: Associated Engineers of Spokane
Numbers of Members: 28

Executive Committee

E. B. Parker, *Chairman*
Alex Lindsay, *Vice-Chairman*
A. R. Karlsten, *Secretary-Treasurer*
H. F. Gauss
D. R. Gray
H. H. Langdon

ITHACA

Organized: 1936
Territory: Radius of thirty miles from Ithaca plus following cities: Binghamton, Corning, Endicott, Geneva, Painted Post
Place of Meeting: Willard Straight Hall, Cornell Campus, Ithaca, N.Y.
Number of Members: 82

Executive Committee

R. E. Kinsman, *Chairman*
C. L. Wilder, *Vice-Chairman*
F. S. Erdman, *Secretary-Treasurer*
F. G. Switzer
M. P. Whitney
N. R. Wickersham

KANSAS CITY

Organized: 1921
Territory: Radius of sixty miles from Kansas City, Mo.
Place of Meeting: University Club
Local Organization: Engineers' Club of Kansas City
Number of Members: 158

Executive Committee

H. L. Crain, *Chairman*
J. R. Stone, *Vice-Chairman*
E. M. Bruzelius, *Secretary*
R. V. Sutherland, *Treasurer*
F. R. Applegate
G. G. Brauninger
C. E. Brown
M. A. Durland
Harold Grasse
E. D. Hay
C. Q. Ward

LOS ANGELES

Organized: 1915
Territory: South of southern boundaries of following counties: Monterey, Kings, Tulares, and Inyo, Calif.
Place of Meeting: Barker Bros. Store
Local Organization: Technical Societies of Los Angeles
Luncheon Meetings Thursdays at 12:00 noon at Engineers' Club
Number of Members: 498

Executive Committee

P. L. Armstrong, *Chairman*
E. K. Springer, *Vice-Chairman*
E. M. Wagner, *Secretary-Treasurer*
J. C. Brown
R. B. Esselman
J. S. Gallagher
J. D. Hackstaff
J. Roy Hoffman
D. A. Lyons
C. H. Shattuck
J. A. Whitaker

Junior Group

R. B. Esselman, *Chairman*

LOUISVILLE

Organized: 1922
Territory: Radius of thirty miles from Louisville, Ky.
Place of Meeting: Engineers and Architects Club of Louisville
Local Organization: Engineers and Architects Club
Number of Members: 52

Executive Committee

Melvin Sack, *Chairman*
W. F. Lucas, *Vice-Chairman*
L. L. Amidon, *Secretary*
J. K. Meyer, *Treasurer*
H. H. Fenwick
F. W. Hampton
L. R. Jackson
J. H. Romann

MEMPHIS

Organized: 1923
Territory: Radius of sixty miles from Memphis, Tenn., and eastern half of Arkansas including all the territory east of a line drawn north and south through the western boundary of the city of Little Rock
Number of Members: 21

Executive Committee

M. D. Rust, *Chairman*
J. A. Mollino, Jr., *Secretary-Treasurer*
E. J. Kueck
W. H. Roberts

METROPOLITAN

Organized: 1910
Territory: Metropolitan District, New York and New Jersey
Place of Meeting: Engineering Societies Building, 29 West 39th Street, New York, N.Y.
Number of Members: 3,253

Executive Committee

A. R. Mumford, *Chairman*
F. D. Carvin, *Secretary*
E. J. Billings, *Treasurer*
W. H. Larkin, *Chairman of Meetings and Program Committee*
T. B. Allardice
W. G. Blake
P. E. Frank
W. S. Gleeson
W. McC. McKee
C. B. Peck

Junior Group

W. W. Lawrence, *Chairman*
C. K. Holland, *Vice-Chairman*
R. F. Warner, Jr., *Secretary*
A. E. Blirer
C. C. Kirby
A. G. Oliver, Jr.
D. E. Zeliff

MID-CONTINENT

Organized: 1919
Territory: Entire State of Oklahoma; territory in Arkansas not included in Memphis Section; part of Louisiana; and territory in Texas north of the southern boundaries of the counties of Gaines, Dawson, Bordon, Scurry, Fisher, Jones, and Shackelford
Place of Meeting: Usually Mayo Hotel, Tulsa, Okla.
Luncheon Meetings with Engineers Club of Tulsa, Mondays at 12:00 noon
Local Organization: Engineers Club of Tulsa
Number of Members: 130

Executive Committee

E. C. Baker, *Chairman*
A. G. Blanchard, *Vice-Chairman*
J. D. McFarland, *Vice-Chairman*
Gwynne Raymond, *Vice-Chairman*
C. A. Stevens, *Vice-Chairman*
M. R. Wise, *Secretary*
C. O. Glasgow, *Treasurer*
E. E. Ambrosius
H. R. Auerswald
R. G. Ayers
W. L. Ducker
J. F. Eaton
T. C. Webb, Jr.

MILWAUKEE

Organized: 1904
Territory: Radius of fifty miles from Milwaukee, Wis.
Place of Meeting: Wisconsin Club
Local Organization: Engineers' Society of Milwaukee
Luncheon Meetings once each month, 3rd Wednesday at Wisconsin Club
Number of Members: 208

Executive Committee

T. Eserkaln, *Chairman*
R. J. Smith, *Secretary-Treasurer*
James Brower
Hans Dahlstrand
F. H. Dorner, Sr.
M. K. Drewry
Walter Ferris
O. A. Kisa
R. C. Newhouse
W. T. Saveland, Jr.

Junior Group

W. T. Saveland, Jr., *Chairman*
R. C. Strassman, *Secretary*
Walter Bundy
J. L. Martin
R. H. Miller
G. V. Minniberger
R. J. Smith

MINNESOTA

Organized: Minneapolis, 1913; St. Paul, 1913; the two Sections merged, 1934
Territory: Entire State of Minnesota
Place of Meeting: Minnesota Union, Univ. of Minnesota
Local Organization: Minneapolis Engineers' Club, Minnesota Federation of Architectural and Engineering Societies
Number of Members: 97

Executive Committee

L. G. Straub, *Chairman*
M. S. Wunderlich, *Vice-Chairman*
N. J. Sternal, *Secretary-Treasurer*
W. H. Erskine
C. F. Shoop
J. C. Vanselow

NEBRASKA

Organized: 1922
Territory: State of Nebraska, and Council Bluffs, Iowa
Place of Meeting: Lincoln and Omaha
Local Organization: Engineers' Club of Lincoln and Omaha
Luncheon Meeting every Wednesday noon at the Omaha Engineers' Club—4th Monday Evening at Lincoln
Number of Members: 32

Executive Committee

A. A. Luebs, *Chairman*
J. H. Colson, *Vice-Chairman*
G. A. Rogers, *Secretary-Treasurer*
G. G. Bachman
J. W. Haney
J. L. White

NEW HAVEN

Organized: 1912, reorganized, 1923
Territory: Portions of New Haven and Middlesex Counties, Conn.
Place of Meeting: Mason Laboratory, Yale University
Numbers of Members: 84

Executive Committee

W. F. Thompson, *Chairman*
L. H. Von Ohlsen, *Vice-Chairman*
F. C. Richardson, *Secretary-Treasurer*
A. L. Breckenridge
C. A. Hempstead
I. T. Hook
L. C. Lichty
W. L. Tann

NEW ORLEANS

Organized: 1916
Territory: All of Louisiana except the northern part allotted to Mid-Continent Section
Place of Meeting: Room 422, St. Charles Hotel
Local Organization: Louisiana Engineering Society
Number of Members: 100

Executive Committee

L. J. Lassalle, *Chairman*
G. R. Hammett, *Vice-Chairman*
L. J. Cucullu, *Secretary-Treasurer*
T. E. Crossan
A. M. Hill
K. P. Kammer
W. S. Nelson
D. W. Stewart

Junior Group

W. S. Nelson, *Chairman*
J. R. Rombach, Jr., *Vice-Chairman*
C. C. Burke, Jr., *Secretary*

NORWICH

Organized: 1930
Territory: Counties of Tolland, Windham, and New London in Connecticut, and Westerly District in Rhode Island
Place of Meeting: Arcanum Club, 150 Main St., Norwich
Number of Members: 37

Executive Committee

W. E. Beaney, *Chairman*
Robert Wosak, *Secretary-Treasurer*
A. D. Andriola
E. S. Dennison
W. L. Edel
F. S. English
J. S. Leonard
Hans Luehrs

NORTH TEXAS

Organized: 1922
Territory: All of Texas north of an approximately straight line through Del Rio, Fredericksburg, Georgetown, Cameron, Nacogdoches, and center, including the cities mentioned, and south of north boundaries of the counties of Parmer, Castro, Swisher, Briscoe, Hall, and Childress. Also the City of Texarkana, Ark.
Place of Meeting: Dallas Power & Light Co. Bldg. Auditorium
Local Organization: Technical Club of Dallas
Number of Members: 107

Executive Committee

R. M. Matson, *Chairman*
F. C. Justice, *Vice-Chairman*
J. K. Chattey, *Secretary-Treasurer*
Leonard Cole
J. A. Noyes
D. C. Pfeiffer
N. G. Hardy, *Ex-Officio*

ONTARIO

Organized: 1917
Territory: Province of Ontario, Canada
Place of Meeting: Hart House, University of Toronto
Number of Members: 173

Executive Committee

S. G. Clarke, *Chairman*
G. E. Ellsworth, *Secretary-Treasurer*
O. H. Anderson
H. H. Angus
W. S. Ball
A. C. Blue
D. F. Cornish
C. R. Davis
W. G. McIntosh
W. E. Micklethwaite
R. L. Rude
W. D. Sheldon
Frederick Truman

Junior Group

Frederick Truman, *Chairman*
M. F. Carriere, *Secretary-Treasurer*
J. H. Miller
W. R. Trusler

OREGON

Organized: 1919
Territory: State of Oregon and that territory in Washington within a radius of thirty miles from Portland, Ore.
Place of Meeting: Usually Public Service Bldg., Portland, Ore.
Local Organization: Oregon Society of Engineers
Number of Members: 49

Executive Committee

A. D. Hughes, *Chairman*
A. A. Osipovich, *Vice-Chairman*
G. C. Tupling, *Secretary-Treasurer*
E. N. Bates
P. L. Heslop
J. C. Othus
Tom Perry

PENINSULA

Organized: 1923
Territory: West of the east boundaries of the following counties: Emmet, Charlevoix, Antrim, Kalkaska, Missaukee, Clare, Isabell, Gratiot, Clinton, Eaton, Calhoun, and Branch, Mich.
Place of Meeting: Grand Rapids, Mich.
Luncheon Meeting Fifth Thursday noon each month
Local Organization: Engineers' Club of Grand Rapids
Number of Members: 50

Executive Committee

C. A. Hamilton, *Chairman*
R. E. Klise, *Secretary-Treasurer*
C. G. Lohmann
E. E. Norman
B. E. Porter

PHILADELPHIA

Organized: 1912
Territory: Counties of Bucks, Montgomery, Chester, Philadelphia, Delaware, Pa., and the State of Delaware
Place of Meeting: Philadelphia Engineers' Club, 1317 Spruce Street, Philadelphia, Pa.
Local Organization: Philadelphia Engineers' Club
Luncheon Meeting every Thursday noon at 12:30 p.m. at Philadelphia Engineers' Club
Number of Members: 922

Executive Committee

L. P. Hynes, *Chairman*
J. S. Morehouse, *Vice-Chairman*
C. S. Gotwals, *Secretary-Treasurer*
L. N. Gulick
E. L. Hopping
F. W. Miller

Junior Group

T. M. Pomeroy, Jr., *Chairman*
J. D. Peterson, *Vice-Chairman*
Elmer Griscom, *Secretary*
William Pegram, *Treasurer*
J. P. Clark
L. N. Gulick
R. K. Knipe
G. G. Martinson
Richard Squires
Z. T. Wobensmith

PIEDMONT—NORTH CAROLINA

Organized: As a Branch, 1923; as a Section 1927; name changed from Charlotte Section to Piedmont—North Carolina, July 1, 1940
Territory: Radius of seventy-five miles from Charlotte, N.C.
Luncheon Meeting every Monday at 1:00 p.m. at Efirds Department Store Dining Room
Local Organization: Charlotte Engineers Club
Number of Members: 43

Executive Committee

R. P. Reece, *Chairman*
T. O. Sills, *Vice-Chairman*
M. D. Thomason, *Secretary-Treasurer*
J. H. Erskine
Asa Hosmer
W. W. Leroy
W. E. McDowell
E. D. Powell
E. E. Williams

PITTSBURGH

Organized: 1920
Territory: Counties bounded by and including Beaver, Butler, Venango, Forest, Jefferson, Indiana, Somerset, Fayette, Greene, and Washington, Pa.
Place of Meeting: Engineers' Society of Western Pennsylvania, William Penn Hotel
Local Organization: Engineers' Society of Western Pennsylvania
Number of Members: 427

Executive Committee

M. M. McConnell, *Chairman*
J. A. Hunter, *Secretary*
K. F. Treschow, *Treasurer*
Alfred Butcher
S. B. Ely
B. C. McFadden

PLAINFIELD

Organized: 1921
Territory: Plainfield and territory included between Elizabeth, Bound Brook, Metuchen, and Watchung, N.J.
Place of Meeting: Elizabeth Carteret Hotel, Elizabeth, and Plainfield Masonic Temple, Plainfield
Local Organization: Plainfield Engineers Club, Singer Engineering Society
Number of Members: 171

Executive Committee

G. E. Leavitt, Jr., *Chairman*
R. C. Heck, Jr., *Vice-Chairman*
F. C. Spencer, Jr., *Secretary*
C. G. Holmberg, Jr., *Treasurer*
D. H. Chason
C. A. Dawley

PROVIDENCE

Organized: 1920
Territory: Radius of thirty miles from Providence, R.I.
Place of Meeting: Providence Engineering Society Building, 195 Angell St., Providence, R.I.
Local Organization: Providence Engineering Society
Number of Members: 157

Executive Committee

A. W. Calder, Jr., *Chairman*
E. W. Freeman, *Vice-Chairman*
R. M. Scott, *Secretary-Treasurer*
S. J. Berard
C. D. Billmeyer
E. H. Bradley
J. D. Eldert
Chester Hacking
P. V. Miller
F. A. Sawyer
H. S. Sizer

RALEIGH

Organized: As a Branch, 1923; as a Section, 1927
Territory: Radius of sixty miles from Raleigh, N.C.
Place of Meeting: N.C. State College, Raleigh, N.C.
Local Organization: N.C. Engineering Council, Raleigh Engineers Club
Number of Members: 28

Executive Committee

C. E. Kerchner, *Chairman*
R. B. Rice, *Vice-Chairman*
R. G. Chapman, *Secretary-Treasurer*
V. L. Kenyan, Jr.
F. J. Reed
L. L. Vaughan
R. S. Wilbur

ROCHESTER

Organized: 1919
Territory: Radius of thirty miles from Rochester, N.Y.
Place of Meeting: Rochester Engineering Society Rooms, Sagamore Hotel
Local Organization: Rochester Engineering Society, Sagamore Hotel
Luncheon Meeting every Tuesday at 12:15 p.m. at Sagamore Hotel
Number of Members: 111

Executive Committee

J. H. Snyder, *Chairman*
I. S. Bradley, *Vice-Chairman*
I. G. McChesney, *Secretary-Treasurer*
J. W. Gavett
K. H. Hubbard
F. D. Punnett
W. D. Wood

Junior Group

I. S. Bradley, *Chairman*
F. D. Punnett
W. D. Wood

ROCK RIVER VALLEY

Organized: 1926
Territory: Radius of thirty miles from Rockford, Ill., plus members in Madison, Wis.
Meeting Place: Place varies
Local Organization: Rockford Engineering Society
Number of Members: 68

Executive Committee

C. L. Avery, *Chairman*
C. A. Jacobson, *Vice-Chairman*
F. J. Zircher, *Secretary Treasurer*
E. L. Dahlund
G. L. Larson
A. H. Lyon
L. A. Wilson

ST. JOSEPH VALLEY

Organized: 1929
Territory: Counties of La Porte, Starke, Pulaski, St. Joseph, Marshall, Fulton, Elkhart, and Kosciusko in Indiana, and Cass and Berrien Counties in Michigan
Place of Meeting: Morningside Hotel, South Bend, Ind.
Local Organization: St. Joseph Valley Engineers' Club
Number of Members: 41

Executive Committee

C. C. Wilcox, *Chairman*
C. R. Adams, *Vice-Chairman*
K. W. Knorr, *Secretary*

ST. LOUIS

Organized: 1909
Territory: Radius of thirty miles from St. Louis, Mo.
Place of Meeting: Place varies
Local Organization: Engineers' Club of St. Louis
Number of Members: 229

Executive Committee

Albert Vigne, *Chairman*
R. W. Merkle, *Vice-Chairman*
C. B. Briscoe, *Secretary-Treasurer*
D. E. Dickey
A. L. Heintze
R. C. Thumser

SAN FRANCISCO

Organized: 1910
Territory: All territory north of the northern boundaries of the counties of San Luis Obispo, Kern, and San Bernardino
Place of Meeting: Engineers' Club, 206 Sansome St.
Luncheon Meetings, Tuesdays, California Hotel, Oakland; Thursdays, Engineers' Club, San Francisco
Local Organization: San Francisco Engineers' Club
Number of Members: 386

Executive Committee

V. F. Estcourt, *Chairman*
H. T. Avery, *Vice-Chairman*
E. H. Cameron, *Secretary-Treasurer*
H. J. Berg
E. C. Floyd
L. M. Martin
G. H. Raitt, *Ex-Officio*

Junior Group

B. S. Truett, *Chairman*
W. C. Cheal
P. E. Dawson
Charles Lippman
C. L. Thorpe
G. L. Woodfield

SAVANNAH

Organized: 1923
Territory: Radius of 125 miles from Savannah in Georgia
Place of Meeting: Savannah Hotel
Local Organization: Engineers' Council of Savannah Chamber of Commerce
Number of Members: 19

SAVANNAH

(*Continued*)

Executive Committee

W. L. Mingledorff, Jr., *Chairman*
J. G. Crowley, *Vice-Chairman*
C. O. Johnson
A. P. Keisker
S. D. Wills

SCHENECTADY

Organized: As a Branch, 1919; as a Section, 1927
Territory: Radius of thirty miles from Schenectady, N.Y.
Place of Meeting: Rice Hall
Number of Members: 193

Executive Committee

R. H. Norris, *Chairman*
R. S. Neblett, *Vice-Chairman*
Carl Schabtach, *Vice-Chairman*
O. L. Wood, Jr., *Vice-Chairman*
S. L. Jameson, *Secretary*
Stanford Neal, *Treasurer*
E. W. D. Bunke
W. R. Foote
A. R. Stevenson, Jr.

SOUTH TEXAS

Organized: 1919
Territory: South Texas and the northern part of the State not included in the North Texas Section territory
Place of Meeting: Electric Bldg., Houston, Tex.
Number of Members: 167

Executive Committee

C. W. Crawford, *Chairman*
C. L. Orr, *Vice-Chairman*
H. F. Moller, *Secretary-Treasurer*
D. D. Alton
J. W. Beretta
H. E. Degler
C. A. Hall
H. G. Hiebeler
J. J. King
E. W. McCarthy
G. E. Neville
J. G. H. Thompson
M. W. Williams

Junior Group

G. F. Fermier, *Chairman*
J. H. Howard, *Vice-Chairman*
G. W. Kline, *Secretary*

SUSQUEHANNA

Organized: 1927
Territory: Counties of Cumberland, Dauphine, Lebanon, Adams, York, and Lancaster
Place of Meeting: Engineering Society of York, and at Lancaster Twice a Year
Local Organization: Engineering Society of York and Engineers' Society of Pennsylvania, Harrisburg, Pa.
Number of Members: 78

Executive Committee

W. E. Beline, *Chairman*
O. E. Weber, *Vice-Chairman*
E. T. P. Neubauer, *Secretary*
E. E. Aughenbaugh
T. K. Breda
M. G. Leeson
H. B. Martin
Andrew Sawyer
G. L. Smith
S. P. Soling

SYRACUSE

Organized: 1920
Territory: Radius of thirty miles from Syracuse, N.Y.
Place of meeting: Ball Room of the Onondaga Hotel
Local Organization: The Technology Club of Syracuse
Number of Members: 84

Executive Committee

M. B. Moyer, *Chairman*
D. V. Shetland, *Vice-Chairman*
E. A. Failmezger, *Secretary-Treasurer*
J. W. Linford
W. E. Renner
E. K. Rhodes
G. I. Vincent

TOLEDO

Organized: 1920
Territory: Radius of thirty miles from Toledo, Ohio
Place of Meeting: University Club, Toledo, Ohio
Local Organization: Affiliated Technical Societies of Toledo
Number of Members: 60

Executive Committee

H. R. Schutz, *Chairman*
R. F. Hill, *Vice-Chairman*
H. E. Happel, *Secretary-Treasurer*
J. W. Dean
F. L. Fuller
P. P. Hale
W. C. Lang
G. Lufkin
R. H. Marker
W. R. Moran
R. J. Mugfor
Joseph Seaman
H. H. Vogel
I. F. Zarobsky

TRI-CITIES

Organized: 1920
Territory: Radius of thirty miles from Moline, Ill.
Place of Meeting: Rock Island, Ill., Moline, Ill., and Davenport, Iowa
Luncheon Meeting every Wednesday, Davenport Hotel, 12:00 noon
Number of Members: 74

Executive Committee

R. A. Cross, *Chairman*
C. D. St. Clair, *Vice-Chairman*
C. A. Carlson, *Secretary-Treasurer*
R. M. Barnes
E. G. Erickson
H. A. Kleinman

UTAH

Organized: 1923
Territory: State of Utah
Place of Meeting: University Club, Salt Lake City
Local Organization: Utah Society of Engineers
Number of Members: 35

Executive Committee

W. D. Turpin, *Chairman*
G. W. Carter, *Vice-Chairman*
R. D. Baker, *Secretary-Treasurer*
C. B. Bowman
F. A. Harris

VIRGINIA

Organized: 1919
Territory: State of Virginia
Place of Meeting: Richmond, Norfolk, Charlottesville, Roanoke, University, Petersburg
Local Organization: Central Virginia Engineers Club
Numbers of Members: 174

Executive Committee

G. C. Molleson, *Chairman*
J. B. Jones, *Vice-Chairman*
F. S. Roop, Jr., *Secretary*
R. M. Johnston, *Treasurer*
G. L. Bascome
L. R. Gardner
H. C. Hesse
D. G. Moorhead
S. B. Roberts
W. E. Segl

WASHINGTON, D.C.

Organized: 1919
Territory: District of Columbia
Place of Meeting: Auditorium, Potomac Electric Power Co., 10th & E Sts., Washington, D.C.
Number of Members: 239

Executive Committee

G. F. Jenks, *Chairman*
W. B. Ensinger, *Vice-Chairman*
M. A Mason, *Secretary-Treasurer*
G. W. Haskins
J. W. Huckert
C. E Miller
H. G. Thielscher

Junior Group

J. W. Huckert, *Chairman*

WATERBURY

Organized: 1917, as a Branch; reorganized as a Section, 1923
Territory: Litchfield County and a portion of New Haven County
Place of Meeting: Elton Hotel
Number of Members: 65

Executive Committee

W. C. Schneider, *Chairman*
R. W. Shoemaker, *Vice-Chairman*
H. C. Ashley, *Secretary-Treasurer*
A. L. Alves
C. W. Childs
A. J. German

WATERBURY
(Continued)

Junior Group

G. H. Hatch, *Chairman*
H. J. Dillon, *Secretary*
R. W. Simpson

WESTERN MASSACHUSETTS

Organized: 1922
Territory: Includes counties of Berkshire, Franklin, Hampden, and Hampshire
Place of Meeting: Highland Hotel, Springfield, Mass.
Local Organization: Engineering Society of Western Massachusetts
Number of Members: 90

Executive Committee

A. E. Benson, *Chairman*
C. F. Dupee, *Vice-Chairman*
L. G. Carlton, *Secretary-Treasurer*
R. A. Packard
E. L. Smith
J. L. Scherner, *Ex-Officio*

WESTERN WASHINGTON

Organized: 1919
Territory: State of Washington west of Columbia River with exception of territory included in 30-mile radius of Portland, Ore.
Place of Meeting: Engineers' Club, Seattle, Wash.
Local Organization: Seattle Engineers' Club
Luncheon Meetings daily at noon at Engineers' Club, Seattle
Number of Members: 115

Executive Committee

R. E. Johnson, *Chairman*
R. Walter, *Vice-Chairman*
J. E. Mylroie, *Secretary-Treasurer*
H. P. Ford
H. C. Krehbiel, Jr.
H. J. McIntyre

WEST VIRGINIA

Organized: 1925
Territory: State of West Virginia, South of Parallel 39
Place of Meeting: Charleston, W.Va.
Number of Members: 58

Executive Committee

M. S. Bloomsburg, *Chairman*
A. H. Cannon, *Vice-Chairman*
H. B. Hickman, *Secretary-Treasurer*
C. B. Cochran, *Assistant Secretary*
G. J. Huber, Jr.
E. L. Hudson
C. L. Johnson
W. C. Norton
F. LeRoy Schaefer

WORCESTER

Organized: 1915
Territory: Radius of thirty miles from Worcester, Mass.
Place of Meeting: Sanford Riley Hall, Worcester Poly. Inst.
Local Organization: Worcester Engineering Society
Number of Members: 122

Executive Committee

R. P. Kolb, *Chairman*
H. P. Crane, *Vice-Chairman*
E. K. Allen, Jr., *Secretary-Treasurer*
E. W. Armstrong
L. R. Ball
F. R. Jones
G. H. MacCullough
C. M. McMahon
F. A. Naughton, Jr.
W. M. Wilcox

YOUNGSTOWN

Organized: 1928
Territory: Counties of Trumbull, Mahoning, and Columbiana in Ohio, and Mercer and Lawrence in Pennsylvania
Place of Meeting: Republic Rubber Co. Club Rooms, Albert St., Youngstown, Ohio
Number of Members: 62

Executive Committee

H. W. Smith, *Chairman*
L. A. Kline, *Vice-Chairman*
C. W. Foard, *Secretary-Treasurer*
F. J. Bowers
W. B. Jenkins
H. E. Melin
E. O. Oyen

STUDENT BRANCHES

Article B6A, Par. 20: The Standing Committee on Relations With Colleges shall, under the direction of the Council, have supervision of the Student Branches of the Society and of such work of the Society as aims to further the education of engineers through the colleges and schools of accepted standing.

STANDING COMMITTEE, RELATIONS WITH COLLEGES

E. W. O'Brien, *Chairman* (1941)
A. C. Chick (1942)
J. I. Yellott (1943)
H. E. Degler (1944)
G. L. Sullivan (1945)

J. W. Haney, B. T. McMinn, R. H. Porter — *Advisory Members* (1941)

J. L. Hall, *Junior Adviser* (1941)

Communicate with Student Branch through Honorary Chairman

Name and Location	Year Authorized	No. of Members †	Chairman	Secretary	Honorary Chairman
Akron, Univ. of, Akron, Ohio	1924	40	L. G. Haddock	John Bezbatchenko	F. S. Griffin
Alabama Polytechnic Inst., Auburn, Ala.	1920	34	T. R. Loder	W. A. Chapman	C. R. Hixon
Alabama, Univ. of, University, Ala.	1931	22	Leonard Mandell	D. A. R. Nelson	J. M. Gallalee
Arizona, Univ. of, Tucson, Ariz.	1937	30	C. E. Chapman	J. D. Caretto	M. L. Thornburg
Arkansas, Univ. of, Fayetteville, Ark.	1910	19	Howard Jenkins	H. H. Clayton	L. C. Price
British Columbia, Univ. of, Vancouver, B.C., Can.	1938	28	C. W. Parker	G. S. Wade	H. M. McIlroy
Brown Univ., Providence, R.I.	1923	20	R. O. Love	G. P. Conrad, II	S. J. Berard
Bucknell Univ., Lewisburg, Pa.	1916	27	R. F. Stone	R. W. Donehower	W. D. Garman
California Inst. of Tech., Pasadena, Calif.	1914	50	Newell Partch	G. K. Woods	R. L. Daugherty
California, Univ. of, Berkeley, Calif.	1912	130	D. J. Graham	Homer Crooks	C. F. Garland
Carnegie Inst. of Tech., Pittsburgh, Pa.	1913	75	J. R. Schietinger	Richard Clement	D. C. Saylor
Case School of Applied Science, Cleveland, Ohio	1913	66	G. R. Graham	E. J. R. Hudec	F. H. Vose
Catholic Univ. of America, Washington, D.C.	1922	55	L. S. Brown, Jr.	Phillipp Goldmann	M. E. Weschler
Cincinnati, Univ. of, Cincinnati, Ohio	1909	113	J. H. Tarkington	Bruce Geiger	C. A. Joerger
Clarkson College of Tech., Potsdam, N.Y.	1930	53	R. C. Ward	A. W. Hogle	J. H. Davis
Clemson A.&M. College, Clemson College, S.C.	1921	37	W. E. Cline	W. L. Richbourg	B. E. Fernow
Colorado State College of A.&M. Arts, Fort Collins, Colo.	1914	27	R. S. Wilson	W. O. Sneddon	J. H. Scofield
Colorado, Univ. of, Boulder, Colo.	1914	39	J. R. Rosenkrans	James Englund	W. S. Beattie
Colorado School of Mines Division, Golden	——	19	H. W. Hicks, Jr.	D. E. Hollard	J. C. Reed
Columbia Univ., New York, N.Y.	1909				
Management Division	——	11	W. J. Jaffe	H. C. Quarles	Fred Dutcher *Walter Rautenstrauch
Mechanical Division	——	34	E. V. DeWitt	R. T. Baum	Fred Dutcher
Cooper Union, New York, N.Y.	1920				
Inst. of Tech.	——	50	Arthur Swenson	Murray Sackson	W. A. Vopat
Night School of Engineering	——	92	J. L. Alpert	James Doyle	E. A. Salma
Cornell Univ., Ithaca, N.Y.	1908	95	R. C. Ross	R. E. Ohaus	P. H. Black
Delaware, Univ. of, Newark, Del.	1929	39	Lewis Parker	A. H. Green	W. F. Lindell
Detroit, Univ. of, Detroit, Mich.	1930	74	H. W. Scott, Jr.	M. M. Calcaterra	F. J. Linsenmeyer
Drexel Inst. of Tech., Philadelphia, Pa.	1920	62	Conrad Cook	J. S. Hunter	W. J. Stevens
Duke Univ., Durham, N.C.	1935	41	H. R. Phillips	Hulme Pattinson	F. J. Reed
Florida, Univ. of, Gainesville, Fla.	1926	30	B. A. Clubbs	R. A. Roberts	R. A. Thompson
George Washington Univ., Washington, D.C.	1924	20	Robert Butterworth	John Goff	A. F. Johnson
Georgia School of Tech., Atlanta, Ga.	1915	47	W. P. McGuire	G. N. MacKenzie	R. S. Howell
Idaho, Univ. of, Moscow, Idaho	1925	51	Edgar Butts	James Gralow	H. F. Gauss
Illinois Inst. of Tech., Chicago, Ill.	1940	210	J. E. Sauvage	Thaddeus Wieczorek	Daniel Roesch
Illinois, Univ. of, Urbana, Ill.	1909	150	T. L. Jackson	E. J. Hoagland	D. G. Ryan
Iowa State College, Ames, Iowa	1919	49	P. D. Metzler	R. A. Rusk	R. E. Roudebush
Iowa, State Univ. of, Iowa City, Iowa	1913	34	E. F. Knott	R. B. Sykes	I. T. Wetzel
Johns Hopkins Univ., Baltimore, Md.	1917	46	P. G. Olson	G. D. Dobler	M. F. Spotts
Kansas State College, Manhattan, Kan.	1914	56	V. G. Mellquist	Albert Schwerin	W. A. Tripp
Kansas, Univ. of, Lawrence, Kan.	1909	31	S. E. Bunn	W. W. Starcke	H. J. Henry
Kentucky, Univ. of, Lexington, Ky.	1911	22	J. V. Kalb	D. W. Denny	C. C. Jett
Lafayette College, Easton, Pa.	1919	40	J. H. Steele	J. W. Bowman	W. G. McLean
Lehigh Univ., Bethlehem, Pa.	1911	72	Robert Caemmerer	J. H. Dudley	T. E. Jackson
Louisiana State Univ., University, La.	1916	61	J. P. Gregor	F. B. Harris	G. F. Matthes
Louisville, Univ. of, Louisville, Ky.	1928	25	Robert Gray	J. R. Strother	H. H. Fenwick
Maine, Univ. of, Orono, Maine	1910	55	H. L. Banton	S. G. Webster	I. H. Prageman
Marquette Univ., Milwaukee, Wis.	1923	31	Carl Tierney	Raymond Szedziewski	R. J. Smith
Maryland, Univ. of, College Park, Md.	1937	33	L. L. Wilson	Charles Beaumont	W. P. Green
Massachusetts Inst. of Tech., Cambridge, Mass.	1909	116	M. P. Moody	W. L. Threadgill	Alvin Sloane
Michigan College of Min. & Tech., Houghton	1930	63	G. E. Dake	S. G. Monroe	H. W. Risteen
Michigan State College, E. Lansing, Mich.	1917	59	W. J. Kingscott	R. W. Howorth	C. N. Rix
Michigan, Univ. of, Ann Arbor, Mich.	1914	74	P. A. Johnson	J. M. Hallissy	E. T. Vincent
Minnesota, Univ. of, Minneapolis, Minn.	1913	117	Gordon Ersted	Karl Behrens	C. A. Koepke
Mississippi State College, State College, Miss.	1926	40	J. B. Buescher	R. T. Staton, Jr.	H. P. Neal
Missouri School of Mines & Metallurgy, Rolla, Mo.	1930	43	Allan Summers	R. E. Fields	R. O. Jackson
Missouri, Univ. of, Columbia, Mo.	1909	51	G. L. Hibbeler	A. A. Schmudde	E. S. Gray

† As of January 1, 1941. * Faculty Adviser.

Name and Location	Year Authorized	No. of Members †	Chairman	Secretary	Honorary Chairman
Montana State College, Bozeman, Mont.	1920	39	W. R. Jeffries	Thayer Landes	R. T. Challender
Nebraska, Univ. of, Lincoln, Neb.	1909	51	W. W. Paschke	Houston Jones	J. K. Ludwickson
Nevada, Univ. of, Reno, Nev.	1928	18	William Mitchell	Harry Dawson	W. H. Davidson
Newark College of Engineering, Newark, N.J.	1924	132	G. N. Hodge	D. H. Mangnali	F. J. Burns
New Hampshire, Univ. of, Durham, N.H.	1926	36	W. A. Gardner	E. P. Nye	E. T. Donovan
New Mexico State College of A.&M. Arts, State College, New Mex.	1938	18	Carlton McGregor	William Frick	M. T. Lewellen
New Mexico, Univ. of, Albuquerque, New Mex.	1935	14	Philip Whitener	Albert Ford, Jr.	M. E. Farris
New York, College of the City of, New York, N.Y.	1922	62	Eli Schefer	Julian Delmonte	S. J. Tracy
New York University, New York, N.Y.	1909				
Aeronautic Division	——	39	P. W. O'Meara	D. C. Watson	J. M. Labberton *F. K. Teichmann
Mechanical Division	——	41	H. H. Haglund	Aurelio Pellino	J. M. Labberton
New York Univ. Evening School, New York, N.Y.	1933	51	A. J. DeMatteo	M. W. Getler	J. M. Labberton
North Carolina State College, Raleigh, N.C.	1920	42	W. A. Dickinson	J. R. Huntley	R. B. Rice
North Dakota Agricultural College, Fargo, N.D.	1929	16	Harry Sheldon	Stewart Bakken	A. W. Anderson
North Dakota, Univ. of, Grand Forks, N.D.	1923	20	Stanley Voak	Robert Chapman	A. J. Diakoff
Northeastern Univ., Boston, Mass.	1922	92			
First Division	——	..	R. W. Ireland	Earl Finkle	A. J. Ferretti
Second Division	——	..	H. J. Ferguson	Richard McManus	A. J. Ferretti
Northwestern Univ., Evanston, Ill.	1935	35	W. M. Rohsenow	L. V. Sloma	E. F. Obert
Notre Dame, Univ. of, Notre Dame, Ind.	1929	30	Robert Odenbach	Frank Cross	C. C. Wilcox
Ohio Northern Univ., Ada, Ohio	1922	19	John Gertz	Merlin Sharer	J. A. Needy
Ohio State Univ., Columbus, Ohio	1911	45	W. H. Kuhn	W. R. Campbell	Paul Bucher
Oklahoma A.&M. College, Stillwater, Okla.	1921	28	John Stewart	George Graff, Jr.	V. L. Maleev
Oklahoma, Univ. of, Norman, Okla.	1917	95	J. D. Taylor	C. P. Brooks	D. O. Nichols
Oregon State Agricultural College, Corvallis, Ore.	1909	30	D. L. Drake	D. F. DeVine	A. D. Hughes
Pennsylvania State College, State College, Pa.	1909	76	R. W. Davis	C. L. McGarr	C. L. Allen
Pennsylvania, Univ. of, Philadelphia, Pa.	1925	33	J. C. Thompson	R. T. Vogdes, Jr.	L. N. Gulick
Pittsburgh, Univ. of, Pittsburgh, Pa.	1917	58	H. G. Skinner	John Proven	G. P. Manifold
Polytechnic Inst. of Brooklyn, Brooklyn, N.Y.	1909				
Day Division	——	46	J. A. Lawrence	H. B. Nelson	A. T. Kniffen
Evening Division	——	11	H. P. Northrup	Frank Hambrecht	A. T. Kniffen
Pratt Inst., Brooklyn, N.Y.	1923	78	F. D. Allman	V. F. Clark	J. W. Hunter
Princeton Univ. Princeton, N.J.	1926	25	F. I. Walsh, Jr.	William Callery	L. F. Rahm
Puerto Rico, Univ. of, Mayaguez, P.R.	1923	25	P. H. Rozas	E. T. Acha	L. A. Stefani
Purdue Univ., W. Lafayette, Ind.	1909	138	T. P. Peppler	C. H. Rockwood	W. J. Cope
Rensselaer Polytechnic Inst., Troy, N.Y.	1910	73	C. L. Martinez	W. C. Osborne	H. A. Wilson
Rhode Island State College, Kingston, R.I.	1930	34	R. R. Afflick	E. J. Feeley, Jr.	C. D. Billmyer
Rice Inst., Houston, Tex.	1926	30	V. B. Meyer	H. H. Orech	A. H. Burr
Rose Polytechnic Inst., Terre Haute, Ind.	1926	31	J. A. Jones	J. A. Lohr	Carl Wischmeyer
Rutgers Univ., New Brunswick, N.J.	1920	35	A. M. Lipsky	N. B. Bagger	N. P. Bailey
Santa Clara, Univ. of, Santa Clara, Calif.	1925	19	Eugene Stephens	Edward McFadden	R. A. Seban
South Dakota State College, Brookings, S.D.	1935	19	Gale House	Don Walin	R. E. Gibbs
Southern California, Univ. of, Los Angeles, Calif.	1929	51	Robert Hoffman	Charles Hurd	William Shallen-berger
Southern Methodist Univ., Dallas, Tex.	1933	18	W. O. Ramsey	Dick Turner	C. H. Shumaker
Stanford Univ., Stanford University, Calif.	1909	31	W. H. Cilker	R. P. Jackson	A. L. London
Stevens Inst. of Tech., Hoboken, N.J.	1908	74	H. R. Roome	C. G. Hebenstreit	E. H. Fezandie
Swarthmore College, Swarthmore, Pa.	1921	16	L. H. Wolfe	C. W. Beck	C. G. Thatcher
Syracuse Univ., Syracuse, N.Y.	1912	36	Howard Hoke	Theodore Foster	S. T. Hart
Tennessee, Univ. of, Knoxville, Tenn.	1923	25	T. C. Searle	Hughes Hall	R. W. Morton
Texas, A.&M. College of, College Station, Tex.	1921	179	J. J. Walker	E. R. Clark	V. M. Faires
Texas Technological College, Lubbock, Tex.	1930	41	W. E. Bauman	G. G. Fairley	H. L. Kipp
Texas, Univ. of, Austin, Tex.	1921	82	A. D. Payne	Austin Leach	M. L. Begeman
Toronto, Univ. of, Toronto, Ont., Can.	1933	51	J. R. Doyle	F. M. Bond	R. C. Wiren
Tufts College, Tufts College, Mass.	1917	39	William Lynch	J. R. Peterson	Edgar MacNaughton
Tulane Univ. of Louisiana, New Orleans, La.	1933	34	B. L. Levy	Arthur Grant, Jr.	E. R. Stephan
U.S. Naval Academy, Postgraduate School, Annapolis, Md.	1925	..	....	....	P. J. Kiefer
Utah, Univ. of, Salt Lake City, Utah	1923	25	D. A. Berg	Ben Shaver	M. B. Hogan
Vanderbilt Univ., Nashville, Tenn.	1928	21	H. B. Tomlin, Jr.	C. K. Dillingham, Jr.	S. H. Acker
Vermont, Univ. of, Burlington, Vt.	1922	9	E. M. Creed	R. G. Ramsdell, Jr.	H. L. Daasch
Villanova College, Villanova, Pa.	1925	30	F. A. Bergner	V. J. A. Gordon	W. J. Barber
Virginia Polytechnic Inst., Blacksburg, Va.	1915	65	J. Q. Peeples	E. W. Christ	F. S. Roop, Jr.
Virginia, Univ. of, University, Va.	1923	..	W. A. Green	M. L. Brown	A. G. Macconochie
Washington, State College of, Pullman, Wash.	1920	37	H. E. Hunt	C. W. Peters	F. W. Candee
Washington Univ., St. Louis, Mo.	1911	25	C. A. Feichtinger	E. E. Wallace	Herbert Kuenzel
Washington, Univ. of, Seattle, Wash.	1917	53	Herbert Chatterton	Clayton Nichols	R. W. Crain
West Virginia Univ., Morgantown, W. Va.	1922	23	J. L. Hawley	G. M. Frisch	L. D. Hayes
Wisconsin, Univ. of, Madison, Wis.	1909	109	R. V. Wright	W. F. Zunke	E. T. Hansen
Worcester Polytechnic Inst., Worcester, Mass.	1914	50	George Knauff	Chandler Walker	E. W. Armstrong
Wyoming, Univ. of, Laramie, Wyo.	1925	31	Wayne Leek	Stanley Abramson	R. S. Sink
Yale Univ., New Haven, Conn.	1910	19	John Markell, Jr.	P. N. Strobell	S. W. Dudley

† As of January 1, 1941. * Faculty Adviser.

RESEARCH COMMITTEES

ARTICLE B6A, PAR. 24: The Standing Committee on Research shall, under the direction of the Council, have supervision of the research activities of the Society.

The first Standing Committee on Research was organized in 1909.

STANDING COMMITTEE

E. G. BAILEY, *Chairman* (1941)
W. TRINKS (1942)
M. D. HERSEY (1943)
C. H. WALKER (1944)
W. R. ELSEY (1945)

LUBRICATION

Appointed October, 1915, to investigate the fundamental problems of lubrication, to formulate results of investigations previously made, and to keep in touch with contemporary research in this field

(Reorganized May, 1936)

G. B. KARELITZ, *Chairman*
S. J. NEEDS, *Secretary*
A. L. BEALL
OSCAR BRIDGEMAN
W. E. CAMPBELL
H. A. EVERETT
A. E. FLOWERS
J. C. GENIESSE
RAYMOND HASKELL
M. D. HERSEY
B. F. HUNTER
C. M. LARSON
F. C. LINN
G. L. NEELY
B. L. NEWKIRK
E. S. PEARCE
ERNEST WOOLER

FLUID METERS

Appointed 1916 to develop the theory of fluid meters of all kinds and to report on the best methods for their installation and use

(Reorganized July, 1926)

R. J. S. PIGOTT, *Chairman*
J. R. CARLTON, *Secretary*
H. S. BEAN
S. R. BEITLER
E. O. BENNETT
R. K. BLANCHARD
B. O. BUCKLAND
LOUIS GESS
A. J. KERR
T. H. KERR
M. P. O'BRIEN
W. S. PARDOE
L. K. SPINK
R. E. SPRENKLE
E. C. M. STAHL
T. R. WEYMOUTH
M. J. ZUCROW

THERMAL PROPERTIES OF STEAM

Appointed in December, 1921, to direct research on the thermal properties of water-vapor and steam from 0 C to the upper limits of temperature and pressure

(Reorganized April, 1929)

W. L. ABBOTT, *Vice-Chairman*
H. N. DAVIS
H. C. DICKINSON
A. M. GREENE, JR.
R. C. H. HECK
D. S. JACOBUS
MAX JAKOB
J. H. KEENAN
F. G. KEYES
L. S. MARKS
G. A. ORROK
R. J. S. PIGOTT
H. V. RASMUSSEN
E. L. ROBINSON

STRENGTH OF GEAR TEETH

Appointed in December, 1921. Is investigating factors affecting the strength and life of gear teeth

R. E. FLANDERS, *Chairman*
C. H. LOGUE, *Secretary*
EARLE BUCKINGHAM
A. M. GREENE, JR.
C. W. HAM
F. E. MCMULLEN
E. W. MILLER
ERNEST WILDHABER

CUTTING OF METALS

Appointed in September, 1923. Is studying the problems of metal cutting, including tool materials, tool design, lubrication, cooling, and speeds and feeds

M. F. JUDKINS, *Chairman*
L. N. GULICK, *Secretary*
L. P. ALFORD
O. W. BOSTON
R. C. DEALE
A. L. DELEEUW
C. M. THOMPSON, JR.

MECHANICAL SPRINGS

Appointed May, 1924, to determine the status of the mechanical-spring art, to promote and conduct necessary and adequate research, and to develop the art to the point of standardization

J. R. TOWNSEND, *Chairman*
C. T. EDGERTON, *Secretary*
C. E. BARBA
R. W. COOK
W. T. DONKIN
RUPEN EKSERGIAN
G. E. HANSEN
BENJAMIN LIEBOWITZ
DAVID LOFTS
(R. D. BRIZZOLARA, *Alternate*)
D. J. MCADAM, JR.
R. E. PETERSON
J. W. ROCKEFELLER, JR.
B. W. ST. CLAIR
M. F. SAYRE
T. R. WEBER
KEITH WILLIAMS
J. K. WOOD
F. P. ZIMMERLI
O. B. ZIMMERMAN

ELEVATORS

Appointed June, 1924, as a subcommittee of the Sectional Committee on Safety Code for Elevators, to study the function and operation of elevator safeties and buffers and their associated mechanisms and to develop methods of test for the approval of elevator safety devices

(Reorganized August, 1940)

D. J. PURINTON, *Chairman*
D. L. LINDQUIST, *Vice-Chairman*
G. H. REPPERT (*Alternate*)
J. A. DICKINSON, *Secretary*
M. G. LLOYD (*Alternate*)
E. M. BOUTON
E. B. DAWSON (*Alternate*)
K. A. COLAHAN
G. P. KEOGH
F. PAVLICEK (*Alternate*)
J. J. MATSON
M. B. MCLAUTHLIN
C. R. CALLAWAY (*Alternate*)
W. S. PAINE
J. L. KEANE (*Alternate*)
C. A. PETERS, JR.

EFFECT OF TEMPERATURE ON THE PROPERTIES OF METALS

Appointed December, 1924, as a joint research committee of the A.S.T.M. and the A.S.M.E. to encourage the investigation and accumulation of data on the properties of metals used in the mechanic arts at extremely high and low temperatures

N. L. MOCHEL, *Chairman*
H. J. KERR, *Vice-Chairman*
J. W. BOLTON, *Secretary*
R. H. ABORN
W. H. ARMACOST
A. B. BAGSAR
A. D. BAILEY
F. E. BASH
C. L. CLARK
E. S. DIXON
F. B. FOLEY
J. R. FREEMAN, JR.
H. J. FRENCH
H. W. GILLETT
A. J. HERZIG
G. F. JENKS
J. J. KANTER
C. E. MACQUIGG
P. E. MCKINNEY
E. L. ROBINSON
A. E. WHITE
Director, National Bureau of Standards, U.S. Department of Commerce
Representative of Bureau of Ships, U.S. Navy Department

BOILER FEEDWATER STUDIES

Appointed March, 1925, as a Joint Research Committee of the American Boiler Manufacturers Association, American Railway Engineering Association, American Water Works Association, Edison Electric Institute, the American Society for Testing Materials, and the A.S.M.E. to study methods of analysis and treatment of boiler feedwater for stationary and railroad practice

BOILER FEEDWATER STUDIES

(Continued)

Executive Committee (Total personnel 41)

C. H. Fellows, *Chairman*
R. C. Bardwell, *Vice-Chairman*
J. B. Romer, *Secretary*
A. G. Christie *
R. E. Coughlin
B. W. De Geer
Max Hecht
H. E. Jordan
P. B. Place
S. T. Powell
F. N. Speller
M. F. Stack
E. H. Tenney
A. E. White *

CONDENSER TUBES

Appointed May, 1925, to investigate and report on the causes of failure of tubes used in steam condensers and similar heat interchange apparatus

A. E. White, *Chairman*
D. C. Weeks, *Vice-Chairman*
P. A. Bancell
H. Y. Bassett
R. A. Bowman
D. K. Crampton
C. A. Crawford
H. M. Cushing
R. E. Dillon
J. R. Freeman, Jr.
V. M. Frost
C. F. Harwood
G. C. Holder
W. C. Holmes
W. B. Price
M. F. Stack
H. A. Staples
W. R. Webster
Director, Bureau of Ships, U.S. Navy Department

WORM GEARS

Appointed May, 1927, to investigate certain problems in connection with the action of worm gear drives and to recommend improvements in their design, manufacture, and use

Earle Buckingham, *Chairman*
G. H. Acker
L. R. Buckendale
D. L. Lindquist
A. A. Ross
B. F. Waterman
Representative of Bureau of Ships, U.S. Navy Department

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* Official A.S.M.E. representatives serving on this committee.

MEASURES OF MANAGEMENT

Appointed March, 1928, to attempt the reconciliation of certain economic laws affecting production, to develop formulas for management, and to collect and report information on management research

W. E. Freeland, *Chairman*
F. E. Raymond, *Secretary*
J. H. Barber
T. H. Brown
R. C. Davis
G. E. Hagemann

STRENGTH OF VESSELS UNDER EXTERNAL PRESSURE

Appointed June, 1929, to develop reliable design data on the strength of cylindrical and spherical surfaces under external pressure, particularly with reference to jacketed vessels

W. D. Halsey, *Chairman*
F. V. Hartman
M. B. Higgins
A. W. Limont, Jr.
H. E. Saunders
E. E. Shanor
D. B. Wesstrom
F. S. G. Williams
D. F. Windenburg

WIRE ROPE

Appointed April, 1930, to investigate existing rope so that it may be better understood and more effectively used

W. H. Fulweiler, *Chairman*
H. LeR. Brink
D. L. Lindquist
G. W. Martin
A. H. McDougall
B. V. E. Nordberg
W. S. Paine
W. J. Ryan
George Simpson
L. E. Young

CRITICAL PRESSURE STEAM BOILERS

Appointed June, 1931, to study the characteristics of high-pressure forced-circulation steam-generating units

H. L. Solberg, *Chairman*
W. H. Armacost
A. D. Bailey
E. G. Bailey
F. S. Clark
C. H. Fellows
H. J. Kerr
G. A. Orrok
E. C. Petrie
E. L. Robinson
P. W. Thompson

COTTONSEED PROCESSING

Appointed December, 1932, to study the mechanical problems involved in storing, conditioning, and cooking cottonseed meats

W. R. Woolrich, *Chairman*
Homer Barnes
C. E. Garner
J. F. Leahy
R. W. Morton
B. J. Sams
R. B. Taylor

ROLLING OF STEEL (PLASTICITY)

Appointed October, 1938, to study plasticity in the particular field of rolling of steel

A. Nadai, *Chairman*
E. C. Bain
C. L. Eksergian
J. H. Hitchcock
G. B. Karelitz
C. W. MacGregor
Morris Stone
W. Trinks

A.S.M.E. Representatives on Other Research Committees

See also A.S.M.E. Representatives on Other Activities, page RI-9

AMERICAN COORDINATING COMMITTEE ON CORROSION

American Society for Testing Materials

S. L. Kerr
(C. H. Fellows, *Alternate*)

CORROSION COMMITTEE

American Society of Refrigerating Engineers

(To be appointed)

FATIGUE PHENOMENA OF METALS

American Society for Testing Materials

C. T. Edgerton

HEAT-TREATMENT OF ROCK DRILL STEELS

Advisory Board of the National Bureau of Standards and Bureau of Mines

(To be appointed)

METALLURGICAL RESEARCH

Advisory Committee to the National Bureau of Standards

C. H. Bierbaum

PROPERTIES OF REFRACTORY MATERIALS

Advisory Committee to the National Bureau of Standards

E. B. Powell

WATER FOR INDUSTRIAL USES

American Society for Testing Materials

J. H. Walker

STANDARDIZATION COMMITTEES

ARTICLE B6A, PAR. 23: The Standing Committee on Standardization shall advise the Council on the dimensional standardization work of the Society, including relations with the American Standards Association.

The first Standing Committee on Standardization was organized in April, 1911

STANDING COMMITTEE

A. L. BAKER, *Chairman* (1941)
J. E. LOVELY (1942)
L. T. KNOCKE (1943)
T. E. FRENCH (1944)
W H. HILL (1945)

STANDARDIZATION AND UNIFICATION OF SCREW THREADS (B1)

** Joint sponsorship with the Society of Automotive Engineers. Sectional Committee originally organized in June, 1921. Reorganized in February, 1929*

A.S.M.E. Members (Total personnel, 35)

R. E. FLANDERS, *Chairman* †
EARLE BUCKINGHAM, *Secretary*
E. J. BRYANT
G. S. CASE
T. G. CRAWFORD
A. M. HOUSER †
H. C. E. MEYER
P. V. MILLER †
W. C. MUELLER
R. H. PERRY
O. B. ZIMMERMAN

SUBCOMMITTEE CHAIRMEN

No. 1 on Scope, Arrangement, and Editing of American National Standard, R. E. FLANDERS
No. 2 on Terminology and Thread Specifications, Except Gages, C. W. BETTCHER
No. 3 on Special Threads and Twelve Pitch Series, Except Gages (to be appointed)
No. 4 on Acme and Other Similar Threads, Except Gages, EARLE BUCKINGHAM
No. 5 on Screw Thread Gages and Inspection, G. S. CASE
No. 7 on Wood Screws, ARTHUR BOOR
Special Subcommittee on Revision of American Standard, P. V. MILLER

PIPE THREADS (B2)

** Joint sponsorship with the American Gas Association. Sectional Committee originally organized in 1913. Reorganized May, 1927*

A.S.M.E. Members (Total personnel, 48)

A. S. MILLER, *Chairman*
C. B. LEPAGE, *Acting Secretary*
A. F. BREITENSTEIN †
E. J. BRYANT
C. S. COLE
E. S. CORNELL, JR.
J. J. CROTTY
A. P. DENTON
J. J. HARMAN
A. M. HOUSER †
A. H. JARECKI
P. V. MILLER †
F. H. MOREHEAD
W. C. MORRIS
S. F. NEWMAN
L. N. SHANNON
FRANK THORNTON, JR.
J. H. WILLIAMS

SUBCOMMITTEE CHAIRMEN

No. 1 on Editing and Gaging, A. M. HOUSER
No. 2 on Taper Pipe Threads, S. B. TERRY
No. 3 on Straight Pipe Threads, A. S. MILLER
No. 4 on Plumbers' Threads, A. F. BREITENSTEIN
No. 5 on Screw Threads for Rigid Steel Conduit, JAMES BARTON
No. 6 on Special Threads for Thin Tubes, C. C. WINTER
Special Subcommittee on Tolerances on Thread Elements, E. J. BRYANT
Special Editing Subcommittee on Taper Pipe Threads, S. B. TERRY
Special Editing Subcommittee on Straight Pipe Threads, PAUL MILLER
Special Subcommittee on Truncation, E. J. BRYANT

BALL AND ROLLER BEARINGS (B3)

** Joint sponsorship with the Society of Automotive Engineers. Sectional Committee organized December, 1920*

A.S.M.E. Members (Total personnel, 20)

W. P. KENNEDY, *Vice-Chairman* †
D. E. BATESOLE †
L. A. CUMMINGS
OSCAR H. DORER
F. G. HUGHES
G. E. HULSE †
L. F. NENNINGER
S. M. WECKSTEIN †
ERNEST WOOLER

ALLOWANCES AND TOLERANCES FOR CYLINDRICAL PARTS AND LIMIT GAGES (B4)

** Sole sponsorship. Sectional Committee originally organized in June, 1920. Reorganized in September, 1930*

A.S.M.E. Members (Total personnel, 43)

J. E. LOVELY, *Chairman* †
F. E. BANFIELD, JR.
F. S. BLACKALL, JR.
E. J. BRYANT
EARLE BUCKINGHAM †
F. H. COLVIN †
T. G. CRAWFORD
R. E. W. HARRISON
F. O. HOAGLAND
N. E. JACOBI
H. C. E. MEYER
P. V. MILLER
W. C. MUELLER
E. C. PECK †
R. H. PERRY
W. C. SCHOENFELDT
C. C. STEVENS
O. B. ZIMMERMAN

SUBCOMMITTEE CHAIRMAN

No. 1 on Tolerance Systems, R. E. W. HARRISON

SMALL TOOLS AND MACHINE TOOL ELEMENTS (B5)

** Joint sponsorship with the National Machine Tool Builders Association and the Society of Automotive Engineers. Sectional Committee organized September, 1922*

A.S.M.E. Members (Total personnel, 25)

W. C. MUELLER, *Chairman* †
F. O. HOAGLAND, *Vice-Chairman*
J. B. ARMITAGE
O. W. BOSTON
E. J. BRYANT
EARLE BUCKINGHAM
F. H. COLVIN †
S. A. EINSTEIN
H. E. HARRIS †
JOHN HAYDOCK
J. P. LAUX †
J. E. LOVELY
A. F. MURRAY †
ERIK OBERG
FRANK THORNTON, JR.

TECHNICAL COMMITTEES

EXECUTIVE COMMITTEE

A.S.M.E. Members (Total personnel, 3)

W. C. MULLER, *Chairman* †
F. O. HOAGLAND, *Vice-Chairman*
H. E. HARRIS †

NO. 1 ON T-SLOTS

A.S.M.E. Members (Total personnel, 7)

ERIK OBERG, *Chairman* †
J. B. ARMITAGE
HARRY CADWALLADER, JR.†
S. A. EINSTEIN
F. O. HOAGLAND †

NO. 2 ON TOOL POSTS AND TOOL SHANKS

A.S.M.E. Members (Total personnel, 8)

O. W. BOSTON, *Chairman*
F. S. BLACKWALL, JR.
GRANGER DAVENPORT †
M. E. LANGE

NO. 3 ON MACHINE TAPERS

A.S.M.E. Members (Total personnel, 21)

E. J. BRYANT, *Chairman* †
C. B. LEPAGE, *Acting Secretary*
J. B. ARMITAGE
F. S. BLACKALL, JR.
EARLE BUCKINGHAM †
F. H. COLVIN
J. B. DILLARD
(T. F. GITHENS, *Alternate*)
B. P. GRAVES
H. E. HARRIS
F. O. HOAGLAND †
J. H. HORIGAN
A. H. LYON
L. F. NENNINGER

SUBGROUP CHAIRMEN

Steep Tapers Series, S. MCMULLAN
Revision of Slow Taper Standard, E. J. BRYANT

* Note: All of these standards committees for which the Society is sponsor or joint sponsor, or on which it has representation, are organized under the procedure of the American Standards Association.

† Official A.S.M.E. representative serving on this committee.

SMALL TOOLS AND MACHINE TOOL ELEMENTS (B5)

(*Continued*)

No. 4 on Spindle Noses and Collets for Machine Tools

A.S.M.E. Members (Total personnel, 26)

J. E. Lovely, *Chairman* †
L. F. Nenninger, *Secretary*
J. B. Armitage
B. P. Graves
F. O. Hoagland †
A. M. Johnson
M. E. Lange
J. H. Mansfield
L. D. Spence

SUBGROUP CHAIRMEN

No. 1 on Milling Machines, Small and Medium, J. B. Armitage
No. 2 on Large Milling Machines, F. B. Kampmeier
No. 3 on Grinding Machine Spindles, H. J. Griffing
No. 5 on Drilling Machines and Horizontal Boring Machines, S. McMullan
No. 6 on Turning Machines, Including Automatic Screw Machines, Lathes, Automatic Lathes, Turret Lathes, and Automatic Chucking Machines, J. E. Lovely
No. 8 on Correlation of Counter Proposals for Spindle Noses, J. E. Lovely

No. 5 on Milling Cutters

A.S.M.E. Members (Total personnel, 20)

J. B. Armitage
A. N. Goddard †
J. H. Horigan
G. L. Markland, Jr.
E. K. Morgan
Erik Oberg †
E. D. Vancil

SUBGROUP CHAIRMEN

No. 1 on Profile Cutters, E. D. Vancil
No. 2 on Keyways, J. B. Armitage
No. 3 on Nomenclature, A. C. Danekind
No. 4 on Limits, J. H. Horigan
No. 5 on Formed Cutters, H. C. Hungerford
No. 6 on Hobs, G. L. Markland, Jr.
No. 7 on Inserted Tooth Cutters, J. B. Armitage

No. 6 on Designations and Working Ranges of Machine Tools

A.S.M.E. Members (Total personnel, 19)

John Haydock, *Chairman*
Earle Buckingham †
T. H. Doan, Jr.
B. P. Graves †
J. J. McBride
E. R. Smith

No. 7 on Twist Drill Sizes

A.S.M.E. Members (Total personnel, 6)

W. C. Mueller, *Chairman* †
J. H. Horigan †

No. 8 on Jig Bushings

A.S.M.E. Member (Total personnel, 8)

J. H. Horigan †

No. 9 on Punch Press Tools

A.S.M.E. Members (Total personnel, 15)

D. H. Chason
N. W. Dorman
H. E. Harris †
D. M. Palmer

No. 10 on Forming Tools and Holders

A.S.M.E. Members (Total personnel, 9)

W. C. Mueller, *Chairman* †
William Hartman †
L. D. Spence

No. 11 on Chucks and Chuck Jaws

A.S.M.E. Member (Total personnel, 10)

J. E. Lovely, *Chairman* †

SUBGROUP CHAIRMEN

No. 1 on Master Chuck Jaws, J. E. Lovely
No. 2 on Adapters for Air Cylinders, J. E. Lovely

No. 12 on Cut and Ground Thread Taps

(Total personnel, 7)

No. 13 on Splines and Splined Shafts

A.S.M.E. Members (Total personnel, 14)

J. B. Armitage †
R. E. W. Harrison
F. O. Hoagland
J. E. Lovely †
B. F. Waterman

No. 17 on Nomenclature for Small Tools and Machine Tool Elements

A.S.M.E. Members (Total personnel, 12)

O. W. Boston, *Chairman and Secretary*
F. S. Blackall, Jr.
F. H. Colvin †
H. E. Harris
F. O. Hoagland

Ex-Officio Members

A. N. Goddard
W. C. Mueller †

No. 19 on Single-Point Cutting Tools

A.S.M.E. Members (Total personnel, 2)

F. H. Colvin, *Chairman* †
O. W. Boston, *Secretary*

No. 20 on Reamers

A.S.M.E. Members (Total personnel, 16)

F. H. Colvin
T. F. Githens †
J. H. Horigan †
H. E. Wells

SUBGROUP CHAIRMAN

No. 1 on Reamer Proposal, C. M. Pond

No. 21 on Tool-Life Tests for Single-Point Tools

A.S.M.E. Members (Total Personnel, 11)

O. W. Boston, *Chairman*
M. F. Judkins

No. 22 Standards of Accuracy for Engines Lathes

Note.—Sectional Committee B5 recognized the Committee on Standards of the Lathe Group of The National Machine Tool Builders' Association as the personnel of this technical committee.

GEARS (B6)

** Joint sponsorship with the American Gear Manufacturers Association. Sectional Committee organized June, 1921*

A.S.M.E. Members (Total personnel, 27)

B. F. Waterman, *Chairman*
Earle Buckingham, *Vice-Chairman* †
C. B. LePage, *Acting Secretary*
G. H. Acker
U. S. Eberhardt
L. H. Fry
C. B. Hamilton, Jr.
D. T. Hamilton
M. R. Hanna
O. A. Leutwiler †
G. L. Markland, Jr.
Carleton Reynell

SUBCOMMITTEE CHAIRMEN

Executive Committee, B. F. Waterman
No. 1 on Program, B. F. Waterman
No. 2 on Editing Reports, B. F. Waterman
No. 3 on Nomenclature, D. T. Hamilton
No. 4 on Tooth Form (Spur Gears), U. S. Eberhardt
No. 5 on Helical Gears, W. P. Schmitter
No. 6 on Worm Gears, T. R. Rideout
No. 7 on Bevel Gears, F. L. Knowles
No. 8 on Materials, C. B. Hamilton, Jr.
No. 9 on Inspection, J. P. Breuer
No. 10 on Horsepower Rating, Earle Buckingham

PIPE FLANGES AND FITTINGS (B16)

** Joint sponsorship with the Heating, Piping, and Air Conditioning Contractors National Association and the Manufacturers Standardization Society of the Valve and Fittings Industry. Sectional Committee organized October, 1921*

A.S.M.E. Members (Total personnel, 48)

C. P. Bliss, *Chairman* †
J. J. Harman, *Secretary*
L. W. Benoit †
A. L. Brown
Sabin Crocker
Ferdinand Fink
H. E. Haller
J. S. Hess
H. A. Hoffer †
E. L. Hopping
A. M. Houser
D. S. Jacobus
C. A. Kelting †
J R. Kruse (John Blizard, *Alternate*)
M. B. MacNeille
F. H. Morehead
L. S. Morse
Ludwig Skog
J. R. Tanner †
J. H. Taylor
H. L. Underhill
G. W. Watts
J. H. Williams

SUBCOMMITTEE CHAIRMEN

Executive Committee, C. P. Bliss †
No. 1 on Cast Iron Flanges and Flanged Fittings, A. M. Houser
No. 2 on Screwed Fittings, F. H. Morehead
No. 3 on Steel Flanges and Flanged Fittings, C. P. Bliss

PIPE FLANGES AND FITTINGS (B16)

(*Continued*)

No. 4 on Materials and Stresses, A. M. Houser
No. 5 on Face to Face Dimensions of Ferrous Flanged Valves, J. R. Tanner
No. 6 on Malleable Iron or Steel Brass Seat Unions (to be appointed)
No. 7 on Rating of Pipe Fittings (to be appointed)
No. 8 on Marking of Pipe Fittings, F. H. Moreland
No. 9 on Port Openings, W. W. Hubbard

SHAFTING (B17)

** Sole sponsorship. Organized October, 1918*

A.S.M.E. Members (Total personnel, 13)

C. M. Chapman, *Chairman* †
C. B. LePage, *Secretary*
H. C. E Meyer
L. C. Morrow
J. M. Shimer
G. N. Van Derhoef †
L. W. Williams †

BOLT, NUT, AND RIVET PROPORTIONS (B18)

** Joint sponsorship with the Society of Automotive Engineers. Sectional Committee organized March, 1922*

A.S.M.E. Members (Total personnel, 52)

H. E. Aldrich
F. C. Billings
B. G. Braine
G. S. Case
T. G. Crawford
H. P. Frear
A. M. Houser †
Herman Koester
S. F. Newman
R. J. Whelan
E. M. Whiting
V. R. Willoughby
(J. J. McBride, *Alternate*)
O. B. Zimmerman

SUBCOMMITTEE CHAIRMEN

No. 1 on Large and Small Rivets (to be appointed)
No. 2 on Wrench-Head Bolts and Nuts, W. K. Mendenhall, Jr.
No. 3 on Slotted Head Proportions (to be appointed)
No. 4 on Track Bolts and Nuts (to be appointed)
No. 5 on Round Unslotted Head Bolts (Carriage Bolts), M. C. Horine
No. 6 on Plow Bolts, O. B. Zimmerman
No. 7 on Body Dimensions and Materials (to be appointed)
No. 8 on Nomenclature, G. S. Case
No. 9 on Socket Head Cap and Set Screws, Herman Koester

PLAIN AND LOCK WASHERS (B27)

** Joint sponsorship with the Society of Automotive Engineers. Sectional Committee organized August, 1925*

A.S.M.S. Members (Total personnel, 37)

Eugene Caldwell
T. G. Crawford
B. S. Lewis †
C. H. Loutrel
J. J. McBride
H. C. E. Meyer
W. C. Mueller †
E. M. Whiting †
O. B. Zimmerman

SUBCOMMITTEE CHAIRMEN

No. 1 on Plain Washers, W. L. Barth
No. 2 on Spring Washers, E. Cowlin

TRANSMISSION CHAINS AND SPROCKETS (B29)

** Joint sponsorship with the Society of Automotive Engineers and the American Gear Manufacturers Association. Sectional Committee organized September, 1917. Reorganized December, 1926*

A.S.M.E. Members (Total personnel, 16)

W. J. Belcher
C. B. Jahnke †
Joseph Joy
L. V. Ludy †
D. B. Perry
C. R. Weiss
G. A. Young
O. B. Zimmerman

SUBCOMMITTEE CHAIRMEN

No. 1 on Roller Chain Standardization (to be appointed)
No. 2 on Silent Chain Standardization, G. A. Young

CODE FOR PRESSURE PIPING (B31)

** Sole sponsorship. Sectional Committee organized March, 1926. Reorganized December, 1937*

A.S.M.E. Members (Total personnel, 101)

E. B. Ricketts. *Chairman*
R. E. Bryant
C. S. Cole
H. C. Cooper
D. H. Corey
Sabin Crocker
H. D. Edwards
E. R. Fish
Charles Fitzgerald
V. M. Frost
T. W. Greene
H. E. Haller
W. D. Halsey
J. S. Haug
H. A. Hoffer
G. G. Hollins
E. L. Hopping
A. M. Houser †
Alfred Iddles †
D. S. Jacobus
T. M. Jasper
C. A. Kelting
G. S. Larsen
M. B. MacNeille
G. W. Martin
H. C. E. Meyer
J. W. Moore
(J. D. Capron, *Alternate*)
F. H. Morehead
(W. W. Crawford, *Alternate*)
(J. J. Harman, *Alternate*)
H. H. Morgan
L. S. Morse
A. W. Moulder
E. W. Norris
C. W. Obert
G. A. Orrok
A. L. Penniman, Jr.
C. S. Robinson †
J. H. Romann
D. B. Rossheim
G. W. Saathoff
G. K. Saurwein
Ludwig Skog
H. S. Smith
(H. H. Moss, *Alternate*)
J. R. Tanner
J. H. Taylor
J. H. Vance
H. L. Whittemore
J. H. Williams
T. F. Wolfe

SUBCOMMITTEE CHAIRMEN

No. 1 on Plan, Scope, and Editing, Sabin Crocker
No. 2 on Power Piping, Alfred Iddles
No. 4 on Gas and Air Piping, J. S. Haug
No. 5 on Refrigeration Piping, A. B. Stickney
No. 6 on Oil Piping, A. D. Sanderson
No. 7 on Piping Materials and Identification, F. H. Morehead
No. 8 on Fabrication Details, Ludwig Skog
No. 9 on District Heating Piping, G. K. Saurwein

WIRE AND SHEET METAL GAGES (B32)

** Joint sponsorship with the Society of Automotive Engineers. Sectional Committee organized November, 1928. Reorganized November, 1939*

A.S.M.E. Members (Total personnel, 34)

A. P. Cottle
J. F. Howe †
F. G. Wilson †

SUBCOMMITTEE CHAIRMAN

Wire and Sheet Metal Gages, H. W. Tenney

SCREW THREADS FOR HOSE COUPLINGS (B33)

** Sole sponsorship. Sectional Committee organized August, 1928*

A.S.M.E. Members (Total personnel, 28)

A. L. Brown, *Secretary*
A. F. Breitenstein †
J. J. Crotty
W. L. Curtiss
W. E. Dunham †
J. J. Harman
(F. C. Ernst, *Alternate*)
A. M. Houser
H. C. E. Meyer
J. H. Williams

SUBCOMMITTEE CHAIRMEN

No. 1 to Draft Recommended Specifications (to be appointed)
No. 2 on Basic Thread Dimensions, D. R. Miller

WROUGHT IRON AND WROUGHT STEEL PIPE AND TUBING (B36)

** Joint sponsorship with the American Society for Testing Materials. Sectional Committee organized April, 1928*

A.S.M.E. Members (Total personnel, 43)

H. H. Morgan, *Chairman*
Sabin Crocker, *Secretary*
J. S. Adelson
H. E. Aldrich
E. L. Hopping
(A. B. Morgan, *Alternate*)
A. M. Houser †
D. S. Jacobus †
(F. S. Clark, *Alternate*) †
J. J. Kanter

WROUGHT IRON AND WROUGHT STEEL PIPE AND TUBING (B36)

(*Continued*)

H. C. E. Meyer
F. H. Morehead
H. B. Oatley †
Ludwig Skog †
F. N. Speller
J. R. Tanner
A. E. White

SUBCOMMITTEE CHAIRMEN

No. 1 on Plan, Scope, and Editing, H. H. Morgan
No. 2 on Pipe and Tubing for Low Temperature Service, J. J. Shuman
No. 3 on Pipe and Tubing for High Temperature Service, J. R. Tanner
No. 4 on Materials, F. H. Morehead

PRESSURE AND VACUUM GAGES (B40)

** Sole sponsorship. Sectional Committee organized July, 1930*

A.S.M.E. Members (Total personnel, 44)

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A. W. Lenderoth, *Secretary* †
E. J. Bryant
J. P. Cavanaugh †
Paul Diserens
C. H. Graesser
W. F. Jones
R. J. Kehl
J. C. McCune †
A. H. Morgan
H. B. Reynolds
W. C. Schoenfeldt

SUBCOMMITTEE CHAIRMEN

No. 1 on Plan and Scope, M. D. Engle
No. 2 on Definitions, C. F. Schwep
No. 3 on Gage Sizes and Mounting Dimensions, H. B. Reynolds
No. 4 on Accuracy and Test Methods, O. J. Hodge

STOCK SIZES, SHAPES AND LENGTHS FOR HOT AND COLD FINISHED IRON AND STEEL BARS (B41)

** Sole sponsorship. Sectional Committee organized June, 1930*

A.S.M.E. Members (Total personnel, 27)

F. H. Dechant
H. D. Tanner
L. W. Williams †
G. H. Woodroffe
O. B. Zimmerman

SUBCOMMITTEE CHAIRMEN

No. 1 on Hot Rolled Steel, Henry Wysor
No. 2 on Cold Finished Steels, L. E. Creighton
No. 3 on Hot Rolled Iron (to be appointed)

SPECIFICATIONS FOR LEATHER BELTING (B42)

** Sole sponsorship. Sectional Committee organized February, 1931*

A.S.M.E. Members (Total personnel, 24)

H. T. Coates
R. W. Drake †
King Hathaway
J. E. Rhoads
G. A. Schieren
O. B. Zimmerman

SUBCOMMITTEE CHAIRMEN

No. 1 on Standard Specifications, R. C. Bowker
No. 2 on Recommendations for Selection, Care, and Installation, G. A. Schieren

MACHINE PINS (B43)

** Joint sponsorship with the Society of Automotive Engineers, Sectional Committee organized March, 1926*

A.S.M.E. Members (Total personnel, 13)

E. J. Bryant †
J. J. McBride
H. C. E. Meyer
O. B. Zimmerman

SUBCOMMITTEES

No. 1 on Straight, Taper, and Dowel Pins (to be appointed)
No. 2 on Split Pins (to be appointed)

CLASSIFICATION AND DESIGNATION OF SURFACE QUALITIES (B46)

** Joint sponsorship with the Society of Automotive Engineers. Sectional Committee organized May, 1932*

A.S.M.E. Members (Total personnel, 63)

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E. J. Bryant
T. G. Crawford
R. C. Deale †
U. S. Eberhardt
S. A. Einstein
R. F. Gagg
W. W. Gilbert
J. J. Harman
R. E. W. Harrison †
F. V. Hartman
F. O. Hoagland
H. J. Holtzclaw
R. T. Kent
H. F. Kurtz
A. H. Lyon
M. W. Petrie
F. C. Spencer
C. C. Stevens
J. S. Tawresey
Stewart Way
C. H. Whitaker
Ernest Wooler
John Wulff

SUBCOMMITTEE CHAIRMEN

Executive Committee (to be appointed)
No. 2 on Surfaces Produced by Molds, Dies, Rolls, or Any Other Means of Deforming Materials (to be appointed)
No. 3 on Coated Surfaces, G. B. Hogaboom
No. 4 on Symbols for Indicating Surface Quality on Drawings, T. G. Crawford
No. 5 on Ways, Means, and Apparatus for Measuring Quality of Surface (to be appointed)
No. 7 on Standards for Appearance of Surfaces (to be appointed)

COMBUSTION SPACE FOR SOLID FUELS (B50)

** Sole sponsorship. Sectional Committee organized June, 1933*

A.S.M.E. Members (Total personnel, 21)

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John Hunter
A. J. Johnson
V. G. Leach †
J. P. Magos
J. F. McIntire
F. L. Meyer
C. A. Reed
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SUBCOMMITTEE CHAIRMEN

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No. 2 on Combustion and Design, B. M. Guthrie
No. 3 on Warm Air Furnaces, J. H. Manny
No. 4 on Steel Heating Boilers, W. B. Russell
No. 5. on Cast Iron Boilers, J. F. McIntire

SCHEME FOR IDENTIFICATION OF PIPING SYSTEMS (A13)

** Joint sponsorship with the National Safety Council. Sectional Committee organized June, 1922*

A.S.M.E Members (Total personnel, 33)

E. E. Ashley
W. L. Bunker
Crosby Field
E. L. Hopping
H. L. Miner
H. S. Smith
Frank Thornton, Jr.

SUBCOMMITTEE CHAIRMEN

Identification by Colors (to be appointed)
Classification, Crosby Field
Identification Markings Other Than Color (to be appointed)
Executive Committee, A. S. Hebble
Editing Subcommittee, A. S. Hebble

MINIMUM REQUIREMENTS FOR PLUMBING AND STANDARDIZATION OF PLUMBING EQUIPMENT (A40)

** Sole sponsorship. Sectional Committee organized August, 1928*

A.S.M.E. Members (Total personnel, 51)

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J. F. Carney
C. S. Cole
A. M. Houser
G. W. Martin
(A. H. Morgan, *Alternate*)
W. K. McAfee
W. R. Webster †

SUBCOMMITTEE CHAIRMEN

No. 1 on Minimum Requirements for Plumbing (to be appointed)
No. 2 on Staple Vitreous China Plumbing Fixtures, H. R. Van Sciver
No. 3 on Staple Porcelain (All Clay) Plumbing Equipment, H. R. Van Sciver
No. 4 on Enameled Sanitary Ware, A. H. Cline, Jr.
No. 5 on Traps, A. R. McGonegal
No. 6 on Brass Plumbing Products, J. L. Murphy

MINIMUM REQUIREMENTS FOR PLUMBING AND STANDARDIZATION OF PLUMBING EQUIPMENT (A40)

(*Continued*)

No. 7 on Brass Fittings for Flared Copper Tubes, F. L. RIGGIN
No. 8 on Cast Iron Soil Pipe and Fittings (to be appointed)
No. 9 on Gasoline, Oil, and Grease Separators (to be appointed)
No. 11 on Soldered Fittings for Tubing, A. M. HOUSER
No. 12 on Minimum Air Gaps in Plumbing Systems, W. K. MCAFEE

Joint Committee on Threaded Cast Iron Pipe, F. H. MOREHEAD

ELECTRIC MOTOR FRAME DIMENSIONS (C28)

** Joint Sponsorship with the National Electrical Manufacturers Association. Sectional Committee organized November, 1927*

A.S.M.E. Members (Total personnel, 28)

C. A. ADAMS
S. A. EINSTEIN
E. W ELY
F. S. ENGLISH
W. F. JONES
A. G. TRUMBULL †

ROLLED THREADS FOR SCREW SHELLS OF ELECTRIC SOCKETS AND LAMP BASES (C44)

** Joint sponsorship with the National Electrical Manufacturers Association. Sectional Committee organized March, 1929*

A.S.M.E. Members (Total personnel, 16)

E. J. BRYANT †
EARLE BUCKINGHAM †
A. B. MORGAN
E. S. SANDERSON †

LETTER SYMBOLS AND ABBREVIATIONS FOR SCIENCE AND ENGINEERING (Z10)

** Joint sponsorship with the American Association for the Advancement of Science, American Institute of Electrical Engineers, American Society of Civil Engineers, and the Society for the Promotion of Engineering Education. Sectional Committee organized January, 1926. Reorganized October, 1935*

A.S.M.E. Members (Total personnel, 39)

S. A. MOSS, *Vice-Chairman* †
K. H. CONDIT
R. J. S. PIGOTT †
(S. R. BEITLER, *Alternate*) †
FRANK THORNTON, JR.

SUBCOMMITTEE CHAIRMEN

Executive Committee, S. A. MOSS, *Vice-Chairman*
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No. 2 on Symbols for Hydraulics, J. C. STEVENS
No. 3 on Symbols for Mechanics, R. E. PETERSON
No. 4 on Symbols for Structural Analysis, ALBERT HAERTLEIN
No. 5 on Symbols for Heat and Thermodynamics, S. A. MOSS
No. 6 on Symbols for Photometry, E. C. CRITTENDEN
No. 7 on Aeronautical Symbols, G. W. LEWIS
No. 8 on Symbols for Electric and Magnetic Quantities, J. F. MEYER
No. 9 on Symbols for Radio, H. M. TURNER
No. 10 on Symbols for Physics, H. K. HUGHES
No. 11 on Abbreviations for Engineering and Scientific Terms, G. A. STETSON
Steering Committee, J. F. MEYER

DRAWINGS AND DRAFTING ROOM PRACTICE (Z14)

** Joint sponsorship with the Society for the Promotion of Engineering Education. Sectional Committee organized July, 1926*

A.S.M.E. Members (Total personnel, 52)

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H. P. FREAR
A. C. HARPER
E. R. HILL
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ALFRED IDDLES
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RUDOLPH MICHEL
F. W. MING
W. C. MUELLER
E. B. NEIL
J. W. OWENS
F. C. PANUSKA
E. S. SMITH †

GRAPHIC PRESENTATION (Z15)

** Sole sponsorship. Sectional Committee organized November, 1926*

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C. M. BIGELOW
WALLACE CLARK
T. E. FRENCH
D. B. PORTER †

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No. 1 on Plan and Scope (to be appointed)
No. 2 on Terminology (to be appointed)
No. 3 on Preferred Practice for Time Series Charts, A. H. RICHARDSON
No. 4 on Engineering and Scientific Graphs, W. A. SHEWHART

SPEEDS OF MACHINERY (Z18)

** Sole sponsorship. Sectional Committee organized May, 1928*

A.S.M.E. Members (Total personnel, 30)

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J. F. DAGGETT
R. C. DEALE †
PAUL DISERENS
F. S. ENGLISH
D. C. JACKSON
JOHN REID
P. G. RHOADS
F. C. SPENCER
O. B. ZIMMERMAN

SUBCOMMITTEE CHAIRMEN

No. 1 on Plan and Scope, A. E. HALL
No. 2 on Questionnaire and Canvass to Industry, F. S. ENGLISH
No. 3—Special Reviewing Committee (to be appointed)

GRAPHICAL SYMBOLS AND ABBREVIATIONS FOR USE IN DRAWINGS (Z32)

** Joint sponsorship with American Institute of Electrical Engineers. Sectional Committee organized April, 1936*

A.S.M.E. Members (Total personnel, 52)

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J. M. BARNES
T. E. FRENCH †
G. F. HABACH
D. T. HAMILTON
A. M. HOUSER
(J. J. HARMAN, *Alternate*)
W. C. MUELLER
L. L. MUNIER
J. W. OWENS
F. C. PANUSKA †
T. R. THOMAS

SUBCOMMITTEE CHAIRMEN

No. 1 on Symbols for Use in Mechanical Engineering, T. E. FRENCH
No. 2 on Symbols for Use in Electrical Engineering, H. W. SAMSON

DEVELOPMENT OF STATISTICAL APPLICATIONS IN ENGINEERING AND MANUFACTURING

Joint Sponsorship with the American Mathematical Society, American Society for Testing Materials, American Statistical Association, Institute of Mathematical Statistics. Appointed in December, 1929

A.S.M.E. Members (Total personnel, 9)

A. G. ASHCROFT
W. H. FULWEILER
L. K. SILLCOX †
J. S. TAWRESEY †

A. S. M. E. Representatives on Miscellaneous Standardization Committees

See also A.S.M.E. Representatives on Other Activities, page RI-9

ACOUSTICAL MEASUREMENTS AND TERMINOLOGY

** Sponsor body: Acoustical Society of America*

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W. B. WHITE
(R. V. PARSONS, *Alternate*)
(J. S. PARKINSON, *Alternate*)

AERONAUTICS

** Sponsor body: Society of Automotive Engineers*

E. A. SPERRY, JR.

APPROVAL AND INSTALLATION REQUIREMENTS FOR GAS BURNING APPLIANCES

** Sponsor body: American Gas Association*

O. F. CAMPBELL

BUILDING CODE REQUIREMENTS FOR LIGHT AND VENTILATION

** Sponsor bodies: Federal Housing Administration and U.S. Public Health Service*

F. R. SCHERER

COAL AND COKE

Committee of American Society for Testing Materials

R. M. Hardgrove

DEFINITIONS OF ELECTRICAL TERMS

* *Sponsor body: American Institute of Electrical Engineers*

C. H. Berry

DRAINAGE OF COAL MINES

* *Sponsor body: American Mining Congress*

O. M. Pruitt

ELECTRIC WELDING APPARATUS

* *Sponsor bodies: American Institute of Electrical Engineers and the National Electrical Manufacturers Association*

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FOREST FIRE PROTECTION

Committee of National Fire Protection Association

C. B. White

GEAR LUBRICANTS

Committee of American Gear Manufacturers Association

G. B. Karelitz

LOADING PLATFORMS AT FREIGHT TERMINALS AND WAREHOUSES

* *Sponsor body: American Trucking Association*

M. C. Maxwell

MANHOLE FRAMES AND COVERS

* *Sponsor bodies: A.S.A. Telephone Group and American Society of Civil Engineers*

Anton Hansen
Homer Rupard

MECHANICAL STANDARDS COMMITTEE

American Standards Association Committee

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(A. L. Baker, *Alternate*)
F. O. Hoagland
F. H. Morehead
(A. M. Houser, *Alternate*)
H. H. Morgan
Edwin B. Ricketts
Frank O. Hoagland
(J. C. Fitts, *Alternate*)
(H. L. Whittemore, *Alternate*)
Executive Committee, Alfred Iddles

METHODS OF TESTING WOOD

* *Sponsor bodies: U.S. Forest Service and the American Society for Testing Materials*

C. M. Bigelow

MISCELLANEOUS OUTSIDE COAL-HANDLING EQUIPMENT

* *Sponsor body: American Mining Congress*

(To be appointed)

PETROLEUM PRODUCTS AND LUBRICANTS

* *Sponsor body: American Society for Testing Materials*

R. G. N. Evans
G. B. Karelitz
(H. J. Masson, *Alternate*)
(S. J. Needs, *Alternate*)

PREFERRED NUMBERS

* *Special Committee of A.S.A.*

K. H. Condit

RATING OF RIVERS

* *Sponsor body: U.S. Geological Survey*

D. W. Mead

ROTATING ELECTRICAL MACHINERY

* *Sponsor bodies: American Institute of Electrical Engineers and National Electrical Manufacturers Association*

Constantine Rick
(C. A. Booth, *Alternate*)

SPECIFICATIONS FOR CAST IRON PIPE AND SPECIAL CASTINGS

* *Sponsor bodies: American Gas Association, American Society for Testing Materials, American Water Works Association, and the New England Water Works Association*

J. E. Gibson
L. R. Howson

SPECIFICATIONS FOR CLEAN BITUMINOUS COAL

* *Sponsor body: American Institute of Mining and Metallurgical Engineers*

R. A. Sherman
(E. L. Lindseth, *Alternate*)

SPECIFICATIONS FOR FIRE TESTS OF BUILDING CONSTRUCTION AND MATERIALS

* *Sponsor bodies: A.S.A. Fire Protection Group, National Bureau of Standards, and the American Society for Testing Materials*

R. C. Parlett

SPECIFICATIONS FOR SIEVES FOR TESTING PURPOSES

* *Sponsor bodies: American Society for Testing Materials and National Bureau of Standards*

R. M. Hardgrove

THERMAL INSULATING MATERIALS

Committee of American Society for Testing Materials

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U.S. INTERDEPARTMENTAL COMMITTEE ON SCREW THREADS

Earle Buckingham
A. M. Houser

VOLUME WATER HEATING

Committee of American Gas Association

Mark Resek

WIRE ROPE FOR MINES

* *Sponsor body: American Mining Congress*

J. L. Harrington

POWER TEST CODES COMMITTEES

Article B6A, Par. 27: The Standing Committee on Power Test Codes shall, under the direction of the Council, have supervision of all the activities of the Society in connection with the A.S.M.E. Power Test Codes, including the interpretation of such codes.

The first Standing Committee on Power Test Codes was organized in December, 1918, to revise and extend the Power Test Codes which had been formulated by various technical committees appointed to develop particular codes. This work began in 1884.

STANDING COMMITTEE

Francis Hodgkinson, *Chairman* (1944)
A. G. Christie, *Vice-Chairman* (1941)
W. W. Lawrence, *Junior Observer* (1941)
H. H. Michelson, *Junior Observer* (1942)

Term expires 1941

A. G. Christie
Paul Diserens
Geo. A. Orrok
L. A. Quayle
W. M. White

Term expires 1942

W. A. Carter
Harte Cooke
E. R. Fish
H. B. Oatley
W. J. Wohlenberg

Term expires 1943

Louis Elliott
G. A. Horne
H. B. Reynolds
P. W. Swain
E. N. Trump

Term expires 1944

C. H. Berry
Francis Hodgkinson
D. S. Jacobus
L. F. Moody
E. B. Ricketts

Term expires 1945

Theodore Baumeister, Jr.
P. H. Hardie
B. V. E. Nordberg
R. J. S. Pigott
M. C. Stuart

---

(1) GENERAL INSTRUCTIONS

Appointed December, 1918
Reorganized, 1939

Theodore Baumeister, Jr., *Chairman*
Paul Diserens
Henry Kreisinger
A. R. Mumford
R. H. Snyder
C. R. Soderberg
M. C. Stuart
P. W. Swain

(2) DEFINITIONS AND VALUES

Appointed December, 1918
Reorganized, 1936

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W. F. Davidson
A. L. Kimball
L. S. Marks
F. G. Philo
J. C. Smallwood
P. W. Swain
A. C. Wood

(3) FUELS

Appointed December, 1918

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E. G. Bailey
B. L. Boye
H. W. Brooks
S. B. Flagg
D. M. Myers
F. G. Philo
G. S. Pope
E. B. Ricketts
F. M. Rogers
E. X. Schmidt
Nicholas Stahl
E. N. Trump

(4) STATIONARY STEAM-GENERATING UNITS

Appointed December, 1918

E. R. Fish, *Chairman*
A. D. Bailey
M. W. Benjamin
B. J. Cross
Martin Frisch
P. H. Hardie
R. M. Hardgrove
Alfred Iddles
E. L. Lindseth
E. L. McDonald
E. B. Powell
R. Shellenberger
R. L. Spencer

(5) RECIPROCATING STEAM ENGINES

Appointed December, 1918
Reorganized, 1931

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Harte Cooke
K. S. M. Davidson
Henrik Greger
J. A. Hunter
H. G. Mueller
B. V. E. Nordberg
A. V. Saharoff
A. G. Witting

(6) STEAM TURBINES

Appointed December, 1918

C. H. Berry, *Chairman*
I. E. Moultrop, *Secretary*
O. D. H. Bentley
W. E. Caldwell
C. B. Campbell
A. G. Christie
H. P. Dahlstrand
V. M. Frost
A. E. Grunert
Francis Hodgkinson
S. A. Moss
R. O. Muller
T. E. Purcell
G. B. Warren

(7) RECIPROCATING STEAM-DRIVEN DISPLACEMENT PUMPS

Appointed December, 1918

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J. N. Chester
J. E. Gibson
G. L. Kollberg
M. B. MacNeille
D. W. Mead
L. A. Quayle

(8) CENTRIFUGAL AND ROTARY PUMPS

Appointed December, 1918
Reorganized, 1936

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R. L. Daugherty
R. G. Folsom
R. C. Glazebrook
W. B. Gregory
R. T. Knapp
J. B. Lincoln
L. F. Moody
Arvid Peterson
F. H. Rogers
W. C. Rudd
Max Spillman
F. G. Switzer
W. M. White
I. A. Winter

(9) DISPLACEMENT COMPRESSORS AND BLOWERS

Appointed December, 1918
Reorganized 1935

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C. R. Houghton
J. F. Huvane
R. M. Johnson
J. F. D. Smith

(10) CENTRIFUGAL AND TURBO-COMPRESSORS AND BLOWERS

Appointed December, 1918
Reorganized, 1929

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Theodore Baumeister, Jr.
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W. H. Carrier
Thomas Chester
L. E. Day
Z. G. Deutsch
S. H. Downs
P. E. Good
J. J. Grob
H. F. Hagen
Paul Hoffman
H. D. Kelsey
A. L. Kimball
R. D. Madison
L. S. Marks
Arvid Peterson
H. F. Schmidt
M. C. Stuart

(12) CONDENSERS, WATER HEATING, AND COOLING EQUIPMENT

Appointed December, 1918

GEO. A. ORROK, *Chairman*
P. H. HARDIE, *Secretary*
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R. N. EHRHART
J. F. GRACE
D. W. R. MORGAN
H. B. REYNOLDS
P. E. REYNOLDS

(13) REFRIGERATING SYSTEMS

Appointed December, 1918
Reorganized May, 1939

B. H. JENNINGS, *Chairman* †
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(R. W. WATERFILL, *Alternate*)
J. C. CONSLEY
(H. B. POWNALL, *Alternate*)
R. J. EWER †
WALTER JONES †
A. W. OAKLEY
C. L. SVENSON
FRANK ZUMBRO †

(14) EVAPORATING APPARATUS

Appointed December, 1918

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E. A. NEWHALL
H. L. PARR
L. C. ROGERS

(15) STEAM LOCOMOTIVES

Appointed December, 1918

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H. B. OATLEY
G. E. RHOADS
L. K. SILLCOX
W. E. WOODARD

(16) GAS PRODUCERS

Appointed December, 1918

C. D. SMITH

(17) INTERNAL-COMBUSTION ENGINES

Appointed December, 1918
Reorganized, 1939

LEE SCHNEITTER, *Chairman*
F. H. DUTCHER, *Secretary*
J. C. BARNABY
G. C. BOYER
HARTE COOKE
H. E. DEGLER
W. L. H. DOYLE
L. B. JACKSON
E. J. KATES
E. C. MAGDEBURGER
B. V. E. NORDBERG
RUSSELL PYLES
M. J. REED
O. D. TREIBER

† Official A.S.M.E. representatives serving on this committee.

(18) HYDRAULIC PRIME MOVERS

Appointed December, 1918
Reorganized, 1931

S. L. KERR, *Chairman*
C. M. ALLEN
L. M. DAVIS
H. L. DOOLITTLE
W. F. DURAND
N. R. GIBSON
J. P. GROWDON
T. H. HOGG
L. J. HOOPER
C. W. HUBBARD
E. C. HUTCHINSON
D. J. MCCORMACK
L. F. MOODY
W. J. RHEINGANS
E. B. STROWGER
R. V. TERRY
W. M. WHITE

(19) INSTRUMENTS AND APPARATUS

Appointed December, 1918

W. A. CARTER, *Chairman*
C. M. ALLEN
W. C. ANDRAE
E. G. BAILEY
H. S. BEAN
L. J. BRIGGS
J. D. DAVIS
K. J. DE JUHASZ
R. E. DILLON
F. M. FARMER
J. B. GRUMBEIN
W. W. JOHNSON
W. H. KENERSON
E. S. LEE
E. L. LINDSETH
OSBORN MONNETT
S. A. MOSS
R. J. S. PIGOTT
E. B. RICKETTS
W. A. SLOAN
R. B. SMITH
I. M. STEIN

(20) SPEED, TEMPERATURE AND PRESSURE RESPONSIVE GOVERNORS

Appointed December, 1921
Reorganized February, 1940

C. R. SODERBERG, *Chairman*
R. J. CAUGHEY
HARTE COOKE
W. L. H. DOYLE
S. L. KERR
A. F. SCHWENDNER
R. B. SMITH

(21) DUST SEPARATING APPARATUS

Appointed October, 1934

M. D. ENGLE, *Chairman*
OLLISON CRAIG, *Secretary*
A. D. BAILEY
H. H. BUBAR
W. G. CHRISTY
H. O. CROFT
J. M. DALLAVALLE
H. O. DANZ
H. C. DOHRMANN
PHILIP DRINKER
J. W. FEHNEL
H. F. HAGEN
P. H. HARDIE
C. W. HEDBERG
J. H. LEECH
H. E. MACOMBER
H. B. MELLER
H. C. MURPHY
B. F. TILLSON

A.S.M.E. Representatives on Other Technical Committees

See also A.S.M.E. Representatives on Other Activities, page RI-9

DEVELOPMENT OF DEFINITIONS FOR THE NET CALORIFIC VALUE AND GROSS CALORIFIC VALUE OF FUELS

Sponsor body: American Society for Testing Materials

W. J. WOHLENBERG

COMMITTEE ON REDEFINING SO-CALLED STANDARD TON OF REFRIGERATION

Sponsor body: American Society of Refrigerating Engineers

G. B. BRIGHT

COMMITTEE ON GASEOUS FUELS

Sponsor body: American Society for Testing Materials

E. X. SCHMIDT

COAL TESTING CODE COMMITTEE

Joint sponsorship with the American Institute of Mining and Metallurgical Engineers

A. R. MUMFORD

SAFETY COMMITTEES

ARTICLE B6A, PAR. 25: The Standing Committee on Safety shall advise the Council on the activities of the Society having to do with engineering and industrial safety, except the activities of the Boiler Code Committee, for which special provision is made.

The first Standing Committee on Safety was appointed in October, 1921.

STANDING COMMITTEE

T. F. HATCH, *Chairman* (1941)
A. W. LUCE (1942)
A. E. WINDLE (1943)
H. C. HOUGHTON (1944)
E. R. GRANNISS (1945)

SAFETY CODE FOR ELEVATORS (A17)

** Joint Sponsorship with the American Institute of Architects and the National Bureau of Standards. Sectional Committee organized November, 1922*

Reorganized July, 1940

A.S.M.E. Members (Total personnel, 45)

O. P. CUMMINGS, *Vice-Chairman*
C. R. CALLAWAY
D. L. HOLBROOK †
D. L. LINDQUIST
N. O. LINDSTROM †
M. B. MCLAUTHLIN
W. S. PAINE

SUBCOMMITTEE CHAIRMEN

Emergency Elevator Rules, D. J. PURINTON
Executive Committee, D. J. PURINTON
Existing Elevators, D. J. PURINTON
Inspectors' Manual, K. A. COLAHAN
Mechanical Safety Equipment, D. L. LINDQUIST
Ways and Means, J. J. MATSON
Wire Rope, D. J. PURINTON
Working (to be appointed)

SAFETY CODE FOR MECHANICAL POWER-TRANSMISSION APPARATUS (B15)

** Joint sponsorship with the International Association of Industrial Accident Boards and Commissions and the National Conservation Bureau. Sectional Committee organized February, 1921*

A.S.M.E. Members (Total personnel, 26)

G. M. NAYLOR, *Chairman* †
P. G. RHOADS, *Secretary*
D. C. WRIGHT †
(G. N. VAN DERHOEF, *Alternate*) †

SUBCOMMITTEE CHAIRMEN

No. 1 on Detail Classification of Belts (to be appointed)
No. 2 on Modification of Rule 223 for Cone Pulley Belts (to be appointed)
No. 3 on Mechanical Power Control, W. S. PAINE
No. 4 on Use of ASA Code Versus State Codes (to be appointed)
No. 5 on Statistics on Place of Occurrence of Accidents (to be appointed)
No. 6 on V-Belt Drives, D. C. WRIGHT

* Note: All of the safety committees for which the Society is sponsor or joint sponsor, or on which it has representation, are organized under the procedure of the American Standards Association.

† Official A.S.M.E. representative serving on this committee.

SAFETY CODE ON COMPRESSED AIR MACHINERY AND EQUIPMENT (B19)

** Joint sponsorship with the American Society of Safety Engineers—Engineering Section, National Safety Council. Sectional Committee organized May, 1923*

A.S.M.E. Members (Total personnel, 24)

D. L. ROYER, *Chairman*
H. D. EDWARDS
W. J. GRAVES

SAFETY CODE FOR CONVEYORS AND CONVEYING MACHINERY (B20)

** Joint Sponsorship with the National Conservation Bureau. Sectional Committee organized November, 1925, Reorganized, April 1937*

A.S.M.E. Members (Total personnel, 53)

D. L. ROYER, *Chairman*
C. T. COLLEY
W. J. GRAVES
M. A. KENDALL †
(N. W. ELMER, *Alternate*) †
P. T. ONDERDONK
C. G. PFEIFFER
R. B. RENNER
F. J. SHEPARD, JR.
J. G. WHEATLEY

SUBCOMMITTEE CHAIRMEN

No. 1 on All Types of Chain Conveyors, Belt Conveyors, Belt Elevators Including Steel Belt, and Screw, Track or Scraper Conveyors, C. G. PFEIFFER
No. 2 on Gravity Conveyors and Chutes, Live Roll Conveyors, H. G. DALTON
No. 3 on Cable-Operated and Cable Flight Conveyors and Cableways, R. McA. KEOWN
No. 4 on Air, Steam, or Liquid Conveyors, J. J. MCNULTA
No. 5 on Tiering, Piling, and Stacking Conveyors, J. G. WHEATLEY

SAFETY CODE FOR CRANES, DERRICKS, AND HOISTS (B30)

** Joint sponsorship with U.S. Navy Department, Bureau of Yards and Docks. Sectional Committee organized November, 1926*

A.S.M.E. Members (Total personnel, 57)

LEWIS PRICE †
F. H. SCHWERIN
R. H. WHITE †
H. L. WHITTEMORE

SUBCOMMITTEE CHAIRMEN

Executive Committee, J. C. WHEAT
No. 1 on Overhead and Gantry Cranes, R. H. WHITE
No. 2 on Locomotive and Tractor Cranes, H. H. VERNON
No. 3 on Derricks and Hoists, LEWIS PRICE
No. 4 on Miscellaneous Equipment for Cranes and Hoists, L. W. HOPKINS
No. 5 on Jacks, E. W. CARUTHERS
Editing Committee, M. G. FLOYD

A.S.M.E. Representatives on Other Safety Committees

See also A.S.M.E. Representatives on Other Activities, page RI-9

SAFETY CODE FOR ABRASIVE WHEELS

** Sponsor bodies: Grinding Wheel Manufacturers Association of United States and Canada, and International Association of Industrial Accident Boards and Commissions*

J. B. CHALMERS

SAFETY CODE FOR CONSTRUCTION WORK

** Sponsor bodies: American Institute of Architects and National Safety Council*

C. H. O'NEIL

COOPERATION WITH OTHER ENGINEERING SOCIETIES

Committee of American Society of Safety Engineers—Engineering Section, National Safety Council

H. L. MINER

ASA SAFETY CODE CORRELATING COMMITTEE

A. W. LUCE
(A. E. WINDLE, *Alternate*)

SAFETY CODE FOR EXHAUST SYSTEMS

** Sponsor body: International Association of Industrial Accident Boards and Commissions*

T. F. HATCH

SAFETY CODE FOR FLOOR AND WALL OPENINGS, RAILINGS, AND TOE BOARDS

** Sponsor body: National Safety Council*

A. E. WINDLE

SAFETY CODE FOR FORGING AND HOT METAL STAMPING

** Sponsor bodies: American Drop Forging Institute and National Safety Council*

C. F. PARK

SAFETY CODE ON COLORS FOR IDENTIFICATION OF GAS MASK CANISTERS

** Sponsor body: National Safety Council*

L. C. LICHTY

SAFETY CODE FOR LADDERS

** Sponsor body: American Society of Safety Engineers—Engineering Section, National Safety Council*

H. C. HOUGHTON

SAFETY CODE FOR LAUNDRY MACHINERY AND OPERATION

** Sponsor bodies: American Institute of Laundering, International Association of Governmental Labor Officials, and National Association of Mutual Casualty Companies*

E. J. Carroll

SAFETY CODE FOR LIGHTING FACTORIES, MILLS, AND OTHER WORK PLACES

** Sponsor body: Illuminating Engineering Society*

A. W. Luce

LOW VOLTAGE ELECTRICAL HAZARDS

Special Committee of the American Society of Safety Engineers—Engineering Section, National Safety Council

J. P. Jackson

SAFETY CODE FOR MECHANICAL REFRIGERATION

** Sponsor body: American Society of Refrigerating Engineers*

O. A. Anderson
Crosby Field
E. W. Gallenkamp
W. F. Jones
(A. W. Oakley, *Alternate to all A.S.M.E. Representatives*)

SAFETY CODE FOR PAPER AND PULP MILLS

** Sponsor body: National Safety Council*

R. L. Weldon

SAFETY CODE FOR POWER PRESSES, AND FOOT AND HAND PRESSES

** Sponsor body: National Safety Council*

J. B. Chalmers

SAFETY CODE FOR PREVENTION OF DUST EXPLOSIONS

** Sponsor bodies: National Fire Protection Association and U.S. Department of Agriculture*

R. M. Ferry

SAFETY CODE FOR PROTECTION OF HEADS, EYES, AND RESPIRATORY ORGANS OF INDUSTRIAL WORKERS

** Sponsor body: National Bureau of Standards*

T. A. Walsh, Jr.
(T. F. Hatch, *Alternate*)

SAFETY CODE FOR PROTECTION OF INDUSTRIAL WORKERS IN FOUNDRIES

** Sponsor bodies: American Foundrymen's Association and National Founders Association*

H. M. Lane

SAFETY CODE FOR RUBBER MACHINERY

** Sponsor bodies: National Safety Council and International Association of Industrial Accident Boards and Commissions*

E. S. Ault

SAFETY IN QUARRY OPERATIONS

** Sponsor body: National Safety Council*

Redfield Proctor
(H. A. Collin, *Alternate*)

SPECIFICATIONS AND METHOD OF TEST FOR SAFETY GLASS

** Sponsor bodies: National Conservation Bureau and National Bureau of Standards*

T. A. Walsh, Jr.

SAFETY CODE FOR TEXTILES

** Sponsor body: National Safety Council*

M. A. Golrick, Jr.

SAFETY CODE FOR VENTILATION

** Sponsor body: American Society of Heating and Ventilating Engineers*

T. F. Hatch

SAFETY CODE FOR WALKWAY SURFACES

** Sponsor bodies: American Institute of Architects and American Society of Safety Engineers—Engineering Section, National Safety Council*

G. K. Palsgrove

SAFETY CODE FOR WORK IN COMPRESSED AIR

** Sponsor body: International Association of Industrial Accident Boards and Commissions*

L. J. Eibsen

BOILER CODE COMMITTEES

Article B6A, Par. 26: The Special Committee on Boiler Code shall, under the direction of the Council, have supervision of all the activities of the Society in connection with the A.S.M.E. Codes for Pressure Vessels, including the interpretations of these codes.

The first Special Committee on Boiler Code was organized in September, 1911.

SPECIAL COMMITTEE

D. S. Jacobus, *Chairman*
E. R. Fish, *Vice-Chairman*
C. W. Obert, *Honorary Secretary*
M. Jurist, *Acting Secretary*
C. A. Adams
H. E. Aldrich
H. C. Boardman
Perry Cassidy
R. E. Cecil
F. S. Clark
A. J. Ely
V. M. Frost
C. E. Gorton
A. M. Greene, Jr.
W. G. Humpton
J. O. Leech
I. E. Moultrop
C. O. Myers
H. B. Oatley
James Partington
Walter Samans
S. K. Varnes
A. C. Weigel

Honorary Members

W. H. Boehm
W. F. Durand
T. E. Durban
C. L. Huston
W. F. Kiesel, Jr.
M. F. Moore
H. H. Vaughan
H. LeRoy Whitney

CONFERENCE COMMITTEE

W. E. Allen, St. Louis, Mo.
T. R. Archer, Delaware
L. M. Barringer, Seattle, Wash.
J. G. Bollock, St. Joseph, Mo.
B. M. Book, Pennsylvania
H. S. Brunson, Minnesota
E. S. Carpenter, Rhode Island
L. M. Cave, Maryland
S. Cherrington, Ohio
City Boiler Inspector, Parkersburg, W. Va.
D. J. Cody, Kansas City, Mo.
A. L. Colby, Louisiana
A. J. Conway, Indiana
M. A. Edgar, Wisconsin
C. W. Foster, Omaha, Neb.
M. R. Francis, West Virginia
W. H. Furman, New York
F. D. Garvin, Houston, Tex.
Gerald Gearon, Chicago, Ill.
C. H. Gram, Oregon
B. Gretzke, Washington
F. A. Heckinger, Memphis, Tenn.
H. K. Kugel, District of Columbia
Joe Kunschik, Texas
P. N. Lehoczky, Ohio
G. A. Luck, Massachusetts
C. E. McGinnis, Los Angeles, Calif.
H. H. Mills, Detroit, Mich.
J. D. Newcomb, Jr., Arkansas
W. L. Newton, Oklahoma
F. A. Page, California
L. C. Peal, Nashville, Tenn.
E. K. Sawyer, Maine
A. H. Schleman, Tampa, Fla.
J. F. Scott, New Jersey
J. N. Seiger, Evanston, Ill.
J. A. Strait, Tulsa, Okla.
C. I. Smith, Utah
Wm. E. Smith, Hawaiian Islands
John H. Thorpe, Michigan
C. E. Ward, North Carolina

EXECUTIVE COMMITTEE

D. S. Jacobus, *Chairman*
H. E. Aldrich, *Vice-Chairman*
E. R. Fish
V. M. Frost
C. E. Gorton
C. W. Obert
James Partington

SUBCOMMITTEES

Boilers of Locomotives

James Partington, *Chairman*
F. H. Clark
J. M. Hall
H. B. Oatley

Care of Steam Boilers and Other Pressure Vessels in Service

F. M. Gibson, *Chairman*
D. C. Carmichael
V. M. Frost
J. R. Gill
Frank Henry
J. A. Hunter
H. J. Kerr
P. B. Place
S. T. Powell
C. W. Rice
J. B. Romer
W. C. Schroeder
Nicholas Stahl
F. G. Straub

Ferrous Materials

D. B. Rossheim, *Chairman*
A. B. Bagsar
E. C. Chapman
A. J. Ely
H. J. French
W. R. Grunow
M. B. Higgins
W. G. Humpton
A. Hurtgen
T. McLean Jasper
J. J. Kanter
H. J. Kerr
A. B. Kinzel
L. J. Mason
P. E. McKinney
N. L. Mochel
E. L. Robinson
S. K. Varnes
A. E. White
R. Wilson

Heating Boilers

C. E. Gorton, *Acting Chairman*
C. E. Bronson
J. A. Darts
Wm. Ferguson
L. N. Hunter
W. E. Stark
J. W. Turner

Material Specifications

Perry Cassidy, *Chairman*
A. M. Greene, Jr.
W. G. Humpton
J. O. Leech
P. J. Smith
A. C. Weigel

Miniature Boilers

C. E. Gorton, *Chairman*
W. H. Furman
G. A. Luck
C. W. Obert

Nonferrous Materials

H. B. Oatley, *Chairman*
J. J. Aull
W. F. Burchfield
D. K. Crampton
J. R. Freeman, Jr.
A. M. Houser
E. F. Miller
Joseph Price
R. L. Templin

Power Boilers

H. E. Aldrich, *Chairman*
Perry Cassidy
E. R. Fish
V. M. Frost
D. L. Royer
A. C. Weigel

Rules for Inspection

(This subcommittee is being reorganized)

Special Design

D. B. Wesstrom, *Chairman*
H. C. Boardman
R. E. Cecil
T. W. Greene
D. B. Rossheim
E. O. Waters
F. S. G. Williams

Unfired Pressure Vessels

E. R. Fish, *Chairman*
C. A. Adams
C. E. Bronson
R. E. Cecil
Paul Diserens
H. S. Smith
D. B. Wesstrom

Welding

Members of A.S.M.E. Boiler Code Committee

James Partington, *Chairman*
E. C. Chapman
J. H. Deppeler
W. D. Halsey
J. C. Hodge
R. K. Hopkins
J. T. Phillips
L. A. Sheldon

Members of Conference Committee of American Welding Society

C. W. Obert, *Chairman*
C. A. Adams
H. C. Boardman
Walter Samans
A. C. Weigel

SPECIAL COMMITTEES

Approval of New Materials

C. A. Adams, *Chairman*

Clad Vessels

S. K. Varnes, *Chairman*

Special Committee on Coordination

V. M. Frost, *Chairman*

Extension of Fusion Welding Requirements

H. E. Aldrich, *Chairman*

Feedwater

C. W. Rice, *Chairman*

Issuance of Code Symbol Stamps

C. O. Myers, *Chairman*

Radiographic Examination of Welded Joints

C. A. Adams, *Chairman*

Revision of Section VIII of the A.S.M.E. Boiler Code

E. R. Fish, *Chairman*

Rules for Bolted Flanged Connections

D. B. Wesstrom, *Chairman*

Rules for Dished Heads

H. C. Boardman, *Chairman*

Rules for Openings

T. D. Tifft, *Chairman*

Safety Valve Requirements

H. B. Oatley, *Chairman*

Work of Boiler Code Committee

H. E. Aldrich, *Chairman*

API-ASME COMMITTEE ON UNFIRED PRESSURE VESSELS

Walter Samans, *Chairman*

A.S.M.E. Representatives

R. E. Cecil
E. R. Fish
D. S. Jacobus
T. McLean Jasper
James Partington

A.P.I. Representatives

A. J. Ely
K. V. King
(P. D. McElfish, *Alternate*)
R. C. Powell
Walter Samans
T. D. Tifft

THE WOMAN'S AUXILIARY TO THE A.S.M.E.

The Woman's Auxiliary to the A.S.M.E. was organized on May 10, 1923, and its Constitution and By-Laws was approved by the Council of the A.S.M.E. on October 27, 1924. The objects of the Auxiliary are to render service to all that pertains to the interest of the profession of mechanical engineering; to cooperate with any committees of the A.S.M.E.; and to assist the sons and daughters of the members of the Society or worthy students of mechanical engineering in obtaining scholarships; and to promote any other objects consistent with the aims or objects of the A.S.M.E.

OFFICERS

President, Mrs. F. M. Gibson
First Vice-President, Mrs. E. C. M. Stahl
Second Vice-President, Mrs. A. R. Cullimore
Third Vice-President, Mrs. Crosby Field
Fourth Vice-President, Mrs. J. Page Harbeson
Fifth Vice-President, Mrs. J. H. Herron
Recording Secretary, Mrs. C. H. Young
Corresponding Secretary, Mrs. C. H. Fay
Treasurer, Mrs. A. H. Morgan

STANDING COMMITTEE CHAIRMEN

Education, Mrs. Roy V. Wright
Membership, Mrs. G. E. Hagemann
Publicity, Mrs. A. R. Cullimore
Custodian, Miss Burtie Haar

COUNCIL REPRESENTATIVES

A. G. Christie
J. H. Herron

OFFICERS OF LOCAL SECTIONS

Baltimore

Chairman, Mrs. D. E. Donovan
Vice-Chairman, Mrs. A. G. Christie
Secretary, Mrs. J. H. Berryman
Treasurer, Mrs. L. F. Coffin

Cleveland

Chairman, Mrs. T. F. Githens
Vice-Chairman, Mrs. J. H. Herron
Secretary, Mrs. Ernest Havillon
Treasurer, Mrs. Walter Baggaley

Los Angeles

Chairman, Mrs. S. S. Hansen
Secretary, Mrs. J. C. Scullin
Treasurer, Mrs. Bernard Toben

Metropolitan

Chairman, Mrs. E. C. M. Stahl
First Vice-President, Mrs. J. Noble Landis
Second Vice-President, Mrs. R. B. Purdy
Third Vice-President, Mrs. Webster Tallmadge
Recording Secretary, Mrs. C. H. Young
Corresponding Secretary, Mrs. A. C. Coonradt
Treasurer, Mrs. C. E. Gus

Philadelphia

Chairman, Mrs. J. Page Harbeson, Jr.
Vice-Chairman, Mrs. E. F. Zeiner
Recording Secretary, Mrs. J. J. McCarthy
Corresponding Secretary, Mrs. F. E. Washburn
Treasurer, Mrs. W. F. Glimm

AWARDS

The following paragraphs deal with the medals, awards, scholarships, and loan funds which come within the jurisdiction of the A.S.M.E. Other awards available to Student Members are listed in *Mechanical Engineering,* February, 1938, page 183. The Society also participates with other engineering societies in a number of joint awards. Further details concerning all the awards will be found in a series of articles beginning in the October, 1938, issue of *Mechanical Engineering.*

Honorary Membership, to which persons of acknowledged professional eminence are elected by unanimous vote of Council under the provisions of the By-Laws and Rules. A list of honorary members is given on page RI-42.

Life Membership, which may be conferred by the Council for distinguished service to the Society; or secured by a member by payment for an annuity in accordance with the provisions of the By-Laws.

A.S.M.E. Medal, established by the Society in 1920 to be presented, together with an engraved certificate, for distinguished service in engineering and science. May be awarded for general service in science having possible application in engineering.

Holley Medal, instituted and endowed in 1924 by George I. Rockwood, Past Vice-President of the Society, to be bestowed, together with an engraved certificate, for some great and unique act of genius of engineering nature that has accomplished a great and timely public benefit.

Worcester Reed Warner Medal, provision for which was made in the will of Worcester Reed Warner, Honorary Member of the Society, is a gold medal to be bestowed, together with an engraved certificate, on the author of the most worthy paper received, dealing with progressive ideas in mechanical engineering or efficiency in management.

Melville Medal, established in 1914 by the bequest of Rear-Admiral George W. Melville, Honorary Member and Past-President of the Society, to be presented, together with an engraved certificate, for an original paper or thesis of exceptional merit, presented to the Society for discussion and publication, to encourage excellence in papers. The medal may be presented annually.

Spirit of St. Louis Medal, established by an endowment fund created in 1929 by citizens of St. Louis, Mo., to be awarded for meritorious service in the advancement of aeronautics. This medal will be awarded at the discretion of the Council of the Society at approximately three-year periods upon the recommendation of its Board of Honors and Awards.

Pi Tau Sigma Medal Award, established in 1938, endowed by Pi Tau Sigma, the national honorary mechanical engineering fraternity, to be presented annually, together with an engraved certificate, to the young mechanical engineer for outstanding achievement in his profession within the ten years after graduation from a regular four-year mechanical engineering course of a recognized American college or university. Any mechanical engineering graduate, not more than thirty-five years of age, whose achievement has been all or in part in any field including industrial, educational, political, research, civic, etc., is eligible.

Junior Award, annual cash award of $50, established in 1914, from a fund created by Henry Hess, Past Vice-President of the Society, to be presented, together with an engraved certificate, for the best paper or thesis submitted by a Junior Member.

Charles T. Main Award, annual cash award of $150, established in 1919 from a fund created by Charles T. Main, Past-President of the Society, to be awarded, together with an engraved certificate, to a Student Member of the Society, for the best paper within the general subject of the influence of the profession upon public life. The exact subject is assigned by the Board of Honors and Awards, subject to the approval of the Council, and is announced each year through the Honorary Chairman of the Student Branches.

Student Awards, two annual cash awards of $25 each, established in 1914, from a fund created by Henry Hess, Past Vice-President of the Society, to be presented, together with engraved certificates, for the best papers or theses submitted by Student Members. The awards for 1932 and subsequent years have been given, one for undergraduate and the other for postgraduate work.

SCHOLARSHIPS AND LOAN FUNDS

Max Toltz: Loan Fund of $15,000 established by Major Max Toltz, former member of the Council of the Society, the income to be used for assistance to Student Members.

John R. Freeman: Fund of $25,000 established in 1926 by John R. Freeman, Past-President of the Society, the income to be used for travel scholarships and research.

Woman's Auxiliary: Scholarship or Fellowship offered by the Woman's Auxiliary to the Society to assist sons and daughters of members or worthy students of mechanical engineering.

RECIPIENTS OF AWARDS

The names of the recipients of the different awards to date are given in the following lists, together with the dates of presentation, and the services or papers for which the awards were made. There were no awards for the years not listed.

A.S.M.E. Medal

1921 Hjalmar Gotfried Carlson, in recognition of the services rendered the Government because of his invention and part in the production of 20,000,000 Mark III drawn steel booster casings used principally as a component of 75-mm high explosive shells, but also used extensively in gas shells and bombs

1922 Frederick Arthur Halsey, for his paper describing the premium system of wage payments presented before the Society at the Providence Meeting in 1891, as the adoption of the methods there proposed has had a profound effect toward harmonizing the relations of worker and employer

1923 John Ripley Freeman, for his eminent service in engineering and manufacturing by his meritorious work in fire prevention and the preservation of property

1926 R. A. Millikan, in recognition of his contributions to science and engineering

1927 Wilfred Lewis, for his contributions to the design and construction of gear teeth

1928 Julian Kennedy, for his services and contributions to the iron and steel industry

1929 William LeRoy Emmet, for his contributions in the development of the steam turbine, electric propulsion of ships, and other power-generating apparatus

1931 Albert Kingsbury, for his research and development work in the field of lubrication

1933 Ambrose Swasey, for his contributions to the advancement of the engineering profession and for his part in the development of the turret lathe and the astronomical telescope

1934 Willis H. Carrier, in recognition of his research and development work in air-conditioning

1935 Charles T. Main, for distinguished achievements in the textile and other industries, in engineering education, and for eminent service to the engineering profession

1936 Edward Bausch, for meritorious mechanical developments in the field of optics

1937 Edward P. Bullard, for outstanding leadership in the development of station-type machine tools

1938 Stephen J. Pigott, for outstanding leadership in marine propulsion and construction

1939 James E. Gleason, for service to the cause of safer and better transportation

1940 Charles F. Kettering, for outstanding inventions and research.

Holley Medal

1924 Hjalmar Gotfried Carlson, for his inventions and processes which made possible the timely production of drawn steel booster casings for artillery ammunition, thereby aiding victory in the World War (diploma in recognition of achievements presented in 1921)

1927 Elmer Ambrose Sperry, for achievements and inventions that have advanced the naval arts, including the gyroscope that has freed navigation from the dangers of the fluctuating magnetic compass

1929 Baron Chuzaburo Shiba, for his contributions to knowledge through fundamental research, including the field of aerodynamics, by the development of ultra-rapid kinematographic methods.

1934 Irving Langmuir, for his contributions to science and engineering, including the development of gas-filled incandescent lamps, thoriated filament for thermionic emission, atomic hydrogen welding, phase control operation of the thyratron tube, and fundamental research in oil films
1936 Henry Ford, for revolutionary influence through invention and practice on transportation and on mass production methods in manufacturing
1937 Frederick G. Cottrell, for preeminent public service—the invention of electric precipitation—advancement of the science of gas liquefaction—gifts for engineering research
1938 Francis Hodgkinson, for meritorious services in the development of the steam turbine
1939 Carl E. Johansson, in recognition of his pioneer work in the development of basic measuring gages
1940 Edwin Howard Armstrong, for his leadership in the field of radio communication.

Worcester Reed Warner Medal

1933 Dexter S. Kimball, for his contributions to efficiency in management as exemplified by his recently revised "Principles of Industrial Organization" and by his many articles, engineering society papers, and public addresses
1934 Ralph E. Flanders, for his contributions to a better understanding of the relationship of the engineer to economic problems and social trends as exemplified by the many papers which he has presented
1935 Stephen Timoshenko, for his contributions to the theory of the design of elastic structures and the treatment of dynamics of moving machinery
1936 Charles M. Allen, for his early and continued hydraulic laboratory work and for the permanent value of the papers on his development of methods of testing large hydraulic turbine installations
1937 Clarence F. Hirshfeld, for his research and contributions to the theory and practice of heat-power engineering as exemplified by books and papers
1938 Lawford H. Fry, for contributions relating to improved locomotive boiler design and utilization of better materials in railway equipment
1939 Rupen Eksergian, for influential papers of permanent value in A.S.M.E. Transactions
1940 William Benjamin Gregory, for distinguished work in hydraulic engineering, which has been the basis for many engineering papers.

Melville Medal

1927 Leon P. Alford, "Laws of Manufacturing Management"
1929 Joseph W. Roe, "Principles of Jig and Fixture Practice"
1930 Herman Diederichs and William D. Pomeroy, "The Occurrence and Elimination of Surge or Oscillating Pressure in Discharge Lines From Reciprocating Pumps"
1931 Arthur E. Grunert, "Comparative Performance of a Pulverized-Coal-Fired Boiler Using Bin System and Unit System of Firing"
1932 Alexey J. Stepanoff, "Leakage Loss and Axial Thrust in Centrifugal Pumps"
1933 William E. Caldwell, "Characteristics of Large Hell Gate Direct-Fired Boiler Units"
1935 Oscar R. Wikander, "Draft-Gear Action in Long Trains"
1936 H. A. Stevens Howarth, "The Loading and Friction of Thrust and Journal Bearings With Perfect Lubrication"
1937 Alfred J. Büchi, "Supercharging of Internal-Combustion Engines With Blowers Driven by Exhaust-Gas Turbines"
1938 Alphonse I. Lipetz, "Air Resistance of Railroad Equipment"
1939 Lester M. Goldsmith, for his paper, "High-Pressure High-Temperature Turbine-Electric Steamship *J. W. Van Dyke*"
1940 Carl A. W. Brandt, for his paper, "The Locomotive Boiler."

Spirit of Saint Louis Medal

1929 Daniel Guggenheim, founder of the Guggenheim Fund for the Promotion of Aeronautics
1932 Paul Litchfield, for his work in encouraging and sponsoring airship design and construction in this country
1935 Will Rogers, for his splendid, constructive, and unselfish work in the achievement of aviation, and the building up of public confidence in aviation through his articles in the press, over the radio, and from the speaker's platform
1938 James H. Doolittle, for meritorious service in the advancement of aeronautics.

Pi Tau Sigma Medal

1938 Wilfrid E. Johnson, for his development work in the field of refrigeration
1939 John I. Yellott, Jr., in recognition of significant achievements in steam-flow research and engineering education; also contributions on "Supersaturated Steam" and "Condensation of Flowing Steam in Diverging Nozzles"
1940 George A. Hawkins, for significant achievements in high-pressure steam research and engineering education.

Junior Award

1915 Ernest O. Hickstein, "Flow of Air Through Thin Plate Orifices"
1916 L. M. McMillan, "The Heat Insulating Properties of Commercial Steam-Pipe Coverings"
1919 E. D. Whalen, "Properties of Airplane Fabrics"
1921 S. Logan Kerr, "Moody Ejector Turbine"
1922 R. H. Heilman, "Heat Losses From Bare and Covered Wrought-Iron Pipe at Temperatures up to 800 Degrees Fahrenheit"
F. L. Kallam, "Preliminary Report on the Investigation of the Thermal Conductivity of Liquids"
1923 S. S. Sanford and S. Crocker, "The Elasticity of Pipe Bends"
1924 R. H. Heilman, "Heat Losses Through Insulating Material"
1925 Gilbert S. Schaller, "An Investigation of Seattle as a Location for a Synthetic Foundry Industry"
1927 William M. Frame, "Stresses Occurring in the Walls of an Elliptical Tank Subjected to Low Internal Pressure"
1928 M. D. Aisenstein, "A New Method of Separating the Hydraulic Losses in a Centrifugal Pump"
1929 Arthur M. Wahl, "Stresses in Heavy, Closely Coiled Helical Springs"
1930 Ed Sinclair Smith, "Quantity-Rate Fluid Meters"
1931 M. K. Drewry, "Radiant-Superheater Developments"
1932 Edmond M. Wagner, "Frictional Resistance of a Cylinder Rotating in a Viscous Fluid Within a Coaxial Cylinder"
1933 Townsend Tinker, "Surface Condenser Design and Operating Characteristics"
1934 John I. Yellott, Jr., "Supersaturated Steam"
1935 Stanley J. Mikina, "Effect of Skewing and Pole Spacing on Magnetic Noise in Electrical Machinery"
1936 Harwood F. Mullikan, Jr., "Evaluation of Effective Radiant Heating Surface and Application of the Stefan-Boltzman Law to Heat Absorption in Boiler Furnaces"
1937 Leslie J. Hooper, "American Hydraulic-Laboratory Practice"
1938 Arthur C. Stern, "Separation and Emission of Cinders and Fly Ash"
1940 Robert E. Newton, for his paper, "A Photoelastic Study of Stresses in Rotary Disks."

Charles T. Main Award

1925 Clement R. Brown, Catholic University of America. Subject: "The Influence of the Locomotive on the Unity of the United States"
1926 W. C. Saylor, Johns Hopkins University. Subject: "The Effect of the Cotton Gin Upon the History of the United States During Its First Seventy Years"
1927 No Award. Subject: "Scientific Management and Its Effect Upon the Industries"
1928 Robert M. Meyer, Newark College of Engineering. Subject: "Scientific Management and Its Effect Upon Manufacturing"
1929 No Award. Subject: "The Influence of Engineering on Farm Production"
1930 Jules Podnossoff, Polytechnic Institute of Brooklyn. Subject: "The Value of the Safety Movement in the Industries"
1931 Robert E. Klise, University of Michigan. Subject: "Interchangeability—Its Development and Significance in Industry"
1932 Marshall Anderson, University of Michigan. Subject: "Apprenticeship and Vocational Training"
1933 George D. Wilkinson, Jr., Newark College of Engineering Subject: "Progress in the Prevention of Smoke and Atmospheric Pollution"
1934 Philip P. Self, Colorado State College. Subject: "Air Conditioning—Its Practicability and Relation to Public Welfare"

1935 G. LOWELL WILLIAMS, Lafayette College. Subject: "Co-ordinated Transportation—An Economic Comparison of Railroad, Bus, Truck, Water, and Air Transportation for Long and Short Haul"
1936 No Award. Subject: "Development in the Generation and Distribution of Power and Their Effect Upon the Consumer"
1937 ALLAN P. STERN, Case School of Applied Science. Subject: "The Influence of the Introduction of Labor Saving Machinery Upon Employment in the United States"
1938 EDWARD W. CONNOLLY, University of Detroit. Subject: "Economic Limitations in Engineering Design, With Concrete Examples"
1939 JAMES R. BRIGHT, Lehigh University. Subject: "The Economics of Investment in New Manufacturing Equipment—With Concrete Cases"
1940 FRANK DE POULD, Case School of Applied Science. Subject: "What Has Been the Effect of Technological Advance on Employment?"

STUDENT AWARD

1916 BOYNTON M. GREEN, Stanford University, "Bearing Lubrication"
HOWARD E. STEVENS, Rensselaer Polytechnic Institute, "An Investigation of the Dynamic Pressure on Submerged Flat Plates"
M. ADAM, Louisiana State University, "The Adaptability of the Internal Combustion Engine to Sugar Factories and Estates"
1917 H. R. HAMMOND and C. W. HOLMBERG, The Pennsylvania State College, "Study of Surface Resistance With Glass as the Transmission Medium"
1919 C. F. LEH and F. G. HAMPTON, Stanford University, "An Experimental Investigation of Steel Belting"
W. E. HELMICK, Stanford University, "An Experimental Investigation of Steel Belting"
1920 HOWARD G. ALLEN, Cornell University, "Wire Stitching Through Paper"
1921 KARL H. WHITE, University of Kansas, "Forces in Rotary Motors"
RICHARD H. MORRIS and ALBERT J. R. HOUSTON, University of California, "A Report Upon an Investigation of the Herschel Type of Improved Weir"
1923 CHARLES F. OLMSTEAD, University of Minnesota, "Oil Burning for Domestic Heating"
H. E. DOOLITTLE, University of California, "The Integrating Gate: A Device for Gaging in Open Channels"
1924 GEORGE STUART CLARK, Stanford University, "Two Methods Used for the Determination of the Gasoline Content of Absorption Oils in Absorption Plants"
L. J. FRANKLIN and CHARLES H. SMITH, Stanford University, "The Effect of Inaccuracy of Spacing on the Strength of Gear Teeth"
1925 HARRY PEASE COX, JR., Rensselaer Polytechnic Institute, "A Study of the Effect of End Shape on the Towing Resistance of a Barge Model"
W. S. MONTGOMERY, JR., and E. RAY ENDERS, JR., Pennsylvania State College, "Some Attempts to Measure the Drawing Properties of Metals"
1926 R. E. PETERSON, University of Illinois, "An Investigation of Stress Concentration by Means of Plaster of Paris Specimens"
CECIL G. HEARD, University of Toronto, "Pressure Distribution Over U.S.A. 27 Aerofoil With Square Wing Tips—Model Tests"
1927 ALFRED H. MARSHALL, Princeton University, "Evaporative Cooling"
ROGER IRWIN EBY, University of Washington, "Measurement of the Angular Displacement of Flywheels"
1928 CLARENCE C. FRANCK, Johns Hopkins University, "Condition Curves and Reheat Factors for Steam Turbines"
1929 FRANK VERNON BISTROM, University of Washington, "An Investigation of a Rotary Pump"
WILLIAM WALLACE WHITE, University of Washington, "An Investigation of a Rotary Pump"
1930 GERARD EDEN CLAUSSEN, Polytechnic Institute of Brooklyn, "High-Temperature Oxidation of Steel"
HAROLD L. ADAMS and RICHARD L. STITH, University of Washington, "A Wind Tunnel for Undergraduate Laboratory Experiments"
1931 JULES PODNOSSOFF, Polytechnic Institute of Brooklyn, "Pressure and Energy Distribution in Multi-Stage Steam Turbines Operating Under Varying Conditions"
1932 H. E. FOSTER, JR., University of Tennessee, "Factors Affecting Spray Pond Design" (Undergraduate Award)
WILLIAM A. MASON, Stanford University, "An Experimental Investigation of the Flame Propagation in Internal-Combustion Engines" (Postgraduate Award)
1933 HUGO V. CORDIANO, Polytechnic Institute of Brooklyn, "Thermal Analysis of Lithium-Magnesium System of Alloys" (Undergraduate Award)
JAMES A. OSTRAND, JR., Princeton University, "Sudden Enlargement in the Open Channel" (Postgraduate Award)
1934 H. REYNOLDS HUDSON, Georgia School of Technology, "Dynamic Balance and Functional Utility Applied to Automotive Design" (Undergraduate Award)
1935 CHARLES P. BACHA, Rutgers University, "The Behavior of Metals Subjected to Combined Stress" (Postgraduate Award)
ROBERT W. BEAL, Oregon State College, "Do Lubricating Oils Wear Out?" (Undergraduate Award)
1936 LEON B. STINSON, Oklahoma Agricultural and Mechanical College, "Polymerized Motor Fuels; Their Economic Significance" (Undergraduate Award)
DEWITT D. BARLOW, JR., Princeton University, "The Critical Speeds of Lateral Vibrations of Shafts with Gyroscopic Effects" (Postgraduate Award)
1937 GINO J. MARINELLI, Rensselaer Polytechnic Institute, "Investigation of the Towing Resistance of a Model Submarine Hull" (Undergraduate Award)
1938 MARSHALL C. LONG, Princeton University, "An Investigation Into the Angular Characteristics of an Adjustable Blade Current Meter" (Postgraduate Award)
DONALD C. MCSORLEY, Michigan State College, "Humidity Insulation" (Undergraduate Award)
1939 DAVID T. JAMES, Michigan State College, "Bells—Concerning Their Tones" (Undergraduate Award)
1940 GEORGE W. SHEPHERD, JR., Princeton University, "An Automatic Mechanical Control for Synchronizing Prime Movers" (Postgraduate Award)
EDWARD D. ROWAN, Oregon State College, "Powder Metallurgy (Undergraduate Award).

FREEMAN TRAVEL SCHOLARSHIP

1927 HERBERT N. EATON
1928 BLAKE R. VAN LEER
1929 ROBERT T. KNAPP
1931 REGINALD WHITAKER
1932 G. ROSS LORD
1933, 1934 H. J. CASEY
1935, 1936 VICTOR L. STREETER

HONORARY MEMBERS

HONORARY MEMBERS IN PERPETUITY

Alexander Lyman Holley, Founder of the Society. Died 1882.

John Edson Sweet, Founder of the Society. Died 1916.

Henry Rossiter Worthington, Founder of the Society. Died 1880.

DECEASED HONORARY MEMBERS

	Elected	Died
Horatio Allen	1880	1889
Sir William Arrol	1905	1913
Sir John Audley Frederick Aspinall	1911	1937
William Wallace Atterbury	1925	1935
Sir Benjamin Baker	1886	1907
Johann Bauschinger	1884	1893
Sir Henry Bessemer	1891	1898
Sir Frederick Joseph Bramwell	1884	1903
John Alfred Brashear	1908	1920
Gustave Canet	1900	1908
Andrew Carnegie	1907	1919
Daniel Kinnear Clark	1882	1896
Rudolph Julius Emmanuel Clausius	1882	1888
Sir John Goode	1889	1892
Peter Cooper	1882	1883
Charles de Fréminville	1919	1936
Carl Gustaf Patrick de Laval	1912	1913
Rudolph Diesel	1912	1913
James Dredge	1886	1906
Victor Dwelshauvers-Dery	1886	1913
Thomas Alva Edison	1904	1931
Alexandre Gustave Eiffel	1889	1923
Marshal Ferdinand Foch	1921	1929
Sir Charles Douglas Fox	1900	1921
John Ripley Freeman	1932	1932
John Fritz	1900	1913
Major-General George Washington Goethals	1917	1928
Franz Grashof	1884	1893
Rear-Admiral Robert Stanislau Griffin	1920	1933
Otto Hallauer	1882	1883
Charles Haynes Haswell	1905	1907
Nathanael Greene Herreshoff	1921	1938
Friedrich Gustav Herrmann	1884	1907
Gustav Adolph Hirn	1882	1890
Joseph Hirsch	1889	1901
Ira N. Hollis	1928	1930
Robert Woolston Hunt	1920	1923
Benjamin Franklin Isherwood	1894	1915
Henri Léauté	1891	1916
Erasmus Darwin Leavitt	1915	1916
Henri Le Chatelier	1927	1936
Anatole Mallet	1912	1919
Charles H. Manning	1913	1919
Rear-Admiral George Wallace Melville	1910	1912
The Honorable Sir Charles Algernon Parsons	1920	1931
Charles Talbot Porter	1890	1910
Auguste C. E. Rateau	1919	1930
Sir Edward J. Reed	1882	1906
Franz Reuleaux	1882	1905
Calvin Winsor Rice	1931	1934
Palmer C. Ricketts	1931	1934
Henri Adolphe-Eugene Schneider	1882	1898
Charles M. Schwab	1918	1939
C. William Siemans	1882	1883
Viscount Ei-ichi Shibusawa	1929	1931
Ambrose Swasey	1916	1937
Elihu Thomson	1930	1937
Henry Robinson Towne	1921	1924
Henri Tresca	1882	1885
William Cawthorne Unwin	1898	1933
Samuel Matthews Vauclain	1920	1940
Oskar von Miller	1912	1934
Francis A. Walker	1886	1897
Worcester Reed Warner	1925	1929
George Westinghouse	1897	1914
Sir William Henry White	1900	1913
Sir Alfred Fernandez Yarrow	1914	1932

LIVING HONORARY MEMBERS

	Elected
William Lamont Abbott	1940
Lorenzo Allievi	1937
Robert W. Angus	1940
Edmund Bruce Ball	1939
Hutchinson I. Cone	1936
Mortimer Elwyn Cooley	1928
Alex Dow	1936
William Frederick Durand	1934
Arthur M. Greene, Jr.	1940
Herbert Clark Hoover	1925
David Schenck Jacobus	1934
Masawo Kamo	1929
Dexter Simpson Kimball	1939
Albert Kingsbury	1940
Charles Thomas Main	1939
George A. Orrok	1936
Grande Ufficiale Ing. Pio Perrone	1920
Edwin Jay Prindle	1939
James A. Seymour	1940
William H. Tschappat	1938
Henry Hague Vaughan	1939
Right Honorable Lord Weir	1920
Orville Wright	1918

PAST-PRESIDENTS

A list of past vice-presidents, managers, treasurers, and secretaries will be found in the 1930 Record and Index, pages 10-12. Dates in parentheses denote year of death.

Alexander Lyman Holley, *Chairman of the Preliminary Meeting for Organization of The American Society of Mechanical Engineers* (1882)

1880–1882	Robert Henry Thurston (1903)
1883	Erasmus Darwin Leavitt (1916)
1884	John Edson Sweet (1916)
1885	Josephus Flavius Holloway (1896)
1886	Coleman Sellers (1907)
1887	George H. Babcock (1893)
1888	Horace See (1909)
1889	Henry Robinson Towne (1924)
1890	Oberlin Smith (1926)
1891	Robert Woolston Hunt (1923)
1892	Charles Harding Loring (1907)
1893–1894	Eckley Brinton Coxe (1895)
1895	Edward F. C. Davis (1895)
1895	Charles Ethan Billings (1920)
1896	John Fritz (1913)
1897	Worcester Reed Warner (1929)
1898	Charles Wallace Hunt (1911)
1899	George Wallace Melville (1912)
1900	Charles Hill Morgan (1911)
1901	Samuel T. Wellman (1919)
1902	Edwin Reynolds (1909)
1903	James Mapes Dodge (1915)
1904	Ambrose Swasey (1937)
1905	John Ripley Freeman (1932)
1906	Frederick Winslow Taylor (1915)
1907	Frederick Remsen Hutton (1918)
1908	Minard Lafever Holman (1925)
1909	Jesse Merrick Smith (1927)
1910	George Westinghouse (1914)
1911	Edward Daniel Meier (1914)
1912	Alexander Crombie Humphreys (1927)
1913	William Freeman Myrick Goss (1928)
1914	James Hartness (1934)
1915	John Alfred Brashear (1920)
1916	David Schenck Jacobus
1917	Ira Nelson Hollis (1930)
1918	Charles Thomas Main
1919	Mortimer Elwyn Cooley
1920	Fred J. Miller (1939)
1921	Edwin S. Carman
1922	Dexter Simpson Kimball
1923	John Lyle Harrington
1924	Frederick Rollins Low (1936)
1925	William Frederick Durand
1926	William Lamont Abbott
1927	Charles M. Schwab (1939)
1928	Alex Dow
1929	Elmer Ambrose Sperry (1930)
1930	Charles Piez (1933)
1931	Roy V. Wright
1932	Conrad N. Lauer
1933	A. A. Potter
1934	Paul Doty (1938)
1935	Ralph E. Flanders
1936	William L. Batt
1937	James H. Herron
1938	Harvey N. Davis
1939	Alexander G. Christie
1940	Warren H. McBryde

Index to Society Records, Part 1

The page numbers in this section are preceded by the letters "RI," which are omitted in the following index.

The Combustion of Waste-Wood Products

By H. W. BEECHER[1] and R. D. WATT,[2] SEATTLE, WASH.

This paper is not intended to be a rigorous argument or a scientific treatise advocating any particular procedure in producing steam from the combustion of wood waste, but rather the compilation of essential facts made apparent by many years of experience in dealing with this rather abstruse subject.

THE waste products resulting from the manufacture of lumber, plywood, or cellulose for conversion into pulp are available as fuel. Manifestly, any portion of the log which can be economically converted into more valuable material should neither be classified as waste wood nor used for fuel. Sawdust and shavings can be handled for burning without further processing. Slabs, edgings, trimmings, and other waste products require further size reduction to prepare them for rapid combustion, easy transportation, and convenient handling. Such material is usually processed by a mechanical masticator, commonly known as a "hog." The product so obtained, together with sawdust and shavings, forms a mixed fuel called "hog fuel." The term "hog fuel" will be used to describe the waste-wood products dealt with in this paper, whether they be sawdust, shavings, or a mixture of these with hogged materials.

Economic Aspects

The cheapest power for operation of a sawmill is energy generated from steam produced by burning the resulting waste materials. This waste must be removed or destroyed and is available to the producing sawmill without transportation costs. Whenever sawmills are not located within economical transportation distance from external hog-fuel markets, large investments in refuse burners are necessary in order to dispose of excess waste fuel. In a country with an abundance of cheap hydroelectric power, fuel must be inexpensive to permit steam-plant competition, except where steam is required for process.

The mill production cost for the hogged portion of fuel is from 10 cents to 15 cents per unit, largely made up of power, operation, and maintenance expenses of the hogging equipment. The mill operator should receive a reasonable return on his investment and, if possible, secure a profit on his waste material. Prices charged for hog fuel vary from 50 cents to $1.50 per unit at the producing mill, depending upon the supply and demand and not upon the cost of production. If a mill is isolated from the market in a community where the other fuel-consuming activities also produce hog fuel as a by-product, there is competition for the consuming market and the mill operator must then be satisfied with smaller returns. Conversely, if the producing mill is located in an industrial center containing fuel-consuming plants which are not producers of fuel, an absorbing market is available, permitting the mill operator to secure a profit on his waste materials. In recent years the revenues from hog-fuel sales have represented a large part of the total profits of many mills.

The large volume and weight of hog fuel per available Btu makes the transportation cost loom large in the total cost to the consumer. The economical marketing zone is limited by transportation costs. Frequently, greater cost is involved in transportation and handling than the actual price paid by the consumer to the producer for the fuel at the point of manufacture. In spite of high transportation costs, hog fuel is generally available to the consumer at a cost per million Btu, comparing favorably with the costs of other types of fuel in the lowest-fuel-cost areas of the United States. Hog fuel is, therefore, the principal fuel used in the Pacific Northwest for steam production.

The high moisture content of hog fuel materially reduces the obtainable thermal efficiencies of boiler plants, as compared with the efficiencies secured with other fuels. This necessitates comparison of hog fuel with other fuels on the basis of their relative cost per available Btu. Many consumers of hog fuel pay as little as 50 cents a unit, delivered. In other Northwest plants the hog-fuel cost reaches $3.50 per unit. A unit of hog-fuel measurement occupies 200 cu ft. An average unit of hog fuel will contain approximately 20,000,000 Btu. Boiler-plant efficiencies with hog fuel vary, depending upon the type of installation, the percentage of rating at which the boiler plant operates, and whether air heaters are installed for recovery of additional heat from the boiler gases. These efficiencies vary from 45 per cent on the poorer installations to 65 per cent on the more modern and better equipped boiler plants.

To indicate the general low cost of hog fuel for the production of steam, it may be noted that with 60 per cent efficiency the available heat per average unit would be 12,000,000 Btu. At a cost of $1 per unit, the corresponding cost of steam production would be $8^1/_3$ cents per million Btu input. With fuel oil costing $1 per bbl and with 83 per cent boiler efficiency, the heat available in steam would be 5,200,000 Btu per bbl and the corresponding cost per million Btu input would be 19 cents. With coal having a heat value of 12,500 Btu per lb and costing $4 per ton and with an average boiler efficiency of 80 per cent, the corresponding cost of steam per million Btu input would be 20 cents. This comparison indicates that, in so far as a consideration of the combustion of hog fuel is concerned, a low-cost fuel is involved which has not yet encouraged the engineering research or the capital investment which would be warranted were it a higher-priced commodity.

Measurement of Hog Fuel

Neither buyers nor sellers of hog fuel have been willing to spend money to measure accurately fuel having such low cost per million available Btu. The seller has not had competition from fuels other than hog fuel from competing mills, as this fuel has been used principally in localities where neither oil nor coal has been competitive. The buyer, realizing the economic advantage in using hog fuel, as compared with other available fuels, has been unmindful of the advantages to be obtained by purchasing scientifically and determining the actual fuel values obtained for his dollar.

To compare and purchase on an available Btu basis one must purchase by weight, rather than by measurement, and make proper corrections for average moisture content. The simplicity and comparatively low cost of volumetric measurement has delayed the adoption of the more scientific system. The general adoption of volumetric measurement dates back 30 years when hog-fuel prices were yet lower than those prevailing today. The "unit" on which most hog fuel is purchased and sold contains 200 cu ft (material and voids) irrespective of the compacting of

[1] Consulting Engineer.

[2] Assistant to H. W. Beecher.

Contributed by the Fuels Division and presented at the Fall Meeting, Spokane, Wash., Sept. 3–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

the fuel brought about by the shape, size, or handling of the measuring containers. The process of mixing hog fuel and sawdust, in which the sawdust tends to fill the voids, provides greater fuel content per unit of measure. No allowance or credit is ordinarily given by the buyer for the greater number of Btu in a compacted unit. As a result, there is wide variation in the heating value per unit as sold. Another element of variation which is waived with the present measurement basis is moisture content with its serious effect on the availability of the heat units. With the present measurement basis, a buyer may obtain more heat per dollar from a unit for which he pays $1.50 than from another unit for which he pays only $1. The development of weighing-belt scales shortens the time until much of the hog fuel used in large plants will be weighed and, with proper moisture determination, be bought, sold, and compared on a Btu basis.

Availability of Waste-Wood Fuel

As long as lumber is cut from logs, there will be waste for which the primary market will be among the consumers of cheap fuel. A detailed survey by the Forest Service and reported by Allen H. Hodgson showed that, in the manufacturing of slightly over 10,000,000,000 fbm of green, rough-sawed lumber in the Douglas fir region in the year 1929, over 619,000,000 cu ft of solid-measure normal sawmill waste was produced. An analysis of the total volume of 1,354,000,000 cu ft of sound wood in the logs showed that approximately 67 per cent (911,000,000 cu ft) was converted into green, rough-sawed lumber; nearly 19 per cent (261,000,000 cu ft) became slabs, edgings, and trimmings; the balance, 14 per cent (191,000,000 cu ft) was sawdust. In addition to the sound wood there was approximately 167,000,000-cu ft solid measure of bark. This indicates that of the solid-wood material, inclusive of bark represented by the logs as delivered to the sawmill, 41 per cent is so-called "waste" and available for fuel.

We have been unable to find the result of any studies made to determine the percentage of solid wood in the tree represented by the solid wood in the saw logs. It is well known that the tops and branches cannot be economically handled, transported, and utilized for the production of lumber, as the available wood content, when converted into lumber, will not bring sufficient revenue to the mill to cover the cost of removing this material from the forest. The branches and tops left in the forest are a fire hazard and, if the cost of handling and removing them approached the price obtainable for the materials produced, they would be removed by the mills at a loss rather than be left in the forest as a menace. It is reasonable to assume that the material left in the forest is sufficient to make up the difference between 41 and 50 per cent, and to state broadly that, of the wood content of the average tree as logged and utilized in the Northwest, less than 50 per cent is converted into lumber and its allied products. The balance is an economic waste except for the value recovered as fuel.

While there may be some slight improvement in utilization, the greatest encouragement for the conservationist comes from the possibility of chemically treating selected portions of existing waste for production of cellulose products. Another field is fermentation and production of alcohol. A third utilization process is destructive distillation which would give charcoal, numerous by-products, and some steam for power production by burning the gas after cleansing of by-products. All these possibilities justify the statement that, as long as sawmills in the Northwest are operating at reasonable capacities, hog fuel in abundance will be available for industrial use.

There will remain a wide price range to the consumer at his place of use. As a consumer goes farther afield to secure his fuel in competition with other consumers, the more fortunately located sawmills with lower transportation costs will raise their prices and derive greater profits. Greater consuming markets will advance the prices so that small mills, beyond the zone of economical transportation at present fuel prices, will be provided with a market for materials now destroyed in refuse burners. Such destruction, while wasteful, is necessary without a market for this material.

Transportation of Hog Fuel

Where water transportation is available without rehandling, it is by far the cheapest method of moving hog fuel. Hog fuel is towed on barges as great a distance as 350 miles.

In certain localities, transportation must be by rail in specially fitted cars. Tariffs are either per car or per 100 lb weight. In either case it is desirable to have car contents a maximum. Cars are constructed with sides extending to maximum clearance height for the division in which they are to operate. A 50-ft car may be constructed to handle 35 units. However, most cars carry from 20 to 30 units.

The use of trucks for hog-fuel transportation is increasing. Special bodies carry from 4 to 6 units where regulations will permit. They load by gravity from bins, or conveyers, and are fitted with power dumps by which they are discharged into hoppers. Truck-transportation cost is dependent upon the length of haul which affects the portion of hourly truck and driver costs chargeable to each unit of hog fuel.

Each transportation problem must be independently treated to obtain the least possible cost. No set schedule of probable cost can be suggested as generally applying to any of the three methods outlined.

In rare cases, the hog-fuel-producing plant is situated near a consuming market and very low transportation costs can be obtained by use of belt conveyers between plants.

Except where necessary to elevate fuel steeply or to take off at several intermediate points, belt conveyers should be used. Even with multiple-point discharge, it frequently becomes desirable to use trough belts with traveling trippers or flat belts with unloading slices rather than to install scraper conveyers.

Belt conveyers are cheaper to operate than flight conveyers as they require less power, attention, and maintenance. Where belt conveyers cannot be conveniently installed, it is necessary to use special scraper or drag conveyers. All boiler-feed conveyers should be provided to convey more than current requirements and to return surplus to supply source, rather than to attempt regulation of the conveyed fuel supply in synchronism with fuel consumptions.

The amount of fuel storage required depends upon reliability of supply, possible interruption to transportation, and the necessity of avoiding plant outage. Plant consumptions, load factors, and operating intervals also affect the amount of storage which should be provided. The possibility of using oil or other more expensive fuel for emergency operation, with its attendant higher fuel cost, should be weighed against the fixed charges on average hog-fuel storage and recovery systems.

Heating Value

The high percentage of oxygen in wood reduces the heat content per pound as it is combined with carbon and hydrogen to form carbohydrates and, therefore, the total heat of combination of the combustibles is not all available. The manner in which these three elements are combined is not definitely known and the use of Dulong's formula, as applied to the ultimate analysis of wood, will not result in a Btu value corresponding to that obtained from calorimetric determinations. Hog fuel as normally delivered to the furnaces contains a high percentage of moisture. A portion of this is extraneous moisture, either resulting from wet logs, water lubrication of saws, or rain when fuel has been ex-

posed. Most of the moisture, however, is in the cellular structure of the wood. The fuel, as received from the average sawmill, contains material with high moisture content from the sap wood of the log, material with a medium moisture content from the heartwood, and material with comparatively low moisture content from wood dried in commercial dry kilns.

In computing combustion results, moisture determinations are reported as the percentage of the total weight of wood and moisture represented by the moisture. This means that fuel containing 50 per cent moisture contains 1 lb of water per lb of dry fuel. With the high oxygen content of wood there would be $1^3/_4$ lb of water per lb of combustible. If the oxygen were combined with the hydrogen, as assumed in Dulong's formula, 50 per cent moisture in the fuel would correspond to approximately 2 lb of water per lb of ununited or available combustible. It is interesting to note, when the moisture content is increased from 50 to 60 per cent, the weight of moisture in the fuel is increased from 1 to $1^1/_2$ lb per lb of dry fuel. The hog fuel used in industrial plants of the Northwest will average from 25 per cent moisture, when principally kiln-dried material, to from 57 to 60 per cent moisture when largely green hemlock.

The available heat in hog fuel is a function of the moisture content. The heat necessary to raise the temperature of the wet fuel to 212 F, evaporate the water, and superheat the vapor to the exit-gas temperature is not available for steam production. This accounts for a substantial portion of the total losses in hog-fuel combustion.

All species of wood considered herein have approximately the same heating value on a bone-dry basis which will average 8900 Btu per lb of dry wood. However, some species are better fuels than others. Hemlock is not as good as fir. Spruce is better then hemlock, but poorer than fir. Cedar is a light fuel and requires a specially designed furnace for good results. Hemlock, as ordinarily available as fuel, has a high moisture content and does not readily part with its moisture. At least 20 to 25 per cent more capacity can be obtained from given furnaces and combustion chambers with fir fuel having about 45 per cent moisture content than with hemlock having 57 per cent moisture.

The heating value of stored hog fuel varies with the time in storage. Storage of hog fuel in the open decreases available Btu in fuel faster than storage under cover. This loss of heat value is attributed to slow oxidation which takes place at low temperatures. Cultures have been made from samples of hog fuel after storage over a considerable period which show an indication of molds and other wood-destroying fungi. These reactions are exothermic and the heat is lost.

Combustion

Coal and oil in age-long processes have both been formed from vegetable matter. All wood, coal, and oil contain the same elemental combustible materials; these elements are, however, combined in different ratios. A typical ultimate analysis of wood is as follows:

	Per cent
Carbon	50.31
Hydrogen	6.20
Oxygen	43.08
Nitrogen	0.04
Ash	0.37

Through the ages, during which coal and oil have been subjected to heat and pressure, much of the oxygen and moisture originally contained in the wood or other cellulose matter from which they were formed have been driven off, leaving a greater concentration of combustibles.

Nearly 45 per cent of the dry weight of wood, independent of the species, is oxygen. The hydrogen-to-carbon ratio in wood is of the same order as in oil and, therefore, for the same excess air, the percentage of water vapor as compared to dry gases will be approximately the same for these two fuels. Coals, as a rule, have much lower hydrogen-to-carbon ratios and, therefore, give combustion gases containing lower percentages of moisture. The heating value of the fixed carbon in wood fuel amounts to from 15 to 20 per cent of the total heat in the fuel. The high moisture and volatile contents of hog fuel delay combustion which proceeds as follows:

1 The driving off of the moisture content and raising the wood to a temperature at which volatiles will be driven off;

2 The actual distillation of volatiles;

3 The combustion of the fixed carbon.

The high oxygen content of wood with its low nitrogen content reduces the percentage of nitrogen in hog-fuel flue gas. Coal of typical analysis, if completely burned without excess air, would produce $18^1/_2$ per cent CO_2 in the combustion gases; similarly, oil of typical analysis, if burned without excess air, would produce $15^1/_2$ per cent CO_2; wood of the typical analysis quoted will give, if completely burned without excess air, approximately 20 per cent CO_2.

The wood itself contains but little noncombustible in the form of ash; however, hog fuel as normally fired may carry with it appreciable quantities of ash-forming material in the nature of extraneous matter embedded in the bark or wood fibers and not removed in preparation, transportation, and handling. This may consist of small pebbles, sand, and shells. Logs which have been transported in salt water give off gases containing salt fume which assists in lowering the fusion temperature of the noncombustible and accelerates the deposit of slag on boiler tubes.

With deep fuel beds, most of the fixed carbon, undoubtedly, leaves the fuel bed as carbon monoxide where it unites with additional oxygen to burn to the dioxide. The incandescent carbon adjacent to the grates burns to the dioxide and then, in passing further through the incandescent carbon, is reduced to the monoxide.

In the cellular type of furnace, it is important to provide secondary or overdraft air. The conical pile of fuel is too thick, except around its edges, to pass the necessary air for rapid combustion. The closing of secondary-air admission ports, resulting from too thick a fuel bed, is quickly evident in the smoking of furnaces. The admission of overdraft air through the grates in the front of the furnace with little resistance to the passage of such air decreases the negative pressure in the furnace and lowers the required average draft throughout the setting. Any decrease in required draft is desirable to avoid infiltration in the convection sections where excess air decreases the efficiency of the boiler.

The standard method of feeding fuel to flat-grate cellular-type furnaces is through feedhole openings located in the furnace roof, the fuel being transported to the furnace through chutes. Reasonable precautions are necessary to limit the amount of air entering the furnace through these chutes; any air so admitted decreases the air to the preheater and also results in furnace stratification. In spite of such precautions, the falling fuel produces an injector action and entrains considerable quantities of air.

Design of Furnaces to Burn Hog Fuel

The problems of proper combustion of hog fuel are greatly increased by the necessity of providing furnaces suitably designed for fuels varying in size from dust to pieces having 3 to 5 cu in. of content and for fuels of variable moisture content. Frequently, slugs of dry and highly combustible fuel are followed by slugs of wet fuel which form a damp blanket on the fuel pile. In the case of hopper-fired sloping-grate furnaces, one side of a hopper may contain dry fuel and the other side wet fuel.

The designer must provide furnaces to handle properly the wet fuel and, at the same time, not to punish unduly refractories during the periods in which only dry fuel is fed. To produce the best average combustion conditions, much study has been given to the use of sloping-grate furnaces where the fuel is admitted in a comparatively thin and uniform layer over a drying hearth, in which portion of the furnace reflected heat is utilized to drive off the moisture and start the distillation process necessary before the fixed-carbon content of the fuel can be ignited. Following this section of the furnace, the fuel flows over grates and, as the volatile content is driven off, combustion of the fixed carbon is maintained by the air passing through the grates and fuel bed.

Theoretically, such furnaces would be preferred to flat-grate, conical-pile furnaces with which by far the greatest number of hog-fuel-fired boilers are equipped. Practically, difficulties are encountered with sloping-grate furnaces caused by:

1 The fuel not being uniform in size and, therefore, containing streaks, or pockets, of greater density than adjacent areas, leading to the formation of blowholes through the fuel.

2 The fuel, not being of uniform moisture content, leads to the formation of areas in which distillation and ignition proceed more rapidly than in adjacent areas, thus resulting in the same formation of blowholes.

With a fuel as light as wood, particularly after the moisture and volatiles have been driven off, leaving charcoal cinders, these blowholes lift the cinders from the grates, depositing them at the foot of the sloping section and, in their formation, prevent the fuel from above the blowhole cascading to cover the hole. If too great ashpit pressures are used, this formation of blowholes is accentuated.

The accumulated charcoal cinders at the toe of the sloping grates affords such high resistance to the passage of air that insufficient air passes through this material. This prevents the combustion at the toe of the grate proceeding with sufficient rapidity to obtain high ratings per square foot of grate area, as the limiting rate for inflow of fuel over the drying-hearth section is the rate at which the fixed carbon can be consumed at the lower end of the grate. Even though sloping-grate furnaces have been tried in the Northwest with 15 to 16 ft of total length, obtainable capacities per foot of width of furnace have been less than those possible with well-designed furnaces of the so-called cellular type. As a result of the greater capacity obtainable in the latter furnace, most of the installations made in recent years have been of this design.

It is possible that extremely long sloping furnaces with special means for controlling the rate of feed and for cleaning the accumulated slag at the toe of the grate, with controlled and zoned air supply, could be developed to give results comparable with those obtained with a flat furnace. Such an installation would involve capital expenditures which do not appear to be commercially justified, as they could not improve materially upon the efficiencies obtained with the present flat cellular-type furnace. An advantage of the cellular type of furnace is the ability to operate a boiler at reduced rating while burning down and cleaning the slag from the grates in one of the multiple cells. Cell-type furnaces are constructed with widths for individual cells ranging from $6^1/_2$ to $8^1/_2$ ft, which appear to be the economical limits of conical piles to be covered by single feedholes.

The combustion-chamber volume, gas-travel length before convection surfaces, and the cross-sectional area of combustion space are related and important in hog-fuel combustion. In comparable installations, the gas weights with hog fuel are approximately 1.7 times the gas weights with oil, and approximately 1.25 times the gas weights with coal. This increased gas weight results in lower combustion-chamber temperatures which are further reduced by the high moisture content of the hog-fuel gases. The decreased temperature does not entirely offset the increased gas weights and larger cross-sectional areas are required when burning hog fuel to give comparable velocities in the combustion space. These factors make it essential to provide larger combustion spaces with hog fuel than with other fuels.

With the modern boiler installation, the increased capacity obtainable with preheated air has been largely responsible for the installation of preheaters rather than any gain in efficiency resulting from their use. When a preheater installation is charged with the extra capital, operating, and power costs, made necessary by the installation of forced- and induced-draft fans, and the necessary gas and air ducts, the low cost per Btu of the fuel precludes the justification of air preheaters on a strictly fuel-saving basis. With hog fuel it is impossible to obtain as low exit-gas temperatures from air preheaters as with other fuels. The high exit-gas temperatures, in part, result from the fact that only 75 to 80 per cent of the air required for combustion can be passed through the air preheater.

With the general introduction of water-cooled combustion chambers in an endeavor to reduce brickwork maintenance, the addition of the preheater has been found desirable in order to decrease the size of the combustion chamber and the length of gas travel necessary from the furnaces to the convection surfaces.

The use of preheaters has made it necessary to use water-cooled grates to avoid excessive grate maintenance. Water-cooled grates have also proved desirable to facilitate grate cleaning. The slag formed from the foreign matter brought in with the fuel does not adhere tenaciously to the water-cooled grates; whereas, with uncooled grates, it is removed with difficulty. Several designs of water-cooled grates have been developed for this service. The heat absorbed in the cooling water is low-potential heat and must be subtracted from the heat available for the production of steam. In many installations the heat obtained in grate-cooling water is used to heat condensate, or make-up water, in this manner supplanting heat which would otherwise be supplied by bled steam which had produced power or by the exhaust from noncondensing auxiliaries. It is, therefore, important in the design and construction of water-cooled grates to provide an arrangement for cooling which will extend the life of the grate, provide for easy cleaning, and, at the same time, extract from the grates and from the preheated air passing through them a minimum amount of this low-potential heat.

Driers

Hog-fuel driers offer attractive potential savings to the power-plant operator. The flue gas leaving an air heater at approximately 500 F contains sufficient heat to remove about $^1/_2$ lb of water per lb of dry wood without dropping the temperature of the gas so low that condensation difficulties will arise. In addition to the savings, the drying of hog fuel gives considerably increased capacity per square foot of grate.

Although hog-fuel driers seemingly have a broad field, the volume of fuel to be dried per available Btu makes necessary a drier of such large physical dimensions that the fixed charges and the operating and maintenance expenses make it difficult to justify the investment.

Several different types of hog-fuel driers have been proposed and tried in this country and abroad, but the authors do not know of any design which has proved completely satisfactory. There is a definite field for a satisfactory hog-fuel drier, but until all of the mechanical difficulties with the prevailing designs can be successfully solved their use will not become extensive.

Cinder Nuisance

The cinder nuisance from hog-fuel-burning plants has increased with higher firing rates required in the modern high-duty boilers

equipped with forced- and induced-draft fans and air preheaters. Modern hog-fuel-burning plants are providing either mechanical separators or flue-gas washers for cinder removal. Starting with the use of large single cyclone dust separators, with relatively poor efficiencies on the fine light cinder particles, the necessity of securing better cinder elimination has led to a trial of various designs of mechanical devices, and to the development of special wet gas washers.

There is but little information available concerning the dust loadings of boiler-exit gases. At recent biddings by prominent boiler manufacturers, great divergence in the assumed percentages of unconsumed combustible showed that there was apparently no actual knowledge of the dust loadings to be expected. Bidders allowed as low as 0.25 per cent unconsumed combustible loss, while others allowed as much as 7.5 per cent. Some bidders gave constant percentages over the entire range of ratings, although it is evident that the unconsumed-carbon loss in the flue gas will increase with the rate of firing.

Recently, extensive and carefully conducted tests have been made by one of the country's foremost manufacturers of gas-cleansing equipment. Tests were made at various rates of operation, with and without preheat on distinctly different types of boilers. These tests will give the first authentic data on the dust loadings in flue gases of hog-fuel-fired boilers under variable conditions of firing, fuel, and rating. Unfortunately, this information is not as yet available to the public. The cinder problem when burning hog fuel at high firing rates is a real one. The industry may expect valuable information affecting the best means of cinder collection from such tests and the developments resulting therefrom.

Boiler Capacities

A high and narrow boiler is less expensive than a low and wide one of equal heating surface. The problems of boiler-plant design are not so much the provision of heat-absorbing surface as in obtaining suitable furnaces for combustion of the wet fuels at high rates per unit area of furnaces. Tandem furnaces do not operate as well as furnaces with a single feedhole per cell. Any boiler should have a minimum of two cells to permit carrying part load during grate-cleaning periods, while three cells permit carrying greater loads during such periods. Boiler plants with several boiler units can use fewer cells per unit without material loss of plant capacity. Many installations have furnaces splayed to a greater over-all width than the boiler to permit greater furnace capacity with narrow boilers.

Numerous factors affect the capacity obtained from hog-fuel-fired boilers. The following table is intended to indicate in a general way what capacity should be expected from a well-designed furnace cell of the general dimensions used in modern installations in the Northwest. Values are given for cells with and without preheat and with good fuels of different moisture content:

Moisture in fuel, per cent	Btu input per sq ft of grate area: Without preheat	With preheat
40	680000	850000
48	550000	690000
56	400000	500000

Caution should be exercised in the use of the capacities tabulated as they reflect what can be accomplished under good conditions and in properly designed furnaces.

Recent Developments in Burning Midwestern Coals on Water-Cooled Underfeed Stokers

By H. C. Carroll,[1] Chicago, Ill.

The author points out that there is an increasing trend toward one-boiler-unit operation in a single plant. This is due to a better understanding of boiler and furnace design, feedwater treatment, heat-recovery devices, and fuel-burning equipment, all tending to promote increased efficiency, greater ease of operation, and greater reliability. The application of water-cooling to underfeed stokers can properly be compared to the application of water-cooling to refractory furnace walls; and it is the author's belief that it will prove fully as beneficial in protecting the grate structure of the stoker as has water-cooling proved its value in the protection of refractory furnace walls. The complications involved in applying water-cooling to underfeed stokers are outlined in the paper and numerous successful installations using Midwestern coals are described.

The first attempt to water-cool multiple-retort underfeed stokers was made in 1929, where the tuyères reciprocated. Flexible hose was necessary to supply water to the cooling tubes, which proved impractical. In 1933, one Midwestern plant found it necessary to burn a cheap strip-mine Illinois coal with preheated air, or convert to gas. The multiple-retort stoker in this plant had stationary tuyères which were cooled by fixed tubes. These were subsequently extended to protect the entire air-emitting surfaces of the stoker.

These original units were protected by small tubes with forced circulation, obtained by taking water from the boiler drum through a booster pump, and pumping it through the stoker cooling tubes and back to the boiler drum. Later designs embodied natural-circulation tubes, connected in the same manner as conventional waterwalls.

At present there are about thirty installations of water-cooled stokers in commercial operation. These vary in size from 35,000 to 225,000 lb per hr, with a combined steam-generating capacity in excess of 3,000,000 lb of steam per hr. These units are distributed over a wide area, using a variety of coals from the eastern Atlantic fields to Nebraska, including Pennsylvania, Ohio, Indiana, Illinois, Kentucky, Iowa, and Kansas coals.

Midwestern Coals

Coal which moves into the Midwestern section of the country from mines located in this area comes from operations in the states of Ohio, western Kentucky, Indiana, Illinois, and Iowa. While the Ohio coals are not, strictly speaking, Midwestern with regard to geographical location, they bear a relation in rank to some of the coals from the Midwestern fields, and are likewise used for industrial purposes there. Table 1 shows in a general way the range of coals coming from these fields.

The approximate tonnages produced during the year 1938, from the states mentioned are as follows:

Ohio	17,920,000
Western Kentucky	7,275,000
Indiana	14,050,000
Illinois	40,650,000
Iowa	3,250,000
Kansas	2,560,000

It will be noted that in general the Midwestern coals are relatively high in moisture, high in ash, high in volatile matter, low in fixed carbon, high in sulphur (except the southern Illinois coals), low in fusion, and dry-heat values from 9800 Btu to 13,800 Btu. Midwestern coals do not stand weathering very well, structurally are not hard, and the grindability index is low. Most of it is classed as free-burning, noncoking, and, due to its high volatile content, easily ignited.

Burning Rates

One of the most important factors in the successful burning of Midwestern coals is the rate at which the coal is burned. We have adopted the projected area of the grate on the underfeed stoker as the area to be used in the calculation of fuel-burning rates.

Where the unit is designed for base-load conditions over the entire period of operation, definite burning rates can be established. However, as is often the case in industrial and isolated city plants which do not have connections with other plants, a range of burning rates must be established which will prove not only economical over the entire range of load, but also will prevent undue maintenance under peak-load conditions.

Keeping in mind the trend toward the single-boiler-unit operation, the water-cooled stoker presents a new possibility in this direction, especially where preheated combustion air is feasible.

Without water-cooling on underfeed grates, and using Midwestern coals with preheated air, continuous operation at a burning rate of 35 lb per sq ft per hr can be maintained conservatively with most coals without undue maintenance; and with possible periods of from 2 to 4 hr of 38 lb per sq ft per hr during peak-load conditions.

However, with water-cooled grates, this range is increased under the same operating conditions to 45 lb per sq ft per hr continuous operation; and for peak periods up to 60 lb per sq ft per hr of from 4 to 6 hr duration, or longer.

This means that a much wider range of loads can be economically carried without undue maintenance, as a smaller grate area can be employed and still have a good efficiency at the lower range of loads.

Where heat recovery is other than in the form of a preheater, such as an economizer, and air at normal temperatures is used for combustion, this range can be materially increased. In the case of the Richmond, Indiana, plant, to be described later, the range

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

TABLE 1 CHARACTERISTICS OF COALS FROM MIDWESTERN FIELDS

	Moisture, per cent	Dry ash, per cent	Dry volatile, per cent	Dry fixed carbon, per cent	Dry Btu	Dry sulphur, per cent	Fusion temp, F
Ohio	3.5– 9	7.0–12	36–42	50–54	12500–13800	2.5–4	2050–2500
Western Kentucky	6.0–10	5.0– 9	35–40	50–55	13300–13800	2.0–4	2000–2250
Indiana	6.0–15	6.5–15	37–44	46–55	12400–13400	1.0–5	1900–2500
Illinois	7.0–15	7.5–15	35–40	50–57	12400–13400	1.0–5	1900–2500
Iowa	15.0–20	12.0–20	35–40	35–40	9800–10500	4.0–6	1900–2200
Kansas	2.0–12	6.0–16	34–40	45–50	10500–13800	3.0–6	1900–2300

of load is from 30,000 to 100,000 lb of steam per hr, with a burning rate for the minimum load, using Indiana Fifth Vein coal of 11,250 Btu, as received, of 14 lb per sq ft per hr. At 100,000 lb of steam per hr, the burning rate is 50 lb per sq ft per hr. The established operating efficiency of this unit for its load range has been at least 83 per cent over long periods of operation.

Without water cooling and under the same load and conditions, an increase of 25 per cent of grate area would be required to hold the burning rates down to prevent undue maintenance. With the larger grate area required, the over-all operating efficiency would be materially lowered, due to very low burning rates at the light load conditions.

Thus it can be seen that the water-cooled stoker has a distinct advantage in covering a wider range of load more efficiently by being able to employ a decreased grate area.

An actual comparison of air-cooled and water-cooled stokers has been made in the power plant of the Pittsburgh Plate Glass Company, Barberton, Ohio, using Ohio coals. The No. 8 unit with a water-cooled stoker has been in operation 2 years, and the following tabulation gives the performance of this unit over that period:

Pressure, gage, psi	825
Total steam temperature, F	750
Air temperature to stoker, F	300
Continuous capacity (maximum), lb per hr	200000
Total steaming time, hr	14513
Coal burned, tons	143568
Steaming time, per cent	82.8
Capacity factor, per cent	72.4
Full-load capacity for actual steaming time, per cent of	87.4

The average combustion rate, covering the years 1938 and 1939, for the No. 8 water-cooled unit was 46 lb per sq ft per hr, as compared with 38.3 lb per sq ft per hr for the No. 7 air-cooled unit.

A new water-cooled unit, No. 3, which has been in service since April, 1940, will obviously show improved performance over the older units. The range of coals used on this unit is as follows:

Fairmont 5/8-in. N & S, Btu	13500
Champion N & S (Pittsburgh Coal Co.), Btu	13000
Ohio No. 8 N & S, Btu	12500
Ohio No. 5 N & S (Metro), Btu	10500

No difficulty has been experienced in burning any of these coals up to a rate of 60 lb per sq ft per hr. The temperature of the preheated air is about 350 F.

It is the opinion of those responsible for the operation of this plant that "The water-cooled stoker is equally as satisfactory as the air-cooled stoker at approximately 20 per cent higher fuel-burning rate, using 350 F air preheat instead of cold air on the air-cooled unit."

Description of No. 3 Unit

No. 3 unit of the Pittsburgh Plate Glass Company at Barberton, designed for a continuous capacity of 180,000 lb of steam per hr at 860 psi gage pressure, 761 F final temperature, is a 4-drum Babcock & Wilcox bent-tube type, containing 15,015 sq ft of heating surface; a continuous-tube superheater having 5200 sq ft of surface; a continuous-loop Elesco economizer with 7700 sq ft of surface; and a tubular heater of 12,175 sq ft. All four sides of the furnace are water-cooled as follows: Front wall, 682 sq ft of plain area; rear wall, 290 sq ft of plain area, and 176 sq ft armored; side walls, 190 sq ft of plain area, with 360 sq ft armored; total water-cooling, plain area, 1162 sq ft, with 536 sq ft armored; furnace volume, 8650 cu ft.

The water-cooled, 13-retort, mechanical-drive stoker for No. 3 unit, equipped with a clinker-grinder-type dump, has an area of 404.3 sq ft, and is designed for a maximum of 350 F. The induced-draft and forced-draft fans are steam-turbine-driven. The stoker was designed for Ohio bituminous coal, having the following approximate analysis:

Moisture, per cent	2.39
Ash, per cent	11.16
Volatile, per cent	38.36
Fixed carbon, per cent	48.09
	100.00
Btu, as fired	12470
Btu, dry	12776
Sulphur, per cent	3.78
Fusion temperature, F	2200

Table 2 gives the results of tests, conducted on the No. 3 units described. The tests were made with Ohio No. 8 coal of about average quality. The ash has an initial deformation point of 2100 F, fusion 2200 F, and fluid 2300 F.

TABLE 2 DATA AND RESULTS OF ACCEPTANCE TESTS ON NO. 3 UNIT

Test number	2
Date of test	Feb. 14, 1940
Duration of test, hr	24
Steam generated, lb per hr	153556
Proximate Analysis of Coal Fired	Per cent
Moisture	3.56
Ash	11.90
Volatile matter	38.53
Fixed carbon	46.01
Heating value, Btu per lb of coal	12422
Combustible (unburned carbon) in refuse, per cent	8.46
Flue-Gas Analysis at Boiler Outlet	Per cent
CO_2	13.24
O_2	6.03
CO	0.20
N_2	80.53
CO_2 in flue gas at economizer outlet	13.01
CO_2 in flue gas at air-heater outlet	13.01
Temperatures	Deg F
Steam temperature at superheater outlet	779.8
Superheat	253.5
Air temperature entering air heater	79.5
Air temperature leaving air heater	302.8
Gas temperature leaving boiler	695.2
Gas temperature leaving economizer	573.5
Gas temperature leaving air heater	369.1
Feedwater entering economizer	222.8
Water temperature leaving economizer	288.7
Wet-bulb temperature of air entering system	53.2
Steam Pressure	
Drum pressure, psi, abs	888
Header pressure at superheater outlet, psi, abs	858
Hourly Quantities	
Coal burned, lb per hr	13458
Combustion rate, lb of coal per sq ft of grate per hr	45.7
Heat Balance	Per cent
Heat absorbed by steam and water in economizer, boiler, and superheater, including blowdown	84.45
Heat loss due to unburned combustible in refuse	1.30
Heat loss due to incomplete combustion of carbon (CO)	0.81
Heat loss due to sensible heat in dry gas	6.93
Heat loss due to evaporation of moisture in coal	0.35
Heat loss due to combustion of hydrogen in coal	4.12
Heat loss due to moisture supplied with combustion air	0.03
Radiation and unaccounted for losses	2.01
	100.00

TABLE 3 DUST WEIGHTS DETERMINED FROM TESTS CONDUCTED AT IOWA STATE COLLEGE

Date, 1939	Rating of unit, lb steam per hr	Run no.	Total weight, grains	Flue gas sampled, cu ft	325 mesh or above, per cent	Grains of 325 mesh or above	Total grains per cu ft of gas	Grains of 325 mesh or above per cu ft of gas
12–6	65000	1[a]	20.9801	337.38	23.39	3.15	0.0622	0.00818
					10.52			
12–6	65000	1[a]	39.1397	457.19	27.81	8.9259	0.0856	0.0915
					27.70			
12–7	80000	2	20.3707	140.61	38.05	7.75	0.1448	0.0551
12–7	80000	2	43.71	172.15	20.90	9.136	0.2539	0.0530
12–7	80000	3[b]	22.8896	168.35	29.88	6.84	0.1359	0.0406
12–7	80000	3[b]	58.00	264.25	41.55	24.10	0.2198	0.0912
					Average..........		0.15036	0.05659

[a] Two sets impinging bottles used during this run. Weights were added.
[b] Ashpit cleaned during this run.
NOTE: These results are with atmospheric conditions, 30 in. Hg and 60 F.
The results disclose dust loading of 1/10 of the guarantee specified.

RESTRICTION OF FLY ASH

With increasing restrictions by the smoke ordinances regarding the emission of objectionable fly ash, in addition to smoke requirements, it is necessary to set a limit in specifying the emission of fly ash from any firing equipment. On underfeed-stoker installations, without provision other than the usual means provided in boiler units for the collection of fly ash, we are specifying that the unit shall have an efficiency of collection such that not more than 0.5 grain of dust + 44 mu in size shall be present per cu ft of flue gas, measured at 70 F, and 29.92 in. Hg barometer.

On a recent water-cooled underfeed-stoker installation, fly-ash tests over a range of loads were conducted. The coal burned had the following characteristics:

Moisture, per cent	15.44
Ash, per cent	18.62
Volatile, per cent	32.68
Fixed carbon, per cent	33.26
	100.00
Btu per lb	9282
Sulphur, per cent	4.48
Fusion temperature, F	1958

It is recognized that size consist of the fuel has considerable to do with emission of fly ash, especially on some types of firing.

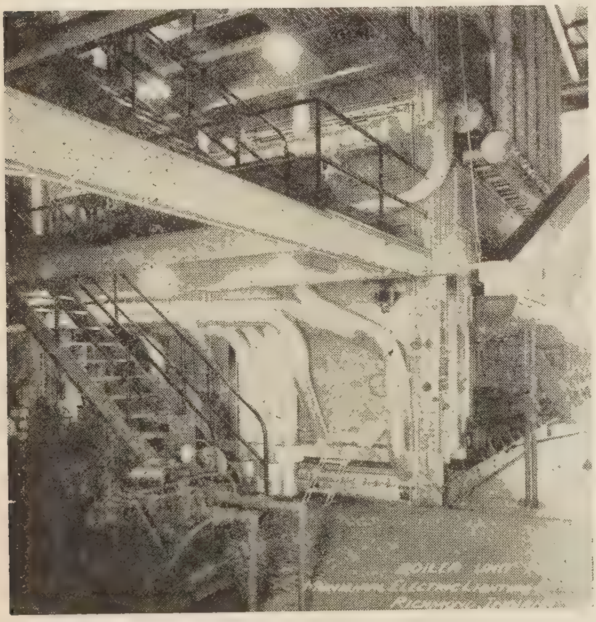

FIG. 1 BOILER UNIT NO. 4 OF THE MUNICIPAL ELECTRIC LIGHTING AND POWER PLANT, RICHMOND, INDIANA

While we cannot be definite in saying that the water-cooled underfeed stoker will emit less fly ash than the air-cooled stoker, yet, the fact remains that lower wind-box pressures prevail at the same burning rates due to a more homogeneous fuel bed, resulting in a more uniform air distribution, made possible by the cooling of the tuyères and the absence of adhering clinkers.

Through the courtesy of the Chicago Smoke Department we were permitted to use their equipment on this test. This equip-

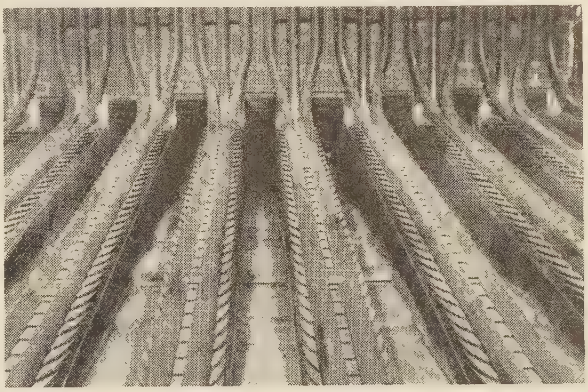

FIG. 2 INTERIOR VIEW, SHOWING SIDE WATERWALLS OF WATER-COOLED STOKER, RICHMOND PLANT

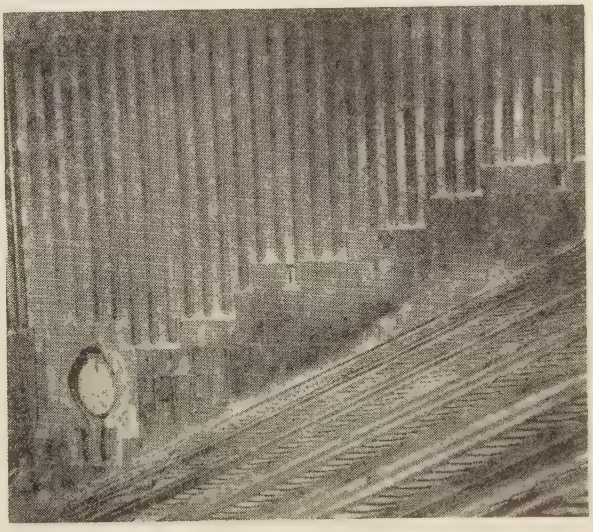

FIG. 3 ANOTHER VIEW OF THE WATER-COOLED STOKER AND THE SIDE WATERWALLS

ment has been described in a recent article.[2] The results are given in Table 3.

Description and Results of Water-Cooled Stoker Unit at Richmond, Indiana, Municipal Plant

Boiler. This Richmond unit, designed for a continuous capacity of 100,000 lb of steam per hr at 430 psi gage pressure and 825 F final temperature, is a Babcock & Wilcox 4-drum, bent-tube type, containing 10,112 sq ft of heating surface, with a pendant-type superheater having 2218 sq ft of surface, and a return-bend economizer of 3350 sq ft. All four sides of the furnace are water-cooled, as follows: Front wall, 440 sq ft; rear wall 142 sq ft with 95 sq ft armored; side walls partially water-cooled, 126 sq ft, and 306 sq ft armored; the remainder of the upper walls is refractory. The total plain surface is 708 sq ft and the total armored surface, 401 sq ft. The furnace volume is 5140 cu ft.

Hagan automatic control is installed, together with complete instrumentation of the unit.

Stoker. The stoker is a continuous-dump, 9-retort, hydraulic-drive, Taylor water-cooled unit with an area of 255 sq ft, designed for a maximum air temperature of 80 F, to burn Indiana coal. The induced-draft fan has two motors on the same fan for 2-speed control.

Overfire air from a plenum box, extending across the entire furnace width and supplied from the main mud box, is distributed above the fuel bed by a series of nozzles inserted through the front waterwall tubes.

This unit was placed in service in March, 1939. The record made during a continuous run from August 1, 1939, to January 22, 1940, when the unit was taken out of service for inspection, is given in Table 4.

TABLE 4 PERFORMANCE OF RICHMOND UNIT DURING CONTINUOUS RUN AUG. 1, 1939, TO JAN. 22, 1940

Total continuous service, hr	4184
Total steam generated, lb	256,935,500
Total coal burned, lb	30,473,400
Average evaporation, lb	8.43
Average heat value of Indiana coal (as fired), Btu	11,266
Total heat per lb of steam generated (435 lb gage; 700 F total temp), Btu	1358
Heat in feedwater to economizer above 32 F (270 minus 32), Btu	238
Btu added to steam generated in unit	1120
Combined efficiency (entire period), per cent	83.8
Average steam per hour, lb	61,500
Minimum loads as low as, lb	25,000
Maximum loads as high as, lb	120,000
Guaranteed performance at 75,000 lb of steam per hr, per cent	82.15

draft loss or efficiency. No new stoker parts were installed at this inspection, or at any previous inspection. The slagging was cleaned up, and the unit returned to service. It has been in continuous service ever since, with equal efficiency.

Six different grades of Indiana Fifth Vein screenings were used during this run, and no difficulty was experienced with any of these fuels. The characteristics of these coals are given in Table 5.

The availability factor for the period from August 1, 1939, to April 30, 1940, was 98.4 per cent; the efficiency for the entire period was 83.1 per cent.

TABLE 6 ACCEPTANCE TEST ON RICHMOND UNIT; HEAT BALANCE AT GUARANTEED POINT

	75,000 Lb per Hr	
	Btu	Per cent
Heating value of fuel, dry basis	12892.00	100.00
Heat absorbed by water in economizer	520.84	4.04
Heat absorbed by water and steam in boiler	8644.08	67.05
Heat absorbed by steam in superheater	1482.58	11.50
Heat absorbed by steam-generating unit	10647.50	82.59
Heat loss due to moisture in coal	168.89	1.31
Heat loss due to water from combustion of hydrogen	555.64	4.31
Heat loss due to moisture in air	21.92	0.17
Heat loss due to dry chimney gases	1063.59	8.25
Heat loss due to unburned gaseous combustible	....	..
Heat loss due to unconsumed combustible in refuse	198.54	1.54
Heat loss due to unconsumed hydrogen, hydrocarbons, radiation, and unaccounted for	235.92	1.83
Total, dry basis	12892.00	100.00

TABLE 7 POWER CONSUMPTION; MEASURED HORSEPOWER INPUT TO AUXILIARIES

Load, lb of steam per hr	50000	75000	100000
Forced-draft fan	29.8	32.0	38.2
Induced-draft fan	23.0	30.83	67.44
Hele-Shaw stoker drive	7.0	7.0	7.0
Total hp input	59.8	69.83	112.64
Equivalent kw	44.9	52.5	84.6
Tons coal fired per hr	2.80	4.3	5.79
Input per kw per all auxiliaries per ton of coal as fired	16.0	12.2	14.6

TABLE 8 ANALYSIS OF ATKINSON COAL

	As fired, per cent	Dry, per cent
Moisture	18.27	...
Ash	10.01	12.25
Volatile	34.12	41.75
Fixed carbon	37.60	46.00
	100.00	100.00
Sulphur	2.87	3.50
Btu per lb	10053	12300
Fusion temperature F	1930 F	

TABLE 5 CHARACTERISTICS OF FIFTH VEIN INDIANA 1½-IN. SCREENINGS

Name of coal	Hercules	Knox	Patoka	Elberfeld	Panhandle	Enos
Moisture, per cent	11.41	8.66	9.91	8.75	10.15	10.18
Ash, per cent	10.72	11.67	8.95	10.78	9.94	10.18
Btu	11873	11493	11950	11440	11205	11462
Sulphur, per cent	2.90	4.56	3.59	3.50	2.79	3.98
Fusion temperature of ash, F	2200	2076	2200	2210	2230	2110

The commercial performance of this unit over the entire operating period approximates very closely the guaranteed and predicted performance set up in the contract.

A careful inspection of the unit at the conclusion of this run, after having burned 23,000 tons of coal since March, 1939, revealed no indication of any burning of either tuyères or tuyère supports in any of the nine retorts. The kicker bars and overfeed mechanism were practically in their original condition. Some indication of overheating was noted on the steplike ends of the bars. The dump plates, with the exception of two regions about 4 in. diam where they were burned, were in perfect condition.

The slagging on the first bank of boiler tubes, which is protected by a slag screen, was not enough to interfere with the

[2] "Low-Cost Flue-Dust Sampler," *Power*, April, 1940, pp. 70–72.

All of the auxiliaries are electrically driven. The induced-draft fan has a low- and high-speed motor.

The 50,000- and 75,000-lb[3] per hr loads on the unit were carried by the low-speed motor; and the effect of the high-speed motor on the 100,000-lb load is reflected in the higher consumption of current per ton of coal fed to the stoker.

Reports From Other Plants Using Midwestern Coals

Municipal Light & Power Plant, Fremont, Neb. This plant has a water-cooled crusher-roll-type stoker, and is designed to generate 75,000 lb of steam per hr at 400 psi gage, and 750 F total steam temperature.

[3] As this unit was designed for maximum over-all economy at 75,000 lb, it is interesting to note that the actual results reflected this economy at the desired point.

The unit is burning Kansas coals originating in the Pittsburgh district. Although these coals carry a relatively high Btu, from 12,400 to 12,700 the fusion temperature of the ash consistently ranges between 2000 and 2100 F.

This unit recently completed a continuous run of 237 days, or 5670 hours. During that period the combined efficiency was 83.4 per cent when burning the Kansas coals of approximately 2000 F ash-fusion point.

Municipal Light Plant, Greenwood, Mississippi. This plant has a 6-retort, 41-tuyère, water-cooled C.A.D. stoker, having the same retort design as now used at Richmond, Ind. Coal originating in Webster County, Kentucky, has been used. The author has been advised that 11,000 tons of coal have been burned without replacement of a single part.

Iowa Electric Light & Power Company, Cedar Rapids, Iowa. This plant has probably burned more Midwestern coal on water-cooled underfeed stokers than any plant in the country with results reported as follows:

"Regarding the underfeed stokers referred to, four of these stokers were started on Atkinson coal in 1934, and four in 1935. This report will show the total amount of coal burned from the time the stokers were first started on Atkinson coal until the first of the year 1940. The total coal on the eight stokers is 574,772 tons.

"In 1935, we installed one stoker of the crusher type, water-cooled; and until January, 1940, this stoker burned 82,450 tons. The total cost on this particular stoker is 2.226 cents per ton of coal burned.

"The balance of the stokers had considerable work done on them last year, which was all on the outside of the stokers, such as ram boxes, crankshafts, crankshaft bearings, and gears. We did not feel that the total amount of this expenditure should be charged against the Atkinson coal, as these stokers were installed at least 10 years previous.

"On these stokers we have tabulated the amount of the expense on the inner parts of the stokers, such as pushers, tuyères, tuyère supports, and other items on the inside of the firebox. This cost is 4.758 cents per ton, made up as follows: Pushers, 1.108 cents; tuyères, 1.827 cents; remainder of stokers, such as side frames, dump plates, and other internal parts, 1.823 cents.

"Some of this cost also includes experimental work with the water-cooling system, as we made about four changes before we actually achieved the desired results.

"The average coal-burning rate on the fuel-burning portion of the stoker is 42 lb per hr; we have run on a week's test as high as 60 lb per hr.

"In regard to the date we first started experimenting with water-cooling, this was the early part of 1933."

Libbey-Owens-Ford Glass Company, Rossford, Ohio. "The Libbey-Owens-Ford Glass Company had three bent-tube boilers of 12,500 sq ft area at the Rossford, Ohio, plant. These boilers were equipped with water-cooled side and rear walls, and multiple-retort underfeed stokers.

"In order to widen the range of fuels that could be burned, it was decided in 1937, to modify these three stokers to include water-cooled grates. As a result of these changes, we have widened the range of coals that can be burned to advantage, and thereby placed ourselves in a better position to purchase any coals.

"We have burned on these stokers, since water-cooling them, over 75,000 tons of coal. At times we have operated only one boiler, and at other times three boilers, according to the steam demand.

"The ratings normally vary from 75 to 225 per cent; the average rating is approximately 175 per cent. The average over-all operating efficiency is better than 80 per cent.

"The stokers installed under boilers *B* and *C* have just been overhauled at a cost of less than $300 each for material and labor. The stoker installed under *A* boiler will be repaired at a later time.

"Approximately 50 per cent of the coal burned was Ohio No. 8 fusing at 2100 F. The remaining 50 per cent was West Virginia and Kentucky coal containing ash which fuses from 2400 to 2600 F. The temperature of the preheated air varies from 280 to 310 F. We have not had to take a boiler off the line in 2 years as a result of stoker trouble."

Advantages of Water-Cooled Grates on Midwestern Coals

Some of the advantages of water-cooled-stoker installations burning Midwestern coal, as compared to air-cooled stokers using the same fuel, have been noted as follows:

1 There is more uniform flow of coal, with a consequent ease of distribution.

2 A more homogeneous fuel bed has been noted than with straight air-cooled stokers; and better air-distribution results.

3 The size of clinkers, with proper burning rates for coal, has not been objectionable, and they have been easily discharged automatically under the bridge wall with continuous-dump design.

4 There is every indication that the maintenance will be low.

5 Indications are that the power consumption is lower.

6 The range of fuels with low ash-fusion characteristic is greater; and the flexibility of fuels is greater.

7 Test efficiencies are more closely approximated in regular plant operation than on the air-cooled stoker because of the ease in obtaining high CO_2 in water-cooled units.

8 The availability of the unit is materially increased.

9 With crusher-type water-cooled stokers less artificial cooling of the ash is required, and, consequently, less moisture put into the flue gases. The quick chilling of molten ash particles assures a continuous and even flow of fuel.

10 The water-cooled stoker provides an additional factor of safety against careless operation.

11 A distinct purchasing advantage is obtained in that fusion temperature and sulphur content are not limiting factors, as is often the case without water-cooling.

12 Greater latitude of operation.

Discussion

T. C. Cheasley.[4] There certainly can be no doubt as to the care exercised by the author in securing data to present in this paper. Also, I think all of us will agree, the statements he has made are conservative and will stand close scrutiny.

Unfortunately, there has been comparatively little opportunity for most of us to see water-cooled tuyères in operation and we must, therefore, draw on our imaginations a good deal to visualize the results reported.

Certain it is that the burning of most Midwestern coals has presented problems to the designers of furnaces and boilers as well as to operating engineers unless, and except, the burning rates have been held at fairly low maximums.

It is generally conceded that the washing of Midwestern coals has done much to improve operation and to reduce stoker and furnace maintenance, but it is the opinion of some of us that the average coal in the average plant is not being given the opportunity to produce its highest combustion efficiency.

[4] Fuel Engineer, Sinclair Coal Company, Kansas City, Mo. Mem. A.S.M.E.

By this is meant the frequent departure from fundamentals in speaking of stoker operation, and also in actual operation. We speak of the "pounds of coal burned per square foot of grate area" and, of course, there is no such operation as burning coal in an underfeed stoker, i.e., if the stoker is being properly operated. A stoker does not burn coal, it converts coal to coke, which is burned in progression. The fact that we talk and think in "pounds of coal burned" is, in the writer's opinion, at least in part responsible for some of the high maintenance costs mentioned. At the same time this condition causes some rather good coals to become discredited.

Extremely high excess air "at the point of volatile release," is probably largely responsible for this, and the "average" CO_2, shown at the usual point of taking gas samples, does not tell the whole story. Undoubtedly, the water cooling of tuyère areas, which are exposed to high temperatures, will reduce the build up of temperature in the stoker parts, and thus should decrease maintenance costs by prolonged life of these parts. However, as the writer views the situation, the chief benefits, as stated by the author, should be the opening up of greater selections of coal with the advantage in many cases of very much lower delivered prices and lower steam costs.

F. S. Scott.[5] Definite and marked improvements have been made in underfeed stokers in recent years. It will satisfactorily meet the present more exacting requirements of steam-generating plants.

This paper describes the water-cooled stoker and its contribution to the art. The idea of water cooling a stoker is not new. It is an advancement of the art. Its development has been retarded by the physical difficulties which are reflected in the cost.

The Westinghouse Electric & Manufacturing Company has developed the underfeed stoker along different lines from water cooling. Our development of the link-grate stoker has made a definite improvement in the ability of the underfeed stoker to handle Midwestern coals. High efficiency, low maintenance, and good reliability are obtained.

This link-grate underfeed-type stoker was first installed in 1928. At first it was only applied with a clinker-grinder-type stoker. The continuous-ash-discharge type without a clinker grinder was first installed in 1931. At present there are more than 150 continuous-ash-discharge link-grate stokers and over 85 link-grate clinker-grinder stokers in operation, using all types of bituminous coals, from those mined in Iowa to the Eastern low-volatile or Pocahontas-type coals.

There are 72 link-grate stokers in operation on Midwestern coals with a base-load capacity in excess of 5,000,000 lb of steam per hr. The peak-load capacity is considerably above this figure.

Burning Rates

The author expresses the opinion that 35 lb of coal per sq ft per hr of projected grate area is conservative for continuous operation with air-cooled underfeed stokers using Midwestern coals. The writer's experience indicates that, if such rates are applied to link-grate underfeed stokers, it unnecessarily increases the first cost and causes difficulties at low loads.

It is very important that serious consideration be given to burning rates on these stokers. The difficulties can be as great with a stoker that is too large as with one that is too small. The maximum burning rates on underfeed stokers are determined by the ability of the fuel bed to stay on the grates. High air velocities cause the coal to "blow" or lift from the fuel bed, thereby limiting the maximum burning rates. The velocity of the air is determined by the quantity flowing through the fixed area of tuyère openings. The quantity of air flowing, when the excess air is constant (or CO_2), is in turn determined by the "heat duty" or the combustible burned. Expressing "heat duty" in other terms, it is the Btu released from the coal per square foot of projected stoker area per hour.

For many years we have based our engineering on Btu release per unit area with due allowance for the agglutinating quality of coal which is a measure of its resistance to being blown away. Without exception, the predicted combustion rates have been readily maintained.

There are two parts which require cooling when a stoker is burning coal, i.e., the grates and the ash adjacent to the grates. If the ash near the grates is cooled well below the "sticky" state and the grates are also cool, there will be no difficulty with the movement of the fuel bed and no burned stoker parts. It is a known fact that high velocities of fluids flowing past solids cause better heat transfer than low velocities. Therefore, the writer believes it to be logical to state that high air velocities through the stoker tuyères, but within the practical limits, will cool both the grates and the adjacent ash better than very low velocities. This fact has been demonstrated in practice. Where stokers have been operated at very low rates or on "bank" for long periods of time, the fuel bed becomes loaded with large clinkers, the stoker iron can be observed "red hot," and parts are burned. These large clinkers are then difficult to move when the load is increased. This indicates some of the dangers inherent in selecting a stoker that is too large.

Stokers can also be selected which are too small. The lack of induced draft or insufficient "black" surface exposed to the fire in the furnace is the more frequent cause of difficulties with fuel beds when the load on the boiler has been increased. It is indicated that burning rates should not be considered in pounds of coal burned per sq ft per hr because a stoker is primarily a machine for converting the potential heat in coal to sensible heat in gases. Burning rates should be measured by the heat duty or the Btu release per square foot per hour. We are interested in materials handling only in so far as it affects the efficient production of heat for use by the heat-exchanging apparatus.

It is thought to be generally considered that 40 lb of coal per sq ft per hr of projected grate area is conservative practice for continuous loads when using Eastern coal of 14,000 Btu per lb as received. This is 560,000 Btu heat release per sq ft per hr of projected area. If the same rate in pounds per square foot is used for Midwestern coals of 11,000 Btu per lb as for Eastern coals, the heat release would be 440,000 Btu per sq ft per hr or a reduction of 21.5 per cent in the heat-release duty per square foot of projected stoker area. If the same rate of heat-release duty is used for Midwestern coals, as for Eastern coals of 560,000 Btu per sq ft per hr, the rate in pounds of 11,000-Btu coal would be about 51 lb per sq ft per hr. This shows an increase in material passed over the stoker of about 27.5 per cent with no increase in the heat-release duty.

The nature of the constituents of the ash in Midwestern coals and their behavior when subjected to furnace conditions indicate that the heat duty should be reduced on stokers using them as against Eastern coals. Long experience indicates that 21.5 per cent reduction in heat duty is considerably greater than is necessary or economical.

A 10.7 per cent reduction in the heat-duty rate of 560,000 Btu per sq ft per hr to 500,000 Btu per sq ft per hr of projected stoker-grate area is conservative. Even though the heat-duty rate has been dropped to 500,000 Btu per sq ft per hr, the amount of material passed over this stoker area is increased from 40 lb per sq ft per hr to 45.5 lb per sq ft per hr. These figures are quite conservative in that there is a considerable number of stokers which are operating satisfactorily at base loads above this point.

[5] Stoker Engineer, N. W. District, Westinghouse Electric & Manufacturing Company, Chicago, Ill.

This brief discussion leads to the conclusion that the basic and most important item in determining the desirable burning rates for stokers is the Btu heat release per square foot per hour of projected grate area.

It has been found that, with Indiana, Illinois, and similar coals, air-cooled link-grate underfeed stokers operate best, throughout their load range, with a continuous maximum burning rate of 500,000 to 550,000 Btu heat release per square foot per hour of projected area. If the coal has 11,000 Btu per lb, this is a rate in pounds of coal material passed over the stokers of 45.5 to 50 lb per sq ft of projected stoker area.

Loads of from 4 to 6 hours in duration can be satisfactorily carried up to 825,000 Btu per square foot per hour or 75 lb of 11,000-Btu coal per sq ft per hr.

Our experience with many stokers indicates that the link-grate underfeed stoker can conservatively burn more coal per square foot per hour, with all the advantages named by the author, than he feels can be justified with the special water-cooled stoker. To demonstrate the foregoing point the following examples are given:

One installation of two 7-retort link-grate continuous-ash-discharge stokers in Iowa burns Iowa coal of 8500 Btu per lb to 10,500 Btu per lb. The stokers having 178 sq ft of projected grate area have been in operation more than 2 years. During the last year the maintenance has been less than 2 cents per ton of coal burned. The normal continuous load with 10,500-Btu coal is 40 lb per sq ft per hr or 420,000 Btu heat release per sq ft of projected area. The same normal continuous steam output is carried when this plant burns 8500-Btu coal. The rate with 8500-Btu coal is 49.5 lb of coal per sq ft or a heat release of 420,000 Btu per sq ft of projected area per hr.

No difficulty has been experienced up to the limit of the induced-draft fans or 20 per cent above these figures. The efficiencies being obtained are approximately the same as those guaranteed.

Another set of installations consisting of five identical stokers in three different plants burns various kinds and grades of Indiana and Illinois coals of approximately 12,000 Btu per lb as received. The normal continuous load carried on these stokers of 153 sq ft of projected area is about 46 lb per sq ft or 550,000 Btu heat release per sq ft per hr. The induced-draft fans limit the maximum load to a rate of approximately 59 lb per sq ft per hr or 710,000 Btu per hr of projected stoker area.

These units have been installed more than 2 years. The maintenance for these stokers has been less than 1 cent per ton of coal burned.

Efficiency

Another point the writer would like to comment upon briefly is that of boiler-unit efficiency. A comparison of boiler-unit efficiencies of units carrying different pressures and of different designs means but little when the discussion concerns firing equipment only. The efficiency of firing equipment, or the combustion efficiency when the coal is the same, depends upon three factors which may be varied (1) the excess air in the products of combustion in the furnace, as measured by the per cent CO_2, (2) the combustible loss to the ashpit, (3) the soot and cinder loss. The greatest of these losses is that in the dry gases or that due to excess air, as measured by the per cent CO_2 in the products of combustion.

The author has shown that both the ashpit loss and the soot and cinder loss are low with underfeed-stoker firing. He has shown in one place the percentage of CO_2 at the boiler outlet. This is for the water-cooled stoker at Barberton, Ohio—it is 13.24 per cent.

In the first example previously mentioned in this discussion, the CO_2 averages 13.5 to 14 per cent at the boiler outlet, in daily operation.

The second set of installations using Indiana and Illinois coals averages 14 to 15 per cent CO_2 at the boiler outlet, in daily operation.

A modern boiler unit, fired by a link-grate underfeed stoker, will produce in excess of 14 per cent CO_2 at the boiler outlet without stack smoke and no use of secondary air at continuous burning rates up to 550,000 Btu per sq ft per hr of projected stoker area.

Advantages Claimed for Link-Grate Stokers

The author indicates that the water-cooled underfeed stoker shows certain advantages over similar air-cooled stokers. It is claimed that the link-grate underfeed stoker shows advantages over other types of underfeed stokers burning Midwestern coals as follows:

1 More uniform flow of coal and, therefore, more even distillation of gases and burning of coke.

2 Absence of large clinkers at the rear of the stoker makes it possible to burn out more of the combustible before the refuse enters the ashpit.

3 Maintenance is low.

4 The range of fuels is greater so that the ash-fusion point of the coal is relatively unimportant. It is possible to handle coal satisfactorily with an ash content of 25 per cent or more.

5 Tests efficiencies can be closely approximated in regular operations.

6 Reliability is increased.

7 High continuous capacities are easily obtained.

R. L. Swinney.[6] The author mentions the trend toward one-boiler-unit operation in a single plant. This trend is the result of the increasing number of steam-generating units which are demonstrating their ability to stay in continuous service over long periods of time.

The unit at the Richmond, Indiana, municipal plant is mentioned as having been in continuous service for a period of 175 days or approximately 6 months.

There are many other similar records of units burning pulverized coal, oil, and gas, some of these plants having 100 per cent feedwater make-up.

Progress in the art of feedwater treatment and improvements in the design of water-cooled furnaces and fuel-burning equipment have made these records possible, but much credit is due the operators of plants where such records have been established. It is predicted that properly designed steam-generating units will be more generally accepted as having dependability for continuous service approaching that of turbines.

A. W. Thorson.[7] The author is to be commended for bringing the literature up to date on the subject of his paper. It is gratifying to know that numerous installations of this type of equipment are performing satisfactorily as indicated by the wealth of operating data included in the paper.

The text and part of the discussion have specified maximum combustion rates either in pounds or Btu per square foot per hour. The writer would suggest that the safe maximum burning rate depends not only upon the heating value of the fuel but to a considerable extent upon the ash content and the fusion characteristics of the ash.

Does the use of the water-cooled stoker bring about any reduction in investment costs? It appears that it might, inasmuch, as steam-generating capacity is increased by the stoker-cooling tubes

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and the cost of the stoker itself is probably not much higher; and also because the maximum possible combustion rate appears to be higher with water cooling than without for the same fuel.

Author's Closure

Mr. Scott's discussion has introduced some interesting information concerning the developments of non-water-cooled stokers. It is the author's opinion, however, that the information furnished by Mr. Scott is largely the commercial viewpoint of one manufacturer.

As brought out in the evidence presented in the paper, the experience of users is that claims for the link-grate-stoker performance, when burning Midwestern coals, are not as consistently maintained as where the water-cooled grate is employed, particularly at the higher coal-burning rates when using preheated air. Therefore, it is logical to conclude, from the experience of users, that water cooling will afford more consistent and reliable protection for the grate than when the sole grate-cooling medium is air. The flow of water through the cooling tubes is not affected by fuel-bed thickness, porosity, or characteristics of the coal; while uniform and desired air flow through the grate is influenced by these factors.

The matter of burning rates mentioned by the discusser is one of proper application to the particular coal under consideration, and it is doubtful whether a fixed rate of heat release can be used to cover the characteristics of all Midwestern coals. The rates set up by the author were determined from long experience in connection with the average coals of the Midwest.

The burning characteristics of Midwestern coals are too greatly diversified to establish any definite percentage of reduction in heat duty when burning these coals, and it is a matter of careful decision on the part of the engineer in setting grate areas.

The Fuel-Bed Tests at Hell Gate Generating Station, 1937-1938

By M. A. Mayers,[1] W. H. Dargan,[2] Joseph Gershberg,[3] B. C. Dalway,[4] M. J. Williams,[5] and E. R. Kaiser[6]

This paper is a final report on fuel-bed tests at Hell Gate Generating Station and includes some of the conclusions reached in the authors' study of the data up to this time. The fuel bed of a multiple-retort underfeed stoker is shown to consist of vertical strata parallel to the center lines of the retorts and tuyère stacks in which coking and ignition processes occur along the entire length of the stoker. The actual burning of the carbonized fuel occurs largely by overfeed action in the burning lanes where the temperature and pressure gradients are determined by the rate of primary-air flow, and the gases, rising from the burning lanes, may require the addition of secondary air for complete combustion. It is shown that control of the contour of the fire in operating practice may affect not only the amount of coke blown from the fire and the degree of gas stratification in the boiler passes, but also the severity of treatment to which the stoker iron is subjected.

Introduction

This report gives the results of measurements of the temperatures, gas analyses, and the air and gas pressures in the fuel bed of a typical large commercial underfeed stoker. The work continued the studies of the Coal Research Laboratory of the Carnegie Institute of Technology on the nature of combustion processes, and arose from a desire to apply the results of previous work to the fuel beds which exist in commercial equipment. The Consolidated Edison Company of New York expressed its interest in such measurements and offered its cooperation, if a practicable program could be worked out. The Coal Research Laboratory invited the participation in the project of the staff of Bituminous Coal Research, Inc., at Battelle Memorial Institute, because of their interest and experience in the investigation of combustion problems. The research organizations submitted a tentative program on March 17, 1937, which called for the investigation of the following main variables: (*a*) The rates of flow of coal and air; (*b*) temperatures; and (*c*) gas compositions at various points in the fuel bed. The temperatures and gas pressures were to be observed, and gas samples for analysis drawn by the use of probes of refractory material inserted into the fuel bed from the wind box through holes in the stoker iron. The influence on the three variables of the following operating factors was to be determined: the rate of burning, the amount of excess air, the contour of the fuel bed, the size and size consist of the coal, and the kind of coal. It was estimated that the studies would require at least 6 months if only a few coals were studied.

The tentative program was approved by the Consolidated Edison Company on April 9, 1937. Hell Gate Generating Station was chosen for the tests because the existence of an isolated bunker at that station would facilitate the use of different coals during the tests. It became apparent on trial that it would be unsafe for men to work in the stoker wind box principally because of the danger of serious injury in the event of a furnace-wall tube failure. Inspection of the stokers installed in the station showed that only on the seventh-row stokers was it possible to insert probes into the retorts from below without interference with the links operating the secondary rams. Hence, an isolated working place from which the probes could be handled was built into the wind box of boiler No. 73 while it was out of service for routine maintenance in July, permitting preliminary tests of the probes during the two-week period, August 19 to September 4.

These tests showed that the isolated working place, later known as the doghouse, provided adequate safety and convenience for the test crew, and that the routine of the fuel-bed tests did not interfere with operation of the boiler. They also showed that it was feasible to insert probes into the fuel bed from below, but indicated that certain changes in their design were desirable. New probes designed in accordance with these indications were made and testing was started November 1. By January 1, 1938, the equipment had been tried out and a single series of measurements at a low burning rate had been finished. These data were embodied in a first progress report distributed to the parties to the investigation at the end of February.

On January 1, 1938, Bituminous Coal Research, Inc., withdrew from cooperation in the project, because the funds it had allotted to the work were exhausted. The program was continued by the Consolidated Edison Company and the Coal Research Laboratory with the assistance of the Pittsburgh Coal Company. Three additional series of tests were made in which the load, the percentage of excess air, and the fire contour were varied separately, but in all of which the same coal, one normally used in the station, was fired. In these series, traverses were made at three sections in the stoker but, as examination of the data indicated the action of the stoker to be much the same at all sections from the neck to the pit, measurements in the last three series, in which special coals were fired, were made at only one section in the center of the stoker. Active testing was finished by April 29, 1938, and the motion pictures of fuel-bed behavior made as a part of this project were completed during the week ending May 14.

The results of the last six series of tests were submitted to the sponsors in a second progress report, distributed in October, 1938. A report embodying an outline of the results and tentative conclusions subject to revision was presented orally before the annual meeting of this Society in December, 1938. In addition to these reports, the apparatus used for the fuel-bed motion pictures has been described in a paper by Markson and Dargan (22)[7] and the

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[5] Pittsburgh Coal Company of Wisconsin, Duluth, Minn.; formerly Pittsburgh Coal Company, Pittsburgh, Pa.

[6] Fuels Division, Battelle Memorial Institute, Columbus, Ohio. Mem. A.S.M.E.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., December 5–9, 1938, of The American Society of Mechanical Engineers.

[7] Numbers in parentheses refer to the Bibliography at the end of the paper.

method of correcting temperatures has been reported by Mayers (25).

Conclusion and Summary

This work represents an initial survey of the conditions under which combustion takes place on multiple-retort underfeed stokers. The data are not as precise or as detailed as might be desired, they apply specifically only to the particular piece of equipment on which the tests were made, and they do not cover a wide range of fuels. It is desirable that further tests of this nature be made on other stokers of different construction, burning many more of the hundreds of coals available for underfeed-stoker firing in this country. Additional tests could be made with much less expenditure of time and money and the experience gained in those reported herewith would be of great value in developing equipment by the use of which more accurate and more detailed information could be obtained. More data on the extremely interesting region in the walls of the burning lane should be secured by inserting probes along diagonals passing from the retort into the burning lane. This was not attempted in these tests because the importance of this region was not appreciated.

Despite its limitations, this investigation is not without immediately applicable results. It shows, for instance, that a change in the contour of the fire may have beneficial results in the uniformity of operation of the stoker, and that, in this installation, the use of the "short" fire leads to less severe treatment of the iron of the stoker. It also shows that what apparently is a thinner fire may actually have more fuel over the tuyères and so be more stable than a thicker fire, with its accompanying narrow burning lanes; and finally, it repeats the demonstration that the gases rising from the active portions of the bed may require additional air for combustion. Another result of importance in the design of stokers is the elucidation of the structure of the fuel bed obtained by these tests, which showed that the mechanism of heat flow into the coal in the retort is similar to that governing heat flow into a slot-type by-product coke oven. This makes available for the use of stoker designers a considerable amount of information concerning heat transfer which is directly applicable to their problem.

The principal facts brought out by this investigation may be summarized as follows:

1 The fuel beds of multiple-retort underfeed stokers exhibit a considerable degree of uniformity of structure relative to the axes of the burning lanes, the positions of which, however, are not fixed with respect to the center lines of the tuyère stacks, but may vary within limits. Fluctuations of temperatures and pressures from the averages at points fixed with respect to these axes are relatively small, but the deviations of gas composition are large, apparently because of the slow speed of diffusion by comparison with the rates of heat transfer and of the transmission of hydrostatic pressure in the system.

2 The fuel beds consist of burning lanes through which most of the air for combustion passes, separated by heaps of coal in the retorts through which very little air flows.

3 The coal in the retorts is retained between walls of coke, at the cooler boundary of which rapid carbonization of the coal occurs and through which heat for carbonization and ignition of the coal is conducted transversely across the stoker from the burning lanes.

4 The fuel has a component of flow transversely across the stoker from the retorts to the burning lanes approximately equal to that calculated from the burning rate and the dimensions of the fuel bed.

5 At the bottom of the burning lanes an extremely high-temperature coke is burned and gasified.

6 The temperature and pressure gradients in the main primary-air stream are determined by the rate of air flow, but the temperatures attained just above the tuyères depend also upon the fire contour.

7 The maximum temperatures observed, 2800 to 3000 F, appear close to the top of the bed, and are nearly the same in all series, regardless of the load, the fuel, or the fire contour.

Results of Tests

A summarized tabulation of the data on the conditions of operation in all the runs, averaged for each series, is given in Table 1. Series 1 and series 2 differ only in load; series 2 and 3 in excess air; and series 3 and 4 mainly in fire contour (refer to paragraph on "Stoker Operation"), although there was an increase in excess air in series 4 over series 3. Series 5, 6, and 7 were run at the same load and excess air as series 2 but with three different coals; series 5 with coal from Pocahontas No. 3 seam; series 6 with

TABLE 1 SUMMARY OF DATA ON CONDITIONS OF OPERATION

Series No.	1	2	3	4	5	6	7
Runs No.	21–44	45–63	64–71, 76	72–75 77–83	84–88	89–93	94–97
Dates	11/23/37 –12/17/37	1/19/38 –3/9/38	3/17/38 –3/24/38	3/25/38 –4/ 7/38	4/14/38 –4/15/38	4/20/38 –4/21/38	4/26/38
Coal	Lower Kittaning, run-of-mine				Pocahontas	Sized, Lower Kittaning	Mallory Gas
Bailey-meter readings, average:							
Steam	149	217	209	206	207	209	207
Air	152	219	236	238	211	211	215
Fire contour	Long	Long	Long	Short	Short	Medium	Long
Average temperatures, F:							
Steam, superheater outlet	618	652	644	662	651	655	660
Feedwater entering boiler	287	292	327	322	315	309	309
Flue gas, boiler outlet	550	619	609	607	597	608	597
Pressure and draft, in. water:							
Wind-box pressure	1.5	2.9	2.7	2.9	3.0	2.7	2.5
Furnace draft	0.12	0.15	0.30	0.21	0.12	0.14	0.11
Boiler-outlet draft	0.30	0.94	1.18	0.97	0.66	0.68	0.67
Excess air, per cent:							
Bailey-meter setting, pens together	37	44	37	49	42	40	37
Actual	40	45	54	72	45	41	42
Hourly rates, 1000 lb per hr:							
Steam flow	82.7	120.3	115.8	112.0	114.5	115.6	114.5
Coal burning (estimated)	8.0	12.2	11.8	11.5	11.3	11.9	11.5
Air flow (estimated)	116	185.5	190.3	208.5	176.4	174.0	173.5
Unit rates, lb per sq ft per hr:	Coal Air	Coal Air	Coal Air	Coal Air	Coal Air	Coal Air	Coal Air
(a)	20.9	31.9	30.8	30.0	29.4	31.0	30.0
(b)	26.2 380	39.9 606	38.5 621	36.6 682	36.5 576	38.9 569	37.6 567
(c)	44.0 640	67.0 1020	64.9 1045	61.5 1145	61.5 970	65.4 957	63.2 954

(a) Referred to total projected area, 382 sq ft.
(b) Referred to actual area, omitting ashpit, 306 sq ft.
(c) Referred to air-admission area, 182 sq ft.

TABLE 2 COAL ANALYSES

Series No.	1	2	3	4	5	6	7
Coal	Eureka, run-of-mine				Pocahontas, nut and slack	Eureka, sized	Mallory
Supplier	Berwind-White Coal Mining Company				Pocahontas Fuel Company, Inc.	Berwind-White Company	Hunt's Point Gas Plant
Source:							
County and state	Cambria, Pa.				West Virginia	Cambria, Pa.	Logan, W. Va.
Seam	Lower Kittaning				Pocahontas No. 3	Lower Kittaning	Eagle
Proximate analysis, per cent:							
Moisture, as received	3.6	3.3	3.3	3.2	4.3	3.3	4.5
Volatile matter, dry	16.4	17.1	16.7	16.6	17.8	17.0	32.5
Fixed carbon, dry	77.1	75.5	75.2	76.4	76.1	75.2	63.2
Ash, dry	6.5	7.4	8.1	7.0	6.1	7.8	4.3
Sulphur, dry	1.13	1.43	1.46	1.38	0.60	1.36	0.57
Heating value:							
As received	14100	14080	13970	14100	14170	13940	14080
Swelling index in V.M. det., per cent	...	...	...	85	95	90	65
Ash-fusion temperatures, F:							
Initial	2560	2450	2610	2510	2550	2630	2650
Softening	2620	2510	2670	2570	2600	2680	2700
Size, per cent:							
On 1 1/2-in. round hole	0.4	1.1	0.8		0.4	0.9	0.4
Through 1 1/2-in. on 1-in.	1.6	3.4	1.6		2.9	3.5	2.6
Through 1-in. on 3/4-in.	2.5	0.7	2.3		3.5	3.1	3.4
Through 3/4-in. on 1/2-in.	..	6.7	5.9		5.4	7.2	8.0
Through 1/2-in. on 3/8-in.	10.8[a]	..	6.2		5.4	8.1	8.1
Through 3/8-in. on No. 4 square mesh	11.1	8.5[b]	9.9		10.4	16.6	17.1
Through No. 4 on No. 8	..	47.9[c]	17.9		18.1	24.0	18.5
Through No. 8 on No. 14	..	14.8[e]	18.9		18.4	16.2	13.8
Through No. 14 on No. 20	44.5[d]	5.1[f]	7.5		7.4	5.2	5.8
Through No. 20 on No. 60	17.1[g]	6.2[h]	19.1		19.4	10.7	11.7
Through No. 60 on No. 100	5.5[j]	1.6	1.4		2.5	0.3	1.8
Through No. 100	6.5	4.0	8.5		6.2	4.2	8.8

[a] Through 3/4-in. on 3/8-in.
[b] Through 1/2-in. on 1/4-in. square mesh.
[c] Through 1/4-in. square on 1/8-in. square.
[d] Through No. 4 on No. 20.
[e] Through 1/8-in. square on No. 20.
[f] Through No. 20 on No. 30.
[g] Through No. 20 on No. 50.
[h] Through No. 30 on No. 60.
[j] Through No. 50 on No. 100.

Lower Kittaning coal, the same as that used in the first four series, except that it had been sized at the washing plant before shipment by the removal of fines passing a 5/32-in. screen; and series 7 with a high-volatile-gas coal from the Hunt's Point Gas Plant of the Consolidated Edison Company. Reference to Table 2 and Fig. 1 shows that the "sized" Lower Kittaning coal used in series 6 had been so degraded during shipment that its size consist was not significantly different from the nut and slack ordinarily used. Thus, this series cannot be expected to show any effect which might be produced by a change in the sizing of the coal fired. No trouble was experienced due to smoke emission in series 7 using high-volatile coal. The flame in the furnace in this series was very dense and hugged the bed very closely so that it was difficult to see the fuel bed through it, but no secondary combustion was observed at the top of the first pass at any time with either a long or a short fire.

The values of steam flow and air flow in Table 1 are averages of the readings recorded on the boiler-room floor at 15-min intervals but, since the load was held very steady during the tests, slight error is expected on this account. Only the total steam readings were averaged; the readings of the meter on the west-side superheater outlet (refer to paragraph on "Test Apparatus") were generally 10 to 20 points lower during series 1 to 3, but differed very little from the totalizing meter in subsequent series. This is believed to be due to the somewhat more severe stratification which existed during the tests with the long fire than in those with short fires. Superheated-steam temperatures at the two outlets differed by amounts up to 30 F, the east side usually being higher. This behavior was irregular but appeared somewhat less frequently with short than with long fires. In calculating the average steam temperatures, a straight arithmetic average of all the readings in each series was taken, as the weighted average differed from this by only 1 to 2 F. The flue-gas temperature leaving the boiler is not accurate, as it was based on the reading of the Bailey-meter gas thermometer which was not checked.

Of the quantities given under the heading "Unit Rates," the first fuel-burning rate, referred to the projected area of the stoker and ashpit, 382 sq ft, is the quantity usually cited in comparisons of stokers. The others are based on measurements of the various elements of the stoker, taken by the test crew, which resulted in

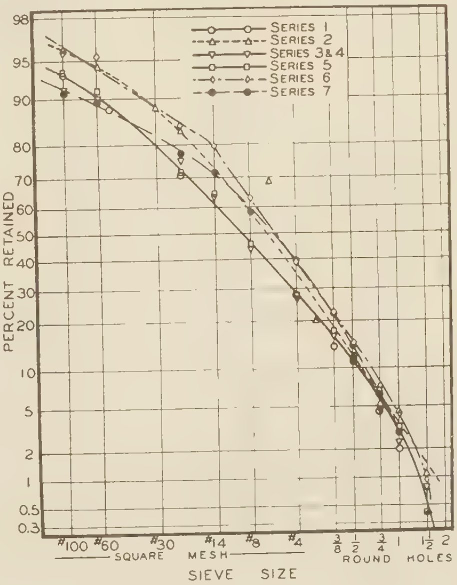

FIG. 1 SIZE DISTRIBUTION OF COALS PLOTTED ON LOG PROBABILITY COORDINATES

the following areas: Total stoker area, omitting ashpit, 306 sq ft; projected area of stoker, omitting ashpit, 274 sq ft; and total air-admission area, including area of tuyère stacks, projected area of side-wall tuyères, and extension grate, 182 sq ft. Fuel-burning and air-flow rates, based on these areas, are useful in the consideration of the significance of the measurements.

As the results of the tests include data on temperatures, pressures, and gas compositions at each of ten or more levels in a total of 79 test runs, it is obviously out of the question to present the complete experimental data in this paper. For this reason, only samples of the tabulated results will be given here; complete tabulations of the data have been prepared and deposited in the archives of the Society and may also be obtained, either as photo-

TABLE 3 TEST RESULTS; RUN NO. 72, NO. 2 TUYÈRE, EAST

Distance above tuyère, in.	Temp, F	Corrected temperatures, F	Gas-pressure differential, in. water	Gas analysis, per cent CO_2	O_2	CO	H_2	CH_4
—1	385	...	2.05	...	...	...	...	...
—1/2	450	...	1.0	..	...	...	...	...
0	450	...	0.44	...	...	...	...	...
1/2	1055	1910	0.44	9.8	10.2	3.5	1.4	...
1	1400	2030	0.43	3.9	16.0	1.3	...	...
1 1/2	1900	2080	0.47	7.6	12.8	...	...	...
2	2295	2500	0.55	16.1	4.6	...	...	...
2 1/2	2490	...	0.60	16.0	3.2	...	...	...
3	2505	...	0.59	11.9	0.4	9.5	2.4	1.5
3 1/2	2475	...	0.90	14.6	0.4	7.6	...	...
4	2535	...	0.80	11.2	0.0	7.0	4.2	0.9
4 1/2	2550	...	0.90	11.9	4.2	3.5	0.6	0.8
5	2580	...	1.1	11.9	0.2	7.6	...	...
6	2655	...	1.25	12.2	0.5	7.2	1.3	0.5
7	2655	...	1.25	12.4	0.6	5.5	0.8	0.4
8	2550	...	1.55	9.2	0.8	8.4	8.8	1.8
9	2750	...	2.1	11.6	0.8	6.8	1.1	0.9
10	2700	...	2.9	13.9	5.7	0.0	...	...
11	2780	...	3.0	...	...	...	...	...
12	2685	...	2.7	...	...	...	...	...
12[c]	2445[a]	...	3.25	16.0	0.1	2.6	...	...
13	2475[b]	...	2.9	15.4	1.9	0.9	...	...
14	2625	...	2.8	17.9	0.0	1.6	0.1	0.1

[a] Momentary temperatures as high as 3025 F.
[b] Probe seen in hole about 2 in. below top of burning lane.
[c] Taken during withdrawal of probe.

bed being built up between parts (refer to paragraph on "Testing Procedure").

Parentheses around corrected temperatures indicate that these readings are unreliable because the observed rate of temperature rise was too great.

Discussion and Interpretation

Structure of the Fuel Bed. In order to facilitate visualizing the results of the tests, Figs. 2 to 6 have been prepared, showing, on cross sections of the fuel bed, contours of equal temperature, constant gas composition, and equal pressure drop. It must be borne in mind that these figures do not represent either averages or a steady state; they are, rather, the authors' best guess, based on plots of the observations, as to what might be observed, if it were possible to take simultaneous readings throughout the region tested at some instant. Fig. 2 shows isothermals on transverse cross sections of the bed at the three positions tested, taken in the planes, differing slightly from the vertical (refer to Fig. 13), in which the probes were advanced through the bed, at a load of 120,000 lb of steam per hr with 45 per cent excess air and a long fire (series 2). The sections show a portion of the bed, consisting of one full retort and tuyère stack and one half of the next adjacent elements; the lines of insertion of the probes are indicated by dot-and-dash lines, with scales showing the distance above the stoker iron in inches. Position No. 1 is near the head of the stoker, while No. 3 is near the extension grate (refer to paragraph on "Equipment and Construction"). The contours are nearly vertical at the sides of the retorts and the temperature increases very rapidly across a narrow region above the walls of the retorts in the direction of approach to the center line of the tuyère stack. The coal in the retort does not rise above room temperature until it approaches the walls of the retort or a thin skin on top, although the width and height of the unheated heap of coal become less as the distance from the head end of the stoker increases. The region of very high temperatures above the tuyères is coincident with the burning lanes which are so prominent a feature of stoker fires. It is evident that the principal direction of heat flow into raw coal normal to the contours is horizontally across the stoker from each burning lane toward the center line of the adjacent retort. It should also be noted that coal in the retort, at the middle of the stoker longitudinally, may rise 23 in. above the floor of the

TABLE 4 TEST RESULTS; RUN NO. 61, NO. 2 RETORT, WEST

Distance above ram, in.	Time after stopping stoker, min	Maximum temperature observed, F	Time after reaching position, min	Corrected temperature, F	Gas-pressure differential, in. water	Gas analysis, per cent CO_2	O_2	CO	H_2	CH
0	..	95	...	..	2.9	..	...	...	..	..
5	..	95	...	..	2.2	..	...	...	..	..
10	..	95	...	..	2.2	..	...	...	..	..
15	..	95	...	..	2.5	..	...	...	..	..
17.5	..	95	...	..	2.6	..	...	...	..	..
18.5	..	95	...	..	2.6	..	...	..	..	.
19.5	..	95	...	..	2.65	..	...	...	..	.
20.5	..	95	...	..	2.65	..	...	...	..	.
21.5	..	95	...	..	2.7	..	...	...	..	..
22.5	..	95	...	..	2.8	..	...	...	..	..
23.5	..	95	...	..	2.8	..	...	...	..	..
24.5	7.0	570	1.0	1250	2.8	5.2	6.2	4.2	10.2	12.2
..	..	..	1.5	1460	...	..	...	...	..	..
25.5	10.0	1195	1.0	1860	2.9	5.2	5.0	5.7	12.7	10.7
..	..	..	2.0	1990	...	..	...	...	..	..
Part 2										
23.5	2.5	250	...	..	3.1	3.6	7.9	...	..	..
24.5	5.5	875	1.0	1500	2.95	5.0	2.5	5.2	14.6	14.4
..	..	..	2.0	1665	...	..	...	...	..	..
25.5	8.5	1835	1.0	(2390)	2.8	5.3	2.0	6.8	17.6	15.1
..	..	..	2.0	(2615)	...	..	...	...	..	..
26.5	11.0	2220	Steady		2.8	8.4	0.9	6.6	..	..
27.5	..	2280	Steady		2.85	14.1	4.1	0.0	1.3	1.2
28.5[a]	..	2325	Steady		3.1	10.1	0.6	7.1	12.3	6.5

[a] Probe 3 in. above fuel bed.

prints or as microfilm, from The American Documentation Institute, Washington, D. C. Details of the procedure and discussions of the accuracy and precision of the results are given in later paragraphs. The results obtained are exemplified by Tables 3 and 4.

The data from a typical run over the tuyère stack are given in Table 3, in which the first column gives the depth of penetration into the fuel bed; the second the average temperature if a steady reading was reached, otherwise, the maximum observed; column three, the corrected temperature, if a steady reading was not attained, when the correction referred to in the section on "Supplementary Data" was applied; the fourth column shows the pressure drop of the combustion gases through the stoker and fuel bed to that point; and columns 5 to 9 give the percentage composition, dry, of the gas sample taken at that point. In Table 4, which presents the data from a typical retort run, the first column gives the distance above the secondary ram; the second, the time after stopping the stoker; and the third, the uncorrected temperature at that time; while the fourth and fifth columns give the time and the corrected temperature at intervals during the period the probe was held at that point. The sixth column gives the pressure differential, and the remaining columns the analysis of the dry gas. The retort runs were completed in two parts, the fuel

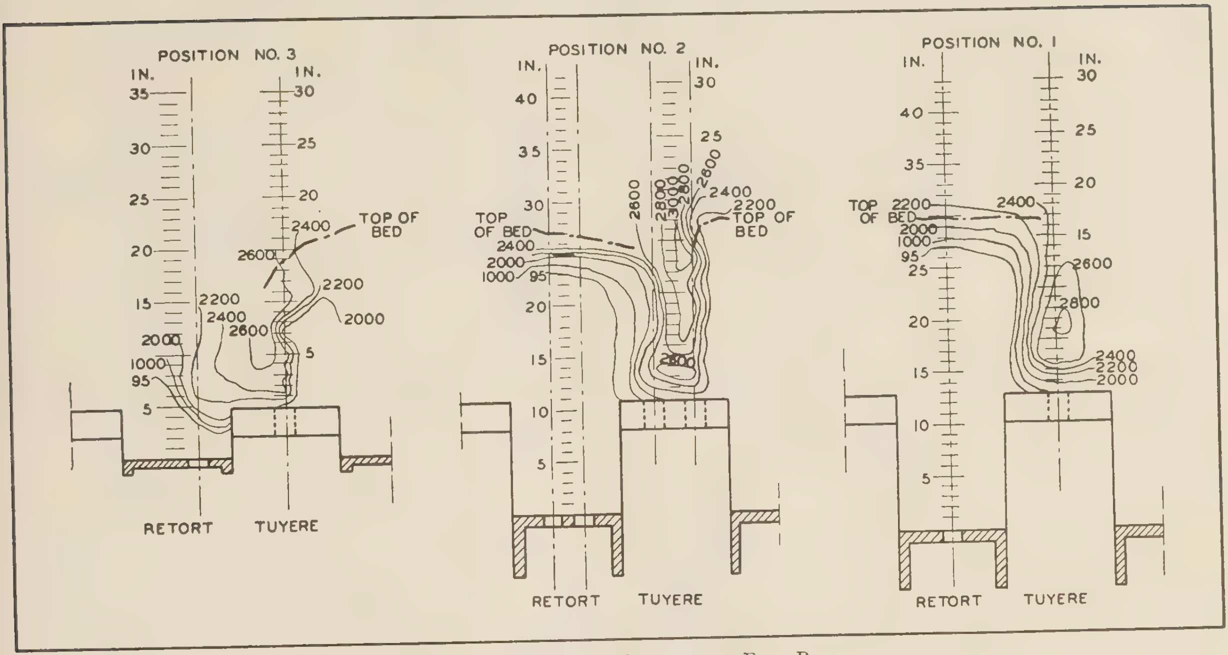

Fig. 2 Temperature Contours in Fuel Bed

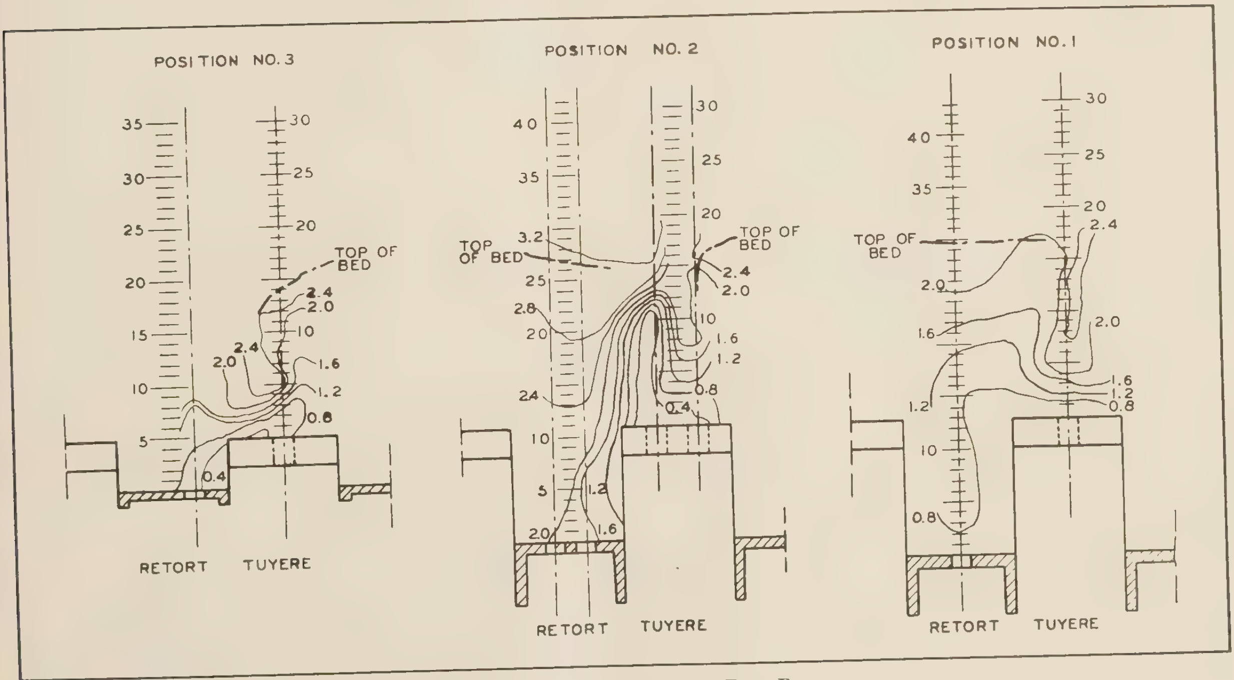

Fig. 3 Pressure Contours in Fuel Bed

retort, or 12 in. above the level of the tuyères, without being heated above room temperature, even though at the same level less than 12 in. away, across the stoker, the temperature in the burning lane may have reached 3000 F.

Fig. 3 shows contours of equal drop in gas pressure on the same cross sections and under the same conditions of firing. These contours suggest that the principal flow of air is upward along the burning lanes and that there is comparatively little flow through the retort. In fact, the nearly vertical course of the contours in the walls of the burning lanes indicates that these walls may be almost completely impervious to air flow. That there is comparatively slight air flow up through the retorts is shown by the low rate of pressure drop through this rather densely packed heap.

Fig. 4 shows lines of constant oxygen, and Fig. 5 lines of constant carbon monoxide and of constant hydrogen-plus-methane concentrations. The air in the retort is not consumed until it approaches the region of very rapid temperature rise shown in

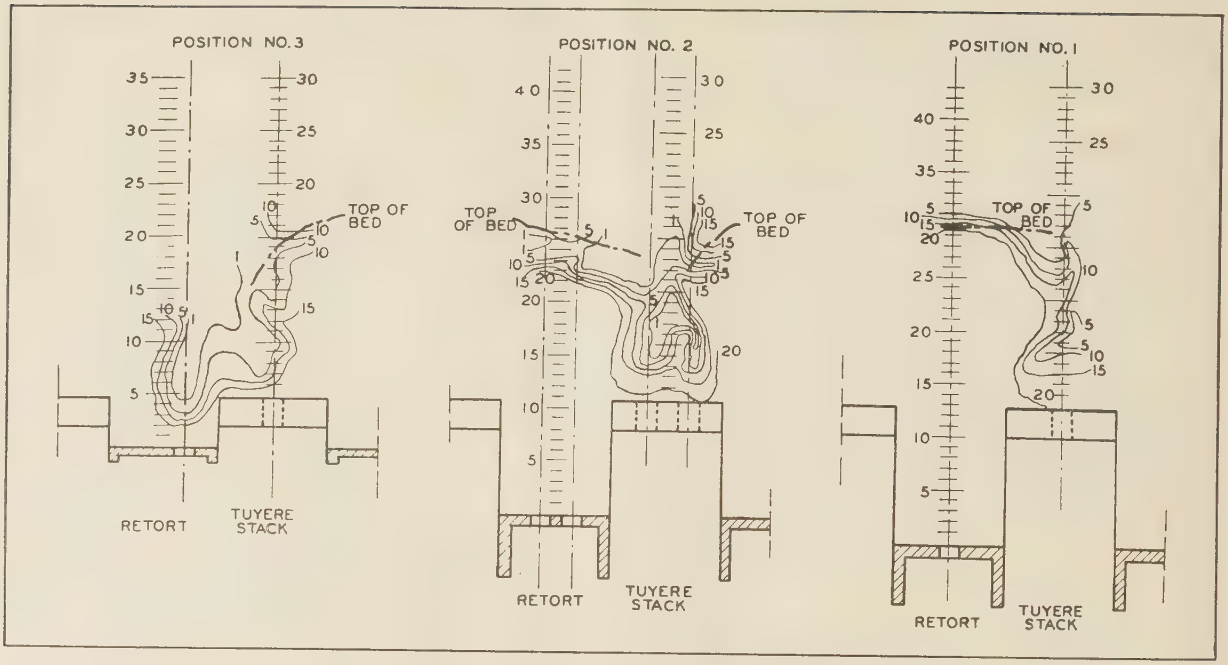

Fig. 4 Contours of Equal Oxygen Concentration

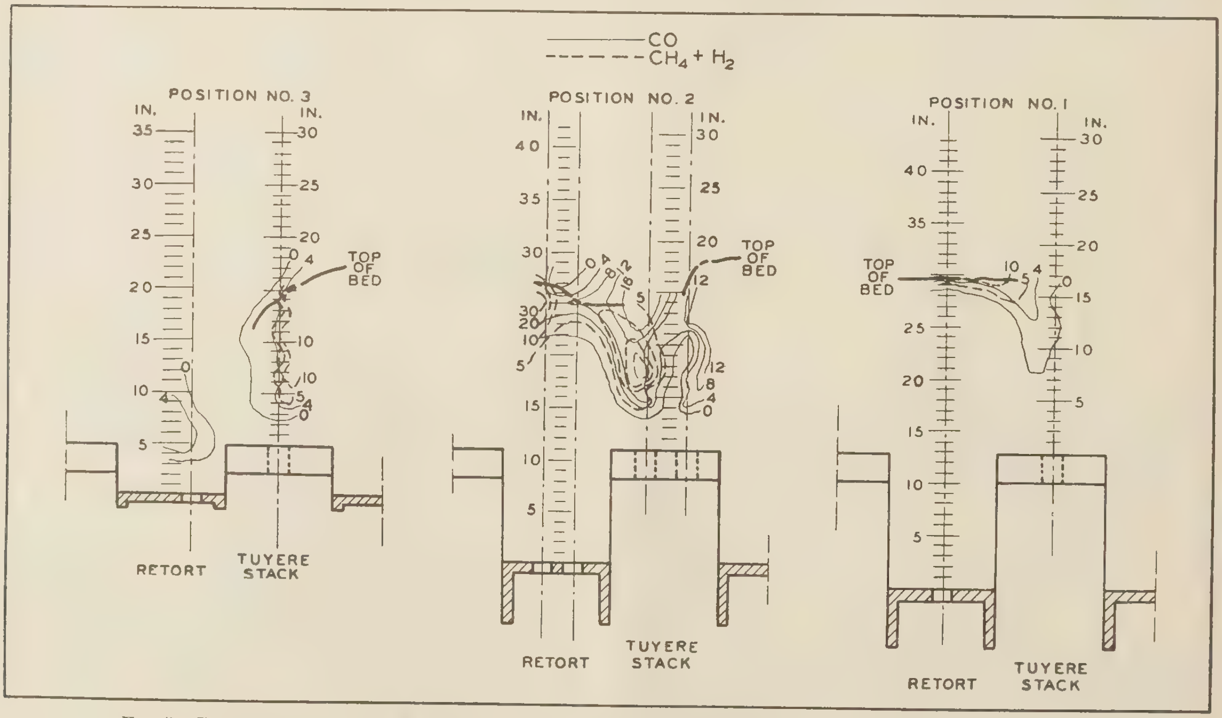

Fig. 5 Contours of Constant Concentrations of Carbon Monoxide and of Other Combustible Gases

Fig. 2. In the burning lane, the oxygen penetrates further into the bed, the closer the center line of the lane is approached; along the walls of the lane it decreases practically to zero at only 3 in. above the tuyères while, at the center of the lane, the air may still contain 5 per cent of oxygen 6 in. above the tuyères. This result accords with the data on the pressure drop, which indicated that the gas velocity varies greatly in a section across the burning lane, from very small values at the walls of the lane to very large values at the center. It is also to be noted that the oxygen concentration appears to increase again toward the top of the lane. This effect is quite marked, and is accompanied by a decrease in the concentration of the combustible constituents, as shown in Fig. 5, and usually by an increase in carbon dioxide, although the latter may be masked by excessive dilution by the fresh air entering at

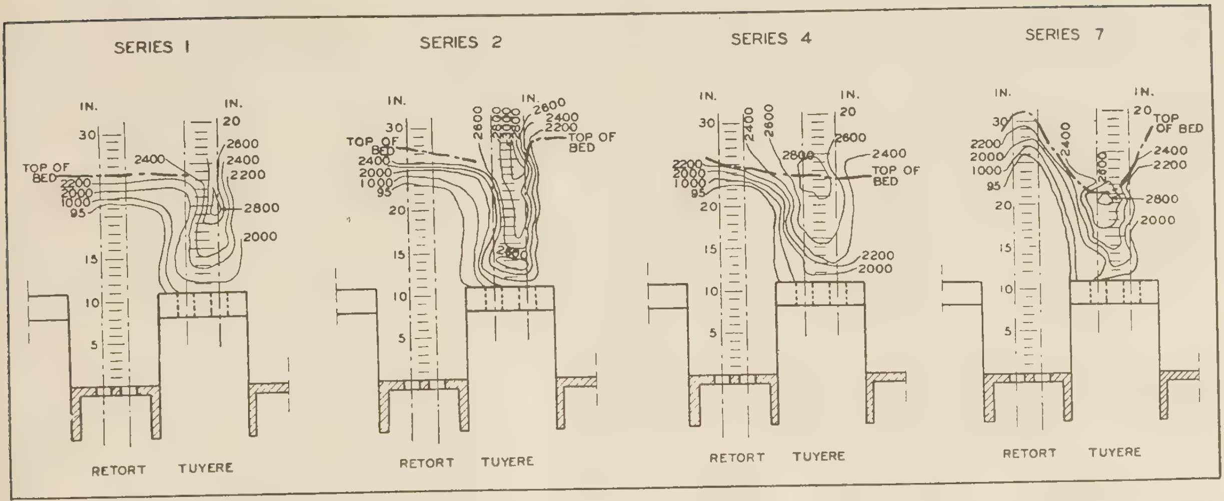

Fig. 6 Temperature Contours for Different Coals and Conditions of Operation

this point. The authors have been unable to agree on an interpretation of this secondary increase in oxygen.

The greatest concentrations of combustible gases appear in the walls of the burning lanes at about the same level as the maximum temperatures in the lanes themselves. In this region, the sum of methane plus hydrogen may reach values above 25 per cent. It can be seen by reference to Fig. 2, that the temperature in this region ranges from 1500 to 2200 F, and from Fig. 3, that it is just this region which appears to be most nearly impervious to gas flow. All these facts point to the conclusion that this is the region in which carbonization of the fuel is occurring at the most rapid rate.

Comparison of charts similar to those just referred to for different parts of the bed, and for different conditions of operation, shows quantitative differences but no qualitative change in the pattern, as indicated in Fig. 6, the isothermals at No. 2 position for a number of different conditions of operation. All these sections show the same pattern of nearly vertical isotherms in the walls of the burning lanes connecting regions low in the burning lanes with regions high in the retorts. Indeed, almost the same maximum temperatures at almost the same levels were observed in all the runs. Plots of the other variables investigated show similar uniformity under different conditions and so are not presented here.

These charts indicate that the structure of the fuel beds of underfeed stokers differs materially from the conventional picture, which consists of a bed of horizontal strata, the bottom one being green coal and the top one being freely burning coke with all gradations occurring in between. These results show that, actually, the stratification is mainly along vertical planes running from the head of the stoker to the end of the underfeed section along the walls of the burning lanes. A section through the fuel bed perpendicular to these planes, that is, across the stoker, shows green coal in the center of the retort extending almost to the top of the bed, then coal being heated, coked, and ignited as the burning lane is approached, with free burning of prepared fuel in the burning lane approximately along the center line of the tuyère stack. The flow of heat proceeds mainly in a horizontal direction, the coke layer forming first as a thin skin on the wall of the burning lane at the head end of the stoker and progressing across the retort toward its center line with increasing distance from the front wall. Thus the retort of such a stoker is similar to a by-

TABLE 5 AVERAGE TEMPERATURES AT NO. 2 TUYÈRE

Series	1		2		3		4
Distance above tuyères, in.	Temp, F		Temp, F		Temp, F		Temp, F
0	1830		..		1660		..
0.5	1920		2190		1885		2030
1.0	1960		2325		1715		1850
1.5	2165		2420		2280		2075
2.0	2050		2475		2261		2065
2.5	2030		2570		2310		2155
3.0	2140		2620		2465		2410
σ	231		203				
	a	b	a	b	(a)	(b)[2]	
3.5	2225	2140	2600	2545	2490	2325	2465
4.0	2305	2105	2620	2385	2730	2400	2480
4.5	2390	2120	2670	2320	2690	2340	2550
5.0	2445	2175	2725	2250	2700		2585
5.5	2430	2235	2695	2235		2265	
6.0	2460	2120	2780	2265		2235	2650
6.5	2505	2060	2810	2090			
7.0	2565	2090	2770	2200		2295	2700
7.5	2590	2020	2880	2255			
8.0	2620	2020		2250		2340	2650
8.5	2730	1960		2310			
9.0	2815	1850		2380		2685	2695
9.5	2825	1740		2415			
10.0	2820	1695					2755
10.5		1695					
11.0		1710					2795
11.5		1725					
12							2755
13							2680
14							2700
σ	133	122	41	195	249[1]		167[3]

[1] σ calculated for series 3 and 3a from 0.5 in. to 5 in.
[2] One run only; σ not calculated.
[3] σ calculated for series 4 from 0.5 in. to 14 in.

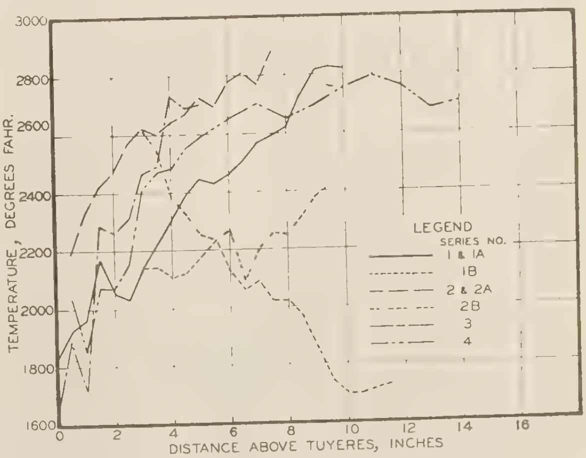

Fig. 7 Average Temperatures at No. 2 Tuyère Position

product coke oven except that the confining walls are absent and combustion takes place in direct contact with the outside layer of coke; the laws (6) which govern heat flow in a by-product oven may be expected to govern the flow of heat into a stoker retort.

Carbonized fuel breaks off from the coke walls because of the formation of shrinkage cracks and the agitating action of the secondary rams, and falls down into the burning lanes where the level of the fuel is considerably lower than it is in the retorts. These lanes may cover only a portion of the width of the tuyère stack; they consist of channels with more or less irregular coke walls, not necessarily continuous along the length of the stoker; and they generally have a more or less densely packed continuous fuel bed with, perhaps, some ash at the bottom, in the lower part of the channel. The primary air flows up through these lanes, burning and gasifying the bed of coke through which it passes. The air velocity varies within wide limits in the burning lane, depending upon the density of the fuel bed through which each filament of flow passes, and the distance required to consume the oxygen in any filament increases with the local flow velocity.

While this description is based upon measurements made on only one stoker using but a limited number of coals, it is probable that, qualitatively at least, it applies quite generally to all stokers of this type. It is evident that the cases will be quite rare in which the air flow from the tuyère stacks spreads with equal intensity through all sections of the coal, both above the tuyères and in the retort; only in those cases, however, will there be any marked deviation from the structure outlined, in which the stratification is along vertical planes and the heat flow into the raw coal is horizontal. It appears also that the high combustion rates, compared to the rates of ignition observed in pure underfeed burning, which may be attained in multiple-retort stokers, are due to the fact that there is no air flow through the planes of ignition to remove the heat conducted into this region from the hotter parts of the fire, and to the possibility that the aggregate of all the zones of ignition, that is, the sum of the areas of the burning-lane walls may be considerably greater than the projected area of the stoker.

Temperatures. The average temperatures at No. 2 tuyère position for the first four series of runs are given in Table 5 and in Fig. 7. The precision (1) of the measurements within each series is indicated by the magnitudes of the standard deviations σ given at the bottom of each column in Table 5.

These standard deviations indicate that the procedure of averaging the runs by series is probably justified, since the probable error, equal for large numbers of observations to 0.6745 σ, is never greater than about 170 F, approximately the order of accuracy which was expected before the tests were made, and is usually much less. It will be observed that the precision is often greater than the accuracy of the measurements (2), estimated as being about 90 F (refer to section on "Supplementary Data"). Table 6, the average differences between various pairs of groups of runs, shows that these differences are statistically significant (11), excepting for the upper portion of the traverses in series 2*a* and 3, since the probability that such differences could occur by chance is always less than 0.05, and usually less than 0.01.

These data indicate that the magnitude of the temperature gradient immediately above the tuyères is proportional to the air-flow rate, as shown in Table 7. The data of tests with special coals are not extensive enough to justify statistical analysis but they indicate that this conclusion holds true also for series 5 to 7 as well. On the other hand, the temperatures attained at 1/2 in. and 3 in. above the tuyères depend not only upon this gradient, but also upon the condition of the fire, as can be seen by comparison of the data for series 2, 3, and 4, all of which have nearly the same air-flow rate. In series 2, which was run at an excess air of 45 per cent with a long fire, the temperature rose to about 2600 F at 3 in. above the tuyères, while the temperature 1/2 in. above the tuyères was about 2200 F. In series 3, with a somewhat greater air flow and correspondingly greater gradient, the temperature 3 in. above the tuyères was only 2460 F, showing the effect of an increase of excess air to 54 per cent and the correspondingly more open fire. In series 4, however, in which the short fire was used, the temperature 3 in. above the tuyères dropped still further to 2400 F.

In the upper part of the bed, it is necessary to differentiate between traverses which passed up through the burning lanes, denoted by the suffix *a* in Tables 5 and 6, and those which were in the walls of the lane, denoted by the suffix *b*. Considering first the *a* runs, it will be observed that the maximum temperature attained in all of those with the long fire was about the same. In series 1, this maximum of above 2800 F was reached at about 9 in. above the tuyères, while in series 2 and 3 the same value was reached somewhat lower, at about 6 in. above the tuyères, even though series 3 was run with high excess air. Data for distances above 5 in. are not given in Table 5 for series 3*a*, as not enough runs were available to justify statistical analysis. In series 4, however, with the short fire, the temperatures were substantially lower throughout the upper part of the bed, and the maximum of just under 2800 F was reached only at 11 in. above the tuyères. The data of series 5 and 6 in which short fires were carried with special coals confirm this trend.

At the top of the bed, when the probe rose into the gas space, the temperature indication fell off quite sharply. This is shown in the averages for series 4, the only one in which a large enough

TABLE 6 SIGNIFICANCE OF DIFFERENCES BETWEEN TEMPERATURES IN DIFFERENT SERIES

Series	Range, in.	Average difference of means	*P*	
2-1	0.5–3	389	<0.01	Significant
2*a*-1*a*	3.5–16	122	<0.05	Significant
2*b*-1*b*	3.5–11.5	320	<0.01	Significant
4-2*a*	0.5–9	224	<0.01	Significant
3-2*a*	0.5–3	263	<0.01	Significant
3-2*a*	3.5–9	15	<0.6	Not significant
3-1*a*	1.5–5	273	<0.01	Significant
4-1*a*	2–8	148	<0.01	Significant

TABLE 7 TEMPERATURES AND MAXIMUM TEMPERATURE GRADIENTS NEAR THE TUYÈRES AT NO. 2 POSITION

Series	Air rate through tuyères, lb per sq ft per hr	Temperatures at 1/2 in., F	Temperatures at 3 in., F	Maximum temperature gradient, deg per in.	Ratio, Temperature gradient / Air-flow rate
1	640	1915	2140	195	0.304
2	1020	2185	2620	235	0.230
3	1045	1885	2465	395	0.378
4	1145	2030	2410	350	0.306
Average					0.304 ± 0.023

TABLE 8 TEMPERATURE DROP AT TOP OF BURNING LANE
(Part of doghouse log; run No. 94; No. 2 tuyère, west)

Time, p.m.	Position, in.	Temperature mv	Temperature F	Bed differential, in. water	Sample No.	Notes
9:05	27	14.2		1.45		
06		14.8	2625		49	
07	28	15.2		1.45		
08		14.9	2640		50	
09	29	15.2		2.35		Probe pushed hard
10		15.4				
11		15.4	2715		51	
12	30	15.7		2.70		
13		15.9	2795		52	
14	31	16.0	2810	2.68		
15		14.9	2640		53	
16	32	13.8		3.00		
17		13.1	2370		54	
18		13.1				
19	33	13.3		2.68		
20		13.8	2475		55	
21	34	14.4		2.70		
22		14.2				
23		14.4	2565		56	
24	35	14.5		2.75		
25		14.5	2580		1	Probe reported 6 in. out of fuel in burning lane

number of probes penetrated the bed to justify carrying the averages that far. This effect was observed, however, in every individual run in which the probe was seen from the rear door of the furnace. A good example is given by run No. 94, series 7, of which the readings for the last 8 in. are given in Table 8. The observation from the boiler-room floor that the probe extended about 6 in. above the level of the fuel bed in the burning lane at level 35 shows that the beginning of the precipitous drop in temperature at level 31 was close to the top of the bed.

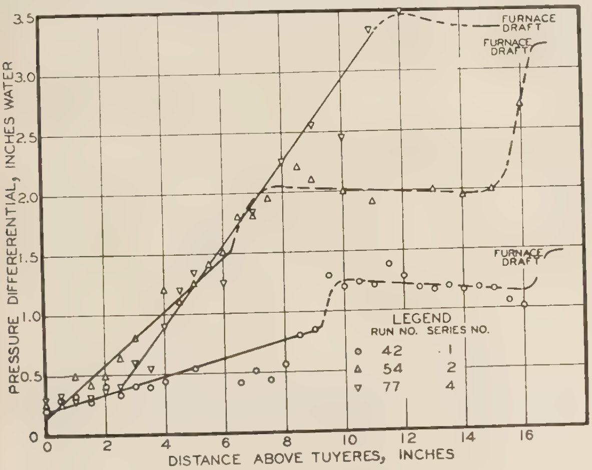

Fig. 8 Pressure Loss as a Function of Distance Above Tuyères

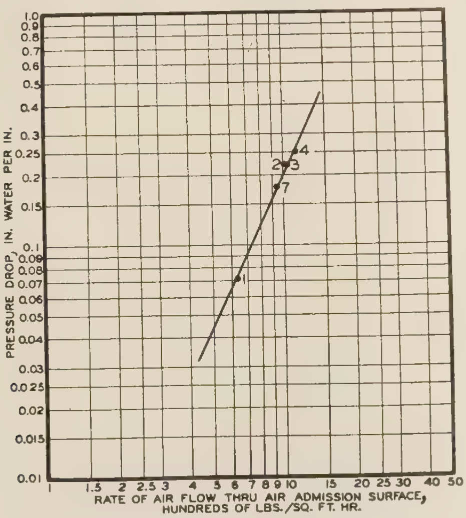

Fig. 9 Pressure Gradient as a Function of Air-Flow Rate

In runs in which the probe was raised, not through the burning lane, but through its walls, the temperature dropped from a maximum in the neighborhood of 3 in. above the tuyères, up to which point it had followed the same course as in runs in the lane, and fell off continuously to as low as 1700 F at 10 in. above the tuyères. The negative gradient in this region appeared to be independent of the load and excess air, while the length of the region of decreasing temperature was quite unpredictable and may have depended only upon the distance from the line of traverse to the center of the burning lane. The fact that these traverses differed from those just described, in being off the burning lane, was substantiated by the observations of the other quantities, gas pressure and analysis, and also by direct observation in many runs of the position of the probe when it had penetrated the fuel bed and was visible from the boiler-room floor. In every case, the region of declining temperature was terminated by an abrupt transition to a region in which temperatures of the same magnitude as those prevailing in the burning lane at these levels were observed. This transition was accompanied by characteristic changes in gas analysis and pressure (refer to section "Pressure Differential") and was often coupled with a momentary great increase in the resistance to motion of the probe through the bed, followed by complete freedom of motion. This phenomenon was interpreted as meaning that at this point the probe broke through the wall and entered the burning lane or emerged from the top of the bed.

Pressure Differential. Plotting pressure differential against distance above the tuyères leads to quite different results for runs in the burning lanes and those out of the lanes. The results of the runs in the burning lanes are readily interpreted, as shown by the examples, from runs at No. 2 position, Fig. 8. The pressure drop is linearly related to the distance above the tuyères beyond a small distance of the order of 1 to $1^1/_2$ in. and extending to 8 to 12 in. above the tuyères. While the absolute values of pressure drop to any point do not agree very well in different tests under the same conditions, the values of the pressure gradients do. These are plotted, to logarithmic scales, against the air-flow rates for the different series in Fig. 9, where it will be observed they fall on a reasonably straight line with a slope of 2.17. The values of air-flow rate used in this calculation are based on the indication of the air-flow pen of the boiler meter, and are referred to the total air-admission surface of the stoker. This calculation is admittedly a rather unsatisfactory approximation; the errors in it arise from two main sources (*a*) the air-flow meter, which is not a precision instrument at best, measures the air flow through the boiler, not that through the stoker, (*b*) there is no assurance that the air flow through different sections of the air-admission surface of the stoker is uniform. No correction has been made for air admitted through the front-wall secondary-air ports.

In spite of these qualifications, the data show that the relation given in Fig. 9 is significant. The result may be compared with the formula given by Diepschlag (8) for the pressure loss through beds of spheres. To obtain the pressure drop observed, the fuel bed would have to be equivalent to a bed of spheres about $^1/_4$ in. diam. Diepschlag's measurements, however, were made using cold air. Applying the corrections for temperature indicated by Carman's analysis (7) the size of the particles would be increased to about 0.63 in. to produce the pressure gradient of 0.196 in. water per in. of fuel-bed depth at 1000 lb air per sq ft per hr, given by Fig. 9. The slope of the line in Fig. 9 is greater than that found by Diepschlag and by Arbatsky (3), in tests using cold air, who obtained values of the exponent of 1.6 to 1.7. This discrepancy, as well as the small particle size indicated by the calculations, may be ascribed to the errors previously noted, but it might also be due to the difference in temperatures since, in a test on a chain-grate stoker burning anthracite, Arbatsky observed a value of the exponent of 2.1 for small values of air-flow rate.

Certain other significant conclusions may be drawn from these data. The linear portion of the curves in Fig. 8 for runs in series 1 to 3 are always terminated by a sharp break upward, followed by a practically flat portion, which is not however at a pressure drop corresponding to the furnace draft but from 0.3 to 0.5 in. water less. It has been shown by Hirst (15) in a discussion of the principles of coal cleaning by pneumatic tables that the maximum pressure gradient which can be attained in a porous bed of broken solids is that equivalent to the weight of the bed per unit thickness. Above this gradient, which for coal is about 0.6 in. water per in. of bed and is nearly independent of particle size, the bed

becomes disrupted. The sharp break referred to, which appears in the data for series 1 to 3, always involves a gradient materially exceeding this limiting value, so it is evident that a continuous fuel bed does not exist above it. It appears at levels of from 8 to 10 in. above the tuyères, which may be 6 to 8 in. below the level of the top of the fuel bed in the retorts. In series 4, however, when the short fire was used, no break appears. The linear portion extends to a distance of 12 in. above the tuyères, at which level the pressure has dropped very nearly to that in the furnace, showing that, in spite of the thin fuel bed of the short fire, there is more fuel over the tuyères in this fire than in the long one.

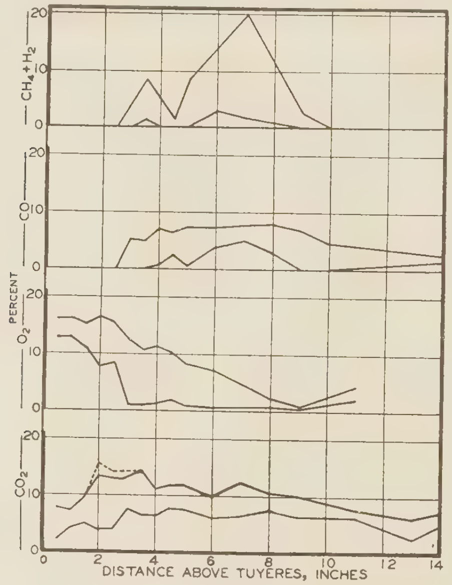

Fig. 10 Distribution of Gas Analyses at No. 2 Tuyère Position

The pressure gradients in the burning lanes at other positions show that in series 1, at low load, the air-flow rate per unit area is much greater at No. 1 position, at the neck of the stoker, and somewhat less at No. 3 position, just above the extension grates, than it is at No. 2. In series 2, the same effect was observed but to a much less marked degree, while in series 3 not enough data are available to permit a comparison. In series 4, the pressure gradients and, hence the rates of air flow per unit area, are very nearly the same at all three positions.

The pressure drops in runs in which the probe did not pass up through the burning lane show none of the regularity just described. They usually show a short section of not over 3 to 4 in. in which the gradient is somewhat less than that observed in the lane, followed by a portion, corresponding in position to the region of decreasing temperature mentioned in the previous paragraph, where the pressure remains very nearly constant. This region is believed to be in close proximity to, or in the interior of a, relatively impervious wall, since the absence of pressure gradient indicates negligible air flow in the direction of motion of the probe. The presence of such a wall can, of course, be accounted for by the presence of unbroken coke or of a plastic mass of coal undergoing carbonization. At the top of this region, in the same place that the abrupt increase of temperature was found, the pressure drops abruptly to a value below the furnace pressure. In the retort runs, the same values of pressure just above the top of the bed, from 0.05 to 0.15 in. of water below the furnace pressure, were observed. These results indicate that there is little flow into the furnace from the space immediately above the retorts, since, if there is no pressure drop due to flow upward from this level, a lower pressure than that observed in the furnace would be measured here, because the furnace-draft connection used for the operating meters is at a level about 10 ft above the fuel bed. Thus, at the level of the bed, the chimney effect of a 10-ft column of hot gas would be added to the furnace draft observed with the regular operating furnace draft gage.

Gas Composition. The gas compositions show much less regularity than the quantities just considered and their analysis does not lead to the type of quantitative results obtained in the consideration of temperatures and pressure drops. This is probably due to the fact that the gas analysis is much more subject to transient variations than are the other two quantities, since diffusion in the gas stream is a much slower process than heat transfer by radiation at the temperatures under consideration, or than the mechanical transmission of pressure through the gas which takes place with the speed of sound. For the same reason, it would obviously be necessary for the sampling point to reach exactly equivalent positions with respect to the filaments of flow and the proximity of surfaces of burning fuel in different runs to attain a high degree of precision of the measurements. Since the fuel bed, by its very nature, represents a single structure only in a statistical fashion, it is obvious that only a statistical approach to an average picture of the gas compositions at various points in the bed is available. The average values at different points might be more closely approximated by samples taken from each position over rather long periods of time but, in view of the comparatively short life of the probes in the fuel bed, it was not practicable to take such samples.

The limits, within which one half of the analyses taken at various levels at No. 2 position in all the series lie, are shown in Fig. 10. This figure was constructed by plotting all the data and then passing the lines through the groups of points at each level in such a way that one quarter of the points would be above the upper line, and one quarter below the lower one. These curves enclose the region within which the analysis of a sample taken at that position would probably lie.

The dispersion of these data is so great that significant differences between the results for the different series of runs cannot be found, but the general trends for all series can be observed in Fig. 10. The oxygen concentration falls off more or less rapidly to small values at distances 5 to 6 in. above the tuyères, remains at this low level up to about 10 in. above the tuyères, and then begins to increase again. This behavior is observable, not only in this statistical representation of the data, but also in each individual run. Carbon dioxide increases, more or less as a mirror image of oxygen concentration, to a maximum at 2 to 5 in. above the tuyères and then decreases, being replaced by increasing concentrations of carbon monoxide. Hydrogen and methane begin to appear at distances of 3 to 6 in. above the tuyères, but never reach very high concentrations in the tests in the burning lanes. In runs out of the lanes, the hydrogen and methane may reach very high values, a typical composition from a point 7 in. above the tuyères (run No. 55) being CO_2, 8.2; O_2, 1.2; CO, 7.3; H_2, 15.1; and CH_4, 4.8. The high ratio of hydrogen to methane, which is typical of the data from this region, indicates that these gases arise either from the later stages of the process of carbonization of the coal (18), or from the cracking of methane. Gas analyses in the retort, on the other hand, usually show a preponderance of methane over hydrogen, which is characteristic of the earlier stages of carbonization.

At levels in the bed above those where methane and hydrogen first appear, diluting the gas stream, the combustible gases burn out. At first, the water-gas reaction appears to account for most of the changes in analysis, as the hydrogen and carbon-dioxide concentrations decrease with a corresponding increase in carbon

monoxide and but little change in methane. Later, however, methane and hydrogen disappear at parallel rates, but carbon monoxide is consumed much less rapidly, invariably being the last combustible gas to vanish. In the initial gasification region, below 6 in. above the tuyères, the fuel is almost pure carbon, that is, a very high-temperature coke, as shown by the fact that the sum of the concentrations of oxygen and carbon dioxide, when the monoxide is absent, always amounts to more than 20 per cent and, when the monoxide is present, frequently reaches 20 per cent + $^1/_2$ (CO), corresponding to the combustion of a pure carbon fuel. This result suggests that combustion in the burning lane takes place in accordance with the laws governing "pure-overfeed" action (24, 29).

Previous Investigations

Current descriptions of the fuel beds of underfeed stokers, with the exception of that of Barnes, appear to be based upon knowledge born of long experience in the operation of this equipment, without the control of exact measurements of conditions in such fuel beds. The work of Barnes (5) was concerned with single-retort stokers of domestic sizes, which differ so greatly from the large multiple-retort type investigated in this study, it could not be predicted that the results of his work would be applicable to these stokers. It appears, however, that if due allowance is made for the differences in the geometry of the system his results may be transferred with little change to a description of the beds of multiple-retort stokers, which usually operate in the region described by Barnes as "black-center" burning, and show many of the characteristics of such operation.

R. A. Foresman (12) has given a description of multiple-retort-stoker fuel beds which agrees quite well with that developed from this study, but no supporting data were given and conclusions which might be drawn from the geometry of the bed were not stated. In particular, he showed a deep pile of uncoked coal in the retort extending to a considerable height above the tuyères, but his description does not suggest the function of the coke wall in delimiting the "burning lane," at least with coking coals, or consider the implications of the horizontal heat flow in determining the rate of ignition. Tobey (33) has also described fuel beds which have many points of similarity to those described here. Houghton (16) has used the results of analyses of the fuel at different points in a fuel bed, which had been suddenly quenched, in an attempt to describe the progress of combustion in multiple-retort stokers, but he appears to share the impression (17) that ignition in such stokers proceeds by a mechanism analogous to that which controls "pure underfeed burning." His emphasis on the desirability of carrying thin fuel beds in burning low-volatile coal is supported by the results of this investigation, which demonstrate the paradox that a thinner fuel bed may actually have more fuel over the tuyères.

In "pure underfeed burning," as described by Nicholls (27, 28), ignition of incoming fuel proceeds in the direction opposite to that of the air flow and takes place, as shown by Mayers (24), if the rate of heat conduction from the zone of free combustion is greater than the rate at which heat is convected back into that zone by the primary air. This type of burning leads to a fuel bed which is stratified in horizontal layers. The upper layers are in a condition similar to that described by Kreisinger, Ovitz, and Augustine (19), except that the temperatures, in the case of underfeed burning, are lower than those found by the last-mentioned authors in agreement with calculation (24). The rate of ignition in such a

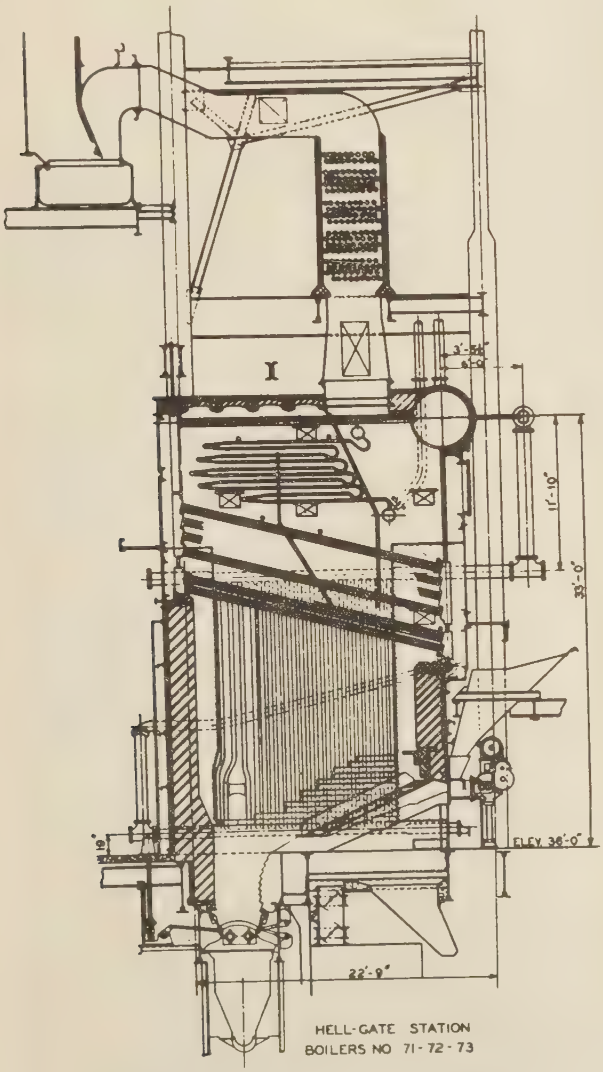

Fig. 11 Cross Section of Boiler and Stoker; Boiler No. 73, Hell Gate Generating Station

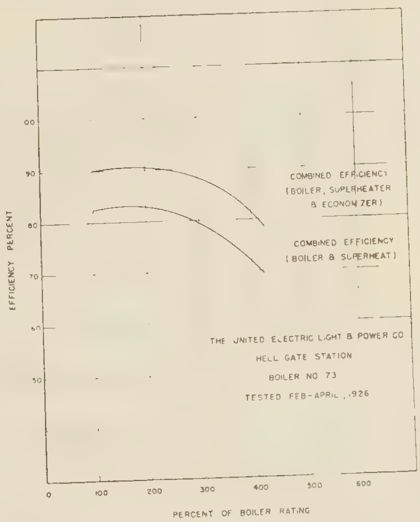

Fig. 12 Boiler Efficiency at Various Loads; Boiler No. 73
(From acceptance tests by United Electric Light & Power Company.)

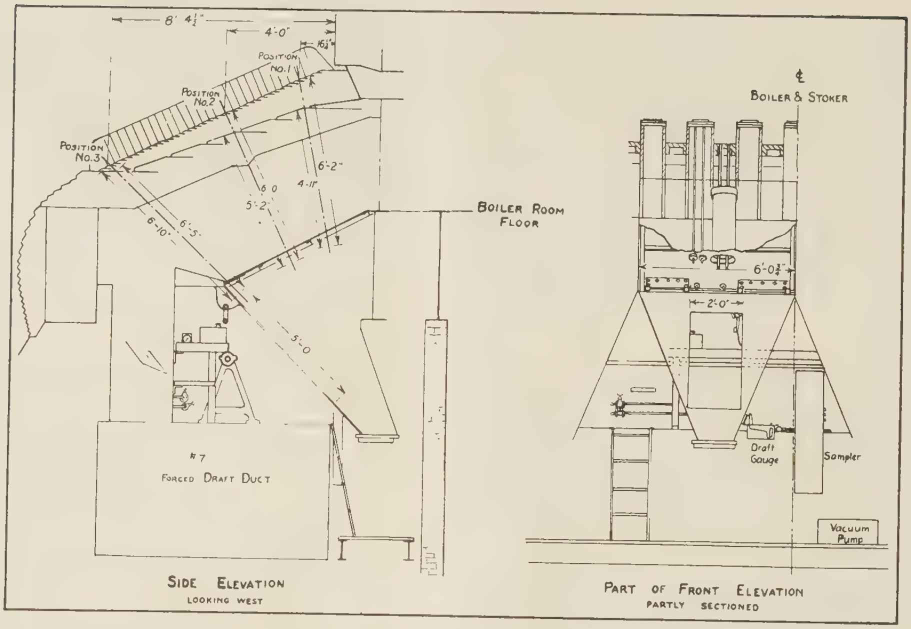

Fig. 13 Arrangement of Test Place or "Doghouse"

system can be calculated (26) from the characteristics of the fuel and the rate of air flow; it has also been determined experimentally (30, 31) as a technical characteristic of the fuel.

In Europe, where traveling-grate stokers are extensively used, it has been shown (14, 21, 23) that the concept of "pure underfeed burning" describes the processes occurring on such stokers, and this has been made use of in setting up model fuel beds (9, 13, 20, 32) for the study of fuel beds on traveling-grate stokers under controlled conditions. These investigations were of little use as a guide to the development of methods of measurement in the present project. In the investigations of traveling-grate stokers of commercial sizes, only grate temperatures and gas analyses over the fuel bed were determined, while the others, in which temperatures within the bed were determined, were carried out on model fuel beds in which equipment could be used which would not stand the conditions of operation in a full-size multiple-retort stoker.

Equipment and Construction

The tests were run on the stoker of boiler No. 73 at Hell Gate Generating Station of the Consolidated Edison Company. The boiler has 12,560 sq ft of heating surface and is a straight-tube, sectional-header type with an overdeck superheater. The furnace is water-cooled on the side walls and has 10 small secondary-air admission ports in the front wall. It has a 14-retort 37-tuyère Taylor stoker with high side-wall tuyères and a clinker-grinder ashpit. The unit is rated at 165,000 lb of steam per hr at 280 psi abs and 700 F steam temperature. A cross section of the boiler and stoker is shown in Fig. 11. The results of acceptance tests, made by the United Electric Light & Power Company in 1926, are shown in Fig. 12. Coal consumption during the tests was calculated from the steam flow indicated by the boiler meter, on the basis of the efficiencies found in these tests, divided by an operating factor of 1.06.

Although the stoker is rated at 37 tuyères, the original tuyères ($1^1/_2$ in. thick) have been replaced by thinner ones, 1 in. thick, so that there are actually 49 tuyères in each stack. There are 5 secondary rams in each retort, all bolted together so that they move as a unit, although provision had been made in the design of the stoker to allow certain amounts of lost motion between adjacent pushers; and the extension grates also move with the secondary rams and have the same stroke. The stroke of the set of rams serving each retort may be controlled from the front of the stoker by movement of an adjustable shoe. The furnace has two large inspection doors in the side walls above the ends of the ashpit. For these tests, an additional small inspection door was provided in the rear wall directly opposite the retort and tuyère stack in which the traverses were made.

Fig. 13 shows the construction of the doghouse, which was built into the sifting hopper to the east of the center line of the stoker, there being 4 hoppers under the stoker. A hole, 2 ft wide and 5 ft long, was cut in the rear sloping wall of the hopper and was provided with a cover plate which could be bolted over it when tests were not being carried on. The top of the hopper was roofed over with $^1/_4$-in. plate, reinforced by 2-in. angles, while the rear 6-in. section of the roof was built in the form of a trap door which could be operated from outside the hopper.

Water, compressed-air, and power connections were brought down along the wall of the wind-box connection, and a table was set up between that wall and the wind-box damper-operating shaft, as shown in Fig. 13. An electric fan standing on the table and directed into the doghouse was used to insure adequate air circulation, and a field-artillery telephone set with a line from the doghouse to the boiler-room floor near the panel of boiler No. 73 allowed ready communication between the two principal test stations.

Eight test points in the stoker were selected, four in the sixth retort from the east side and four in the tuyère stack adjacent to it on the east. These were divided into three groups: Position No. 1, 16 in. from the front wall, having one point in the tuyère stack and one in the retort; position No. 2, 4 ft from the front wall, having two points in each; and position No. 3, 8 ft $2^1/_2$ in. from the front wall, having one in each. It was originally planned to place the two points in both tuyère stack and retort at No. 2 position unequal distances from the center lines, but this proved to be impracticable because of constructional difficulties. In order to clear the operating link for the extension grate, it was necessary to place the point in No. 3 retort position 2 in. east, that is, toward the test-tuyère stack, of the center line of the re-

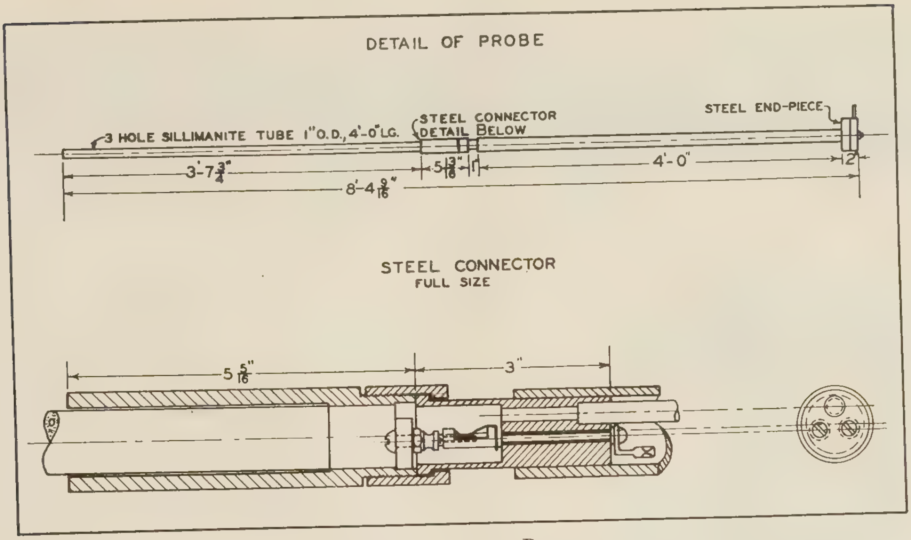

Fig. 14 Uncooled Probe

tort. At each test point, the stoker iron was burned out to accommodate a length of $1^1/_2$-in. standard pipe welded in place, which extended down through the roof of the doghouse. The guide tubes from the tuyère stacks passed through packing caps, welded to the roof, while those from points in the retort passed through 3-ft lengths of flexible steel tubing, welded to the roof at their lower ends and to plates welded to the guide tubes at their upper ends. All of the guide tubes terminated below the roof of the doghouse in square-cut surfaces which were used as reference planes to determine the distance of penetration of the probes into the fuel bed.

Test Apparatus

The probes used to measure temperatures and to take gas samples were required to withstand temperatures up to 3150 F, temperature gradients of 1000 deg per in., and the mechanical stresses incident to being forced through fuel beds 1 to 3 ft deep, containing coal in various stages of carbonization, coke, and occasional clinker, but still had to be small enough so that they did not seriously disturb the fuel bed and so that they might reach temperature equilibrium with the bed within a reasonable time. It was anticipated that each traverse of the fuel bed would probably be terminated by failure of the probe, but it was hoped that many of the traverses might extend to the top surface of the bed.

The water-cooled probe had a cooled section 8 ft long and a 12-in. mullite tip, $^7/_{16}$ in. diam. It was found that the tip was not strong enough to withstand the mechanical stresses to which it was subjected in tests over the tuyère stack, so its use was discontinued for such tests after run No. 20, the last of the preliminary group. Until run No. 48, the water-cooled probe was used in retort runs to locate the region of rapid temperature rise before stopping the stoker, but this procedure, which involved changing probes during a run, was abandoned after it was discovered that the uncooled probes were sensitive enough for this purpose. In run No. 43 the water-cooled probe was used with a shortened tip to withdraw gas samples from the upper levels of the fuel bed and to attempt to observe the probe from the boiler-room floor above the tuyères, which had not previously been done in regular runs. No temperature measurements were made during this run, as it was felt that the close proximity of water-cooled surfaces to the measuring junction would invalidate the observations. The attempt to observe the probe was unsuccessful, but this was accomplished in regular runs in the later series.

With the exceptions noted, uncooled probes, shown in Fig. 14, were used for all the measurements reported. These probes consisted of 1-in. mullite tubes, 4 ft long, with three holes, two $^1/_{16}$-in-bore for thermocouple leads, and one $^1/_8$-in-bore for gas sampling. The bottom six inches of the mullite were covered with a sprayed coating of copper so that the mullite tube could be soldered into the $1^1/_2$-in-outside-diam sleeve by which the probe end was connected to the 1-in. pipe extension or carrier. This sleeve, shown in detail in the lower part of Fig. 14, was connected to the carrier by a copper-gasketed joint and carried a bakelite junction piece on which the thermocouple wires were connected by binding posts to pins which mated with the jacks in the carrier from which compensating lead wire ran to a polarized porcelain receptacle in the hexagonal base of the carrier. A copper tube, $^3/_8$ in. diam, connected the gastight space in the connecting pieces with a nipple on the hexagonal base. A sleeve of $^1/_4$-in. standard pipe, welded to one side of the hexagonal base of the carrier parallel to the axis of the probe, was drilled every $^3/_8$ in. so that a chain passed through it to support the probe could be fixed in position by a pin passing through a set of holes and the chain. The pipe extension was marked by grooves extending one quarter of the distance around the pipe at every inch and by numbered grooves extending all the way around the pipe at every 5 in.

Thermocouples were made of No. 24 gage platinum and platinum 10 per cent rhodium wire. Silica sleeves, 24 in. long and $1^1/_2$ in. outside diam, were cemented to the measuring ends of probes which were to be used in the burning lanes so that they projected $^1/_8$ in. or somewhat less beyond the end of the probe, and cemented caps of Alundum (commercial alumina) and Insa-lute, a proprietary sodium-silicate cement, were laid over the thermocouple junctions in the cups so formed. After drying for 3 to 4 days, these caps were baked in a muffle furnace at a temperature of 800 F for 8 hr and allowed to cool slowly. The development of this procedure took until about the middle of the third series; prior to this time, the caps had not been baked and only those that had been stored for at least 3 weeks penetrated far into the fuel bed. Probes which were to be used in the retorts were not equipped with sleeves or caps after run No. 58, as the conditions in this region were not severe enough to require them. A view of a probe end, showing the sleeve and cap, is given in Fig. 15.

The apparatus for drawing gas samples, designed and built by the Research Bureau of the Consolidated Edison Company and illustrated in Fig. 16, was arranged to evacuate the single-opening

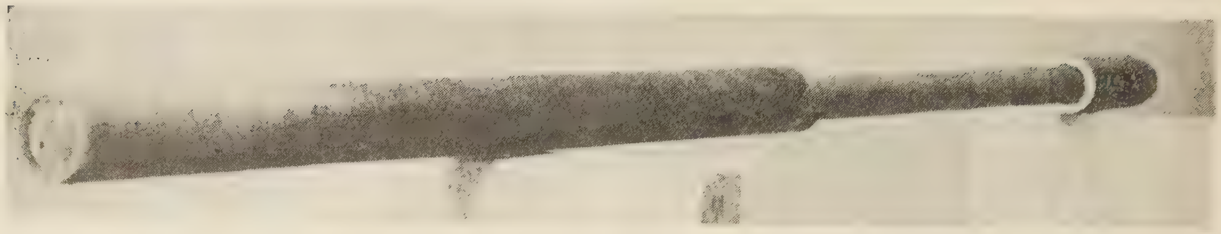

FIG. 15 END OF UNCOOLED PROBE, SHOWING SLEEVE AND CAP

FIG. 16 GAS SAMPLER

sample bottles, flush the sampling line, withdraw a spot sample of the gas over mercury, and compress it into the sample bottle at a pressure 20 to 30 mm of mercury above atmospheric. The copper-tube sampling line ran from a convenient point in the doghouse, where a rubber-tube connection to the probe was wired to it, to a point near the sampler to which it was also connected by rubber tubing. It contained a tee, isolated by plug cocks, through which it was connected to the low-pressure side of a double-range oil-filled draft gage. The high-pressure side of the gage was connected to the wind-box section immediately below the portion of the stoker which was being tested, so that the gage read directly the pressure drop through the stoker and fuel bed. This arrangement was used to reduce the influence of minor fluctuations of wind-box pressure on the readings.

Gas samples were analyzed by means of a special Ellison gas analyzer having a 50-ml burette graduated to 31 per cent and with 250-ml solution containers, and by a large mercury-sealed apparatus, similar to that described by Evans and Davenport (10). All samples were run first in the small analyzer to determine CO_2, O_2, and CO. If the total of these compounds fell below 18.5 per cent plus the amount of CO, the samples were transferred to the large apparatus, in which the absorption of CO was completed, O_2 was added, and the sample burned on a platinum filament. The results of the combustion were calculated on the assumption that the gas consisted only of H_2 and CH_4. No attempt was made to analyze for unsaturated hydrocarbons. The speed of analysis was limited by the time required for combustion. When a long series of samples containing combustibles was being analyzed, the combustion analysis of every second or third sample was sometimes omitted, as each sample required approximately 40 min. Determination of CO_2 and O_2 on the small analyzer could be made continuously at the rate of about 15 samples per hr.

Motion pictures of the fuel bed were taken, following the completion of the tests. Plans had been made for taking these pictures during the tests themselves, but the apparatus was not completed in time to make this possible. The piece of equipment which made successful pictures possible was the pyroscope which has already been described (22). It was designed and constructed by the research bureau of the Consolidated Edison Company in cooperation with the Bausch & Lomb Optical Company, Rochester, N. Y. In these tests, the pyroscope and camera were usually mounted behind the boiler, and pictures were taken through the special observation door at a rate of one frame every 3 sec for periods of several hours. Shots were also taken from the side door of the furnace, looking across the fuel bed, using a camera speed of 64 frames per sec (slow motion) in order to observe the action of "popcorn," or flycoke.

In addition to the instruments just described which were developed especially for these tests, the regular operating instruments were also used and these were supplemented in some cases by calibrated test instruments. Steam flow was read from the Bailey boiler meter, which, in this installation contains two steam-flow elements, one for the west-side superheater outlet, and one, connected with the west-side meter by a totalizing linkage, for the

TABLE 9 BAILEY AIR-FLOWMETER CHECKS; BOILER NO. 73 HELL GATE GENERATING STATION

Date	11/18/37		2/24/38		3/11/38		4/6/38	
Steam-flow reading	148		224		220		207	
Air-flow reading	143		231		232		215	
	Gas-analysis traverse, per cent CO_2							
Distance from side wall, ft	E	W	E	W	E	W	E	W
1	8.0	9.7	9.9	9.8	10.6	10.0	8.6	8.0
3	11.6	10.8	11.0	10.6	11.4	10.2	11.0	11.1
5	15.6	13.6	11.4	12.3	13.3	11.9	13.4	12.5
7	16.8	15.1	12.6	13.0	13.8	12.7	14.0	12.7
9	16.7	15.6	14.7	13.6	14.4	14.2	14.4	12.7
11	15.8	16.6	14.6	14.2	15.3	14.2	14.1	14.0
Average CO_2, per cent	13.8		12.3		12.7		12.2	
Corresponding total air, per cent	135		152		147		153	
Total air with pens together, per cent	140		147		140		148	
CO_2 with pens together, per cent	13.4		12.7		13.4		12.7	

NOTE: Boiler out for overhaul, thoroughly cleaned January 25 to February 24. External cleaning only, March 19–20.

east side. These meters were checked against a static water column several times during the course of the tests and at no time was any correction found necessary. Air flow was read from the Bailey-meter air-flow pen, the setting of which was checked against Orsat traverses four times during the course of the tests, with the results shown in Table 9. Both steam- flow- and air-flow-meter checks were run by the Station Service Bureau of the Technical Service Department, using their standardized procedure. The air-flow-pen setting was not changed during the course of the tests, so that the variations in excess air with pens together are due to changes in the cleanness of the boiler. These variations were taken into account in calculating air flows from the observations. Furnace- and boiler-outlet drafts were read on an inclined draft gage, and the wind-box pressure under the test section was read on a vertical draft gage. The latter reading was compared with the operating wind-box pressure-gage reading, taken below the division plates between sections, to assure uniform operation during the tests. Steam temperatures were read on two engraved-stem test thermometers set in wells in the two superheater outlet headers, and feedwater temperature to the boiler was read on a test thermometer in the economizer outlet. A Ranarex CO_2 recorder was installed with a sampling line drawing from the third pass of the boiler directly above the test section, in order to observe the effect of changes in operating conditions on the gas stratification in the boiler passes.

The speed of the stoker was observed by timing a revolution with a stop watch and the length of the strokes of the secondary rams was measured with a foot rule. A log of all the ram strokes was kept after run No. 68; prior to this time only the stroke in the test retort had been recorded.

Coal Used in Tests

Coal is delivered to the station by barges from which it is hoisted to unloading towers containing screens and crushers which may be by-passed. From the tower it is dropped into cars operating on a cable railway which distribute it to the various bunkers. Samples for proximate analysis and heating-value determination are taken from the stream flowing from the coal tower into the cars. Coal for these tests was delivered to bunker No. 6, of about 350 tons capacity. The lorry serving the sixth and seventh rows of boilers could be filled from No. 6 bunker by the use of a transfer feed screw. Throughout the first four series of tests, the bunker was kept filled with Lower Kittaning run-of-mine coal, excepting for a period of about 2 days around March 15 (just before series 3), when a special coal under test by the operating department was in the bunker. For each of the last three series of tests in which special coals were used, the bunker was carefully cleaned before the special coals were dumped. When the Pocahontas No. 3 and "sized" Lower Kittaning coals were being hoisted, the crusher was by-passed, but at all other times the coal passed through the crushers as in normal operation. The stoker hopper was filled at hourly intervals from the weigh lorry, the weight of each dump being recorded. The lorry was completely emptied before drawing coal from No. 6 bunker for the test boiler.

Samples of coal for size analysis were taken from the stoker hopper by means of a long-handled scoop with a capacity of about 3 lb. Four scoopfuls were taken at points spaced approximately evenly across the stoker hopper to give gross samples of about 300 lb. The size analysis was determined by the use of a Tyler 12-in-square rocking-sieve shaker for sieves coarser than $^3/_8$-in. round hole, and 8-in. round hand screens for finer sizes. No measurements were made of segregation in the stoker hopper, as visual observation indicated this to be negligible as is to be expected in view of the use of a lorry for filling. The analyses of the coals are given in Table 2; the size analyses are plotted to log probability coordinates in Fig. 1. The size distributions of the coals used in series 1, 3, 4, and 5 are so much alike that they may be represented by a single curve, which is not, however, a continuous straight line. Series 2 has a coarser size distribution than that of the other series using Lower Kittaning run-of-mine coal, because the crusher usually used was out of service during a part of the period. The coal of series 6 was sized at the cleaning plant by removal of the $^5/_{32}$-in. fines, and has a somewhat coarser distribution than the other coals. There was a considerable amount of degradation in transit so that almost 45 per cent of it is finer than $^1/_8$ in., as received. The high-volatile coal, used in series 7, is almost as coarse in the large sizes as the coals of series 2 and 6 but contains an excessive amount of superfines, about 9 per cent passing a No. 100 screen.

All analytical data excepting sizing were determined by the technical service department of the company, using standardized procedures.

Testing Procedure

A full crew for the tests consisted of seven men, four being assigned to the doghouse, and one each to the boiler-room floor, to the chemistry, laboratory, and to relief and supervision. Of those assigned to the doghouse, two handled the probes within the doghouse itself, one operated the gas sampler and read the pressure differential, and one read the potentiometer, handled communications with the boiler-room floor, and recorded data. Operation of the gas sampler required the development of a considerable degree of skill, so that every effort was made to keep the same man on this job. The man on the boiler-room floor read the boiler-operating and test instruments at 15-min intervals, took coal samples from the stoker hopper each time it was filled, supervised the operation of the stoker, and kept the crew in the doghouse informed of the conditions of operation. The man assigned to the chemistry laboratory, who operated the gas-analysis apparatus with the intermittent assistance of the man on relief, had to develop a high degree of skill, and continuity in this assignment was essential for satisfactory analytical results. The relief man generally supervised the entire procedure, transported samples and probes, assisted in the gas analysis, and relieved the men in the doghouse at intervals. Relief was especially necessary for the men assigned to handling the probe, as the doghouse was usually hot and sometimes was rather gassy because of fumes blown down through the guide tubes from the fuel bed. Tests could be run for short periods with only six men, but this was avoided whenever possible, since it invariably resulted in the gas analysis falling behind.

Different procedures were used in tuyère tests and tests over the retort. In a tuyère test, the probe was inserted in the guide tube and raised until the tip was about 10 in. below the level of the tuyère plates. It was then blown out from the compressed-air line and the gas-sampling line and thermocouple-extension leads were connected. At this time the potentiometer, gas sampler, and draft gage were checked. Then the probe was moved up to 1 in. below the level of the tuyère. The chain was passed through the sleeve and pinned in place, the temperature and draft readings were started. The man on the boiler-room floor, who had previously notified the doghouse that conditions were sufficiently steady and representative for testing, was informed by telephone of the beginning of the test. The probe was advanced by $^1/_2$-in. steps, temperatures and pressure differentials being read at each step, and from $^1/_2$ in. above the level of the tuyère gas samples were taken at each position. Beyond 5 in. above the level of the tuyères the probe was advanced by 1 in. at each step after run No. 64. The probe was held in each position for at least $2^1/_2$ min, which was sufficient to allow reading the draft gage, a 40 to 50-sec flushing of the gas-sampling line, and drawing a gas sample,

TABLE 10 SAMPLE LOG OF TUYÈRE TEST; RUN NO. 72

(No. 2 tuyère, east; zero = 25½ in.; March 25, 1938)

Time	Position, in.	Potentiometer reading, mv	Temp, F	Bed differential, in. water	Sample no.
2:19	24½	0.8		2.05	
		1.1			
		1.3			
		1.4	385		
2:21	25	1.7		1.0	
		1.7			
		1.7	450		
2:22	25½	1.6		0.44	
		1.6			
		1.7	450		
2:23½	26	2.1		0.44	43
		3.3			
		4.2			
		4.6			
		4.9	1055		
2:25½	26½	5.0		0.43	16
		5.4			
		6.0			
		6.4			
		6.8			
		6.9	1400		
2:28	27	7.1		0.47	17
		7.6			
		8.4			
		8.8			
		9.3			
		10.0	1900		
2:31	27½	10.5		0.55	18
		10.7			
		12.2			
2:32½	27½	12.5			
		12.6	2295		
2:33½	28	12.9		0.60	19
		13.2			
		13.9			
		13.8	(13.9)		
		13.9	2490		
		14.0			
2:36	28½	14.2		0.59	20
		14.2			
		14.2	(14.0)		
		13.9	2505		
		13.8			
		13.7			
2:38½	29	13.6		0.90	21
		13.6			
		13.6			
		13.7	(13.8)		
		14.1	2475		
		14.1			
		14.1			
2:42	29½	13.9		0.80	22
		14.1			
		14.2			
		14.3	(14.2)		
		14.4	2535		
		14.2			
		14.1		0.90	
2:46	30	13.8			
2:47		14.3	(14.3)		23
		14.2	2550		
		14.3			
2:48½	30½	14.3		1.10	28
		14.4			
		14.4			
		14.6	(14.5)		
		14.6	2589		
2:51	31½	14.4			
		14.9		1.25	29
		14.9	(15.0)		
		15.1	2655		
		15.2			
		15.1			
2:54	32½	15.3		1.25	30
		15.4			
		15.1	(15.0)		
		15.1	2655		
		14.8			
		14.8			
		14.8			
2:57	33½	14.8		1.55	31
		14.7			
		14.3	(14.3)		
		14.2	2550		
		13.8			
		14.0			
		14.3			
3:00½	34½	14.7		2.10	32
		14.9			
		15.1			
		15.3	(15.6)		
		15.6	2750		
		15.7			
		15.6			

while temperatures were read every ½ min. The probe was advanced until it broke, as indicated by open circuit of the thermocouple, or until the probe was sighted above the level of the fuel bed. A portion of the log from run No. 72 is shown in Table 10,

TABLE 11 SAMPLE LOG OF RETORT TEST; RUN NO. 61

(No. 2 retort, west; zero = 7½ in.; March 9, 1938)

Time	Position, in.	Potentiometer reading, mv	Temp, F	Bed differential, in. water	Sample no.
		Stoker stopped at 10:09			
	7½	0.2		2.9	
	12½	0.2	95	2.2	
	17½	0.2		2.2	
	22½	0.2		2.5	
	25	0.2		2.6	
	26	0.2		2.6	
	27	0.2		2.65	
	28	0.2		2.65	
	29	0.2		2.7	
	30	0.2		2.8	
	31	0.2		2.8	
10:14	32	0.2	95	2.8	
10:14½		0.6	200		
15		1.1	320		
		1.8	470		
16		2.3	570		50 Slow
	33	3.6	820	2.9	
17		4.0	890		
		4.7	1020		
18		5.1	1090		
		5.5	1160		
19		5.7	1195		51 Slow
	Pulled probe out				
	Stoker stopped at 10:53				
10:54	31	0.5	175	3.1	
55		0.6	200		
		0.8	250		52
56	32	1.1	320	2.95	
		1.7	450		
57	32	2.3	570		
		2.9	685		
58		3.4	780		
		3.9	875		53
59	33	4.4	965		
		5.5	1160	2.8	

NOTE: No sleeve and no Alundum on tip of probe.

The procedure in the retort developed as the tests were carried on. The problem that necessitated a difference in procedure between the retort and tuyère tests was that introduced by the motion of the stoker. Over the tuyères, the fuel did not move enough to break the probe but, in the retort, the relative motion between fuel and pusher was sufficiently great to shear the probe off at every stroke. Preliminary tests showed that the probe could be kept in the fuel bed during the time the pusher was moving down the stoker, that is, toward the bridge wall, without breakage, but on the return stroke the probe was broken every time it was allowed to remain in. Since a time longer than the period between strokes was required for taking readings at each position, it was obvious that the stoker, or at least the section of it under test, must be stopped while the readings were taken. About 10 min prior to the test, the test section of the stoker was put in high speed by means of the gear-change box. The probe was inserted to a position several inches below the level of the secondary ram and the instruments checked. Then the man on the boiler-room floor was instructed to stop the test section and inform the doghouse .he instant this was done. The prob was raised to the level of the top of the ram and advanced by 5-in. steps as rapidly as readings of temperature and pressure differential could be made. This was continued up to the neighborhood of the region of rapid temperature rise, known from previous experience to be more than 25 in. above the level of the ram at No. 1 position, and more than 20 in. at No. 2 position. From here on the probe was advanced by 2-in. and then by 1-in. steps, holding it at each position for about ½ min, in order to make certain that any rise in temperature would be observed. As soon as the galvanometer needle moved off zero, the advance was stopped and the temperature was recorded at ½-min intervals, while the pressure differential was read and a gas sample drawn. From this point, the probe was advanced 1 in. at a time, taking 2½ min at each position, until not more than 13 min had elapsed after stopping the stoker, when it was withdrawn. The section of the stoker was again started in high speed and run for about 15 to 20 min to build up the fuel bed to its original condition. If the

TABLE 12 SAMPLE OF BOILER-ROOM FLOOR LOG; RUN NO. 72; MARCH 25, 1938

Chart time	Steam Total	Steam West	Air	Draft Furnace, in. water	Draft Third pass, in. water	Wind-box pressure, in. water	B.O. gas temp, F	Superheat temperature East, F	Superheat temperature West, F	Feed-water temp, F	Pusher stroke, in.	Time for 1 revolution, min-sec	Ranarex CO_2, per cent
				0 = 0.10									
2:15	195	160	237	0.27	1.03	2.80	595	655	658	323	5/8	1–11	13.0
30	200	180	237	0.28	1.035	2.80	598	656	658	326	5/8	1–11	12.5
45	195	180	237	0.275	1.045	2.80	600	661	657	328	5/8	1–8	12.5
3:00	200	180	237	0.275	1.01	2.70	600	662	655	327	5/8	1–8	12.5
15	196	182	236	0.295	1.035	2.70	595	660	654	325	5/8	1–6	13.0

Fire Conditions

Chart time

2:15—Fuel bed appears even over entire stoker from west door, but from east door appears heavier on west side from neck down to 1/2 stoker length. Fire light but not much "popcorning." Considerable amount of small clinkers on extension grate, and under fuel in retorts and on tuyères; fuel rather well burned out on extension grates. Furnace clear except at neck where it is quite smoky; flame short; no secondary combustion top of first pass.

Chart time

2:38—Speeded up stoker on rheostat to 1 revolution in 1 min 8 sec. Half-moon effect on extension grate.

2:45—Burning lane at test position veering off to right, viewed from rear door.

3:00—Fire light.

3:10—Saw probe in center of burning lane. Tip of probe about 2 in. below top of burning lane and about 7 to 8 in. below top of fuel bed.

probe had not been observed during the previous part of the run the stoker was stopped again and the probe was again inserted and advanced by 1-in. steps from 1 in. below the last previous position. The probe was invariably sighted from the boiler-room floor during the second half of the test before 13 min had elapsed.

In tests in No. 3 retort position, the procedure was yet different since at this position the guide tube terminated in the dead plate just above the extension grate, where the motion of the fuel was not sufficient to break probes when they were less than 5 in. above the level of the dead plate. In tests at this position, the probe was inserted and moved upward by 1-in. steps to 5 in. without stopping the stoker. When high temperatures were observed, gas samples were taken at each level. When the probe reached 5 in. above the dead plate, the stoker was stopped and the run was continued in just the same way as in any other retort test. A sample log of a retort run (No. 61) is given in Table 11, and a sample of the boiler-room floor log is given in Table 12.

Stoker Operation

During the tests, more uniform fires than those required for normal operation had to be maintained to permit duplication of conditions in the various tests in each series, and to allow the calculation of the air flow through the test section as the average air-flow rate through the stoker. For this reason, it was necessary to place the operation of the boiler on a special schedule. Over week ends the doghouse was closed so that siftings could be discharged through the trap in the doghouse roof and the boiler was in normal operation on automatic control. From Monday to Friday the fire was cleaned as early as possible on the 12 to 8 watch, but the pit was not ground down after cleaning. The fire was normally banked during and after cleaning, and was operated on automatic control in accordance with load requirements after coming off bank until 8 a.m., when it was placed on hand control and run at the test rating during the remainder of the day. At the end of the tests, it was put back on automatic control and was in normal operation for the remainder of the 4 to 12 watch. The doghouse was opened at 8 a.m. on Monday and was left open, siftings being allowed to accumulate on the roof, until Friday night. Slagging of the first pass of the boiler with hand water lances, which had to be done three times a week on the day watch, was carried out the first thing in the morning on Tuesdays, Thursdays, and Saturdays, thus interfering with testing on only 2 days, when the testing day was shortened by about 1 hr.

During normal operation, the output of the boiler is governed by an automatic regulator (Smoot combustion control) which controls the air flow through the boiler in accordance with impulses sent out by a master controller, actuated by the main steam pressure. This regulator controls the draft at the boiler outlet; the wind-box pressure is controlled by another regulator,

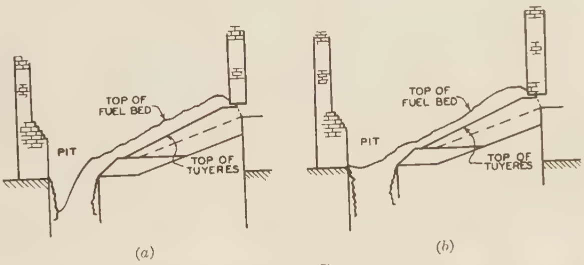

Fig. 17 Fire Contours
(a, Normal or long fire; b, short fire.)

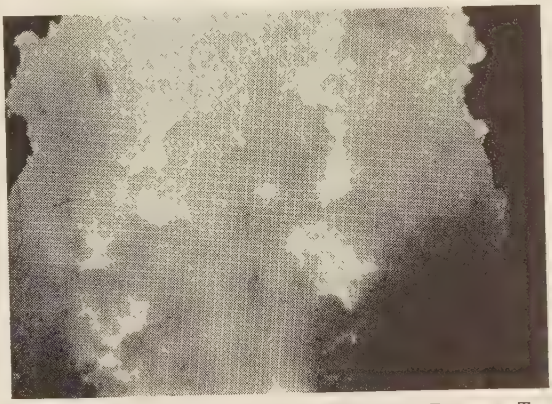

Fig. 18 Long Fire; Single Frame From Motion Pictures Taken May 13, 1938
(Load 124,000 lb of steam per hr; excess air 36 per cent.)

Fig. 19 Short Fire; Single Frame From Motion Pictures Taken May 14, 1938
(Load 119,000 lb of steam per hr; excess air 53 per cent.)

not subject to loading by the master controller, which maintains a constant furnace draft. Stoker speed is determined by the voltage generated by a separate motor generator set for each row of boilers, not usually operated on the automatic control, modified by individual stoker-motor field rheostats at each boiler. During tests, the boiler outlet draft was taken off automatic control so that the regulator did not receive the impulses from the master controller, and was set by hand at a constant load. The windbox pressure regulator was left in service and the stoker-motor motor generator set was adjusted to give a voltage corresponding to the test load.

The stoker was operated by the regular operators with the advice and assistance of the test crew. This assistance became more extensive as the tests advanced and the conditions of operation departed more and more from normal. During the first two series, while the test crew was relatively unacquainted with the standard of operations and when it was desirable that the operating conditions be as nearly normal as possible, the test crew exercised but little supervision beyond recording changes in the settings of the various controls. With the beginning of series 3, which was run with high excess air, the necessity of operating with a thinner fuel bed than normal made it necessary for the test crew to take part in the adjustment of the stoker to a greater extent than previously; this necessity continued throughout the remaining series, especially when coals which were unfamiliar to the station operators were being fired.

During the first three series of runs, the normal fire contour, sketched in Fig. 17(*a*), was maintained. The sketch shows the approximate profile of the general level of the fire, set by the level of the fuel in the retorts. It could be seen from the rear observation door that the burning lanes were open channels in which a fuel level could not be observed. When the fire was thinned down to secure high excess-air operation in series 3, it was found extremely difficult to keep fuel on the tuyères at the head end of the stoker and to keep blowing from the front wall within bounds. Drifts of "popcorn" built up along the bridge wall and on one occasion the fuel was blown completely away from 3 or 4 ft of the upper end of the stoker. To overcome this difficulty, the contour of the fire was changed to that shown in Fig. 17(*b*), referred to as the short fire, by shortening the strokes of the secondary rams, and testing was continued as series 4.

Short-fire operation was continued throughout the remainder of the tests except with the high-volatile coal, both with high and with normal excess air, as it was found not only to decrease the amount of "popcorning," but also to decrease slightly the degree of stratification in the flue gas, as can be observed by comparing the traverse shown in Table 11 on April 6, run with a short fire, with the others in the table run with a normal fire. It caused a marked decrease in secondary combustion, observed at the top of the first pass, and produced a fire which was more completely burned out on the tail than the normal practice. In fact, it was necessary to carry the pit at a much higher level than the usual practice in this station in order to prevent the appearance of holes in the lower end of the fire. The differences in the general appearance of the long and short fires are shown in Figs. 18 and 19, made from single frames of the motion pictures taken after the tests. Part of the increased clearness of Fig. 19 is due to the increased excess air, as this picture was taken under conditions approximating those of series 4, while Fig. 18 was typical of series 2. The increase in the width of the burning lanes with the short fire is, however, quite apparent.

When special coals were burned for the tests, the new coal was burned for 24 hr before any tests were run, in order to assure that only the special coal would be in the fire during the tests and to give the test crew time to work out the proper conditions for firing the new coal.

Preliminary Tests

The results of the tests run in August and September, 1937, have already been referred to as showing the feasibility of the proposed test procedure and indicating the need for redesign of the original probes to the forms described. In addition to these, the first 20 runs made after the beginning of intensive testing on November 1 were in the nature of preliminary runs and were devoted mainly to a comparison of the compositions of gas samples drawn by the water-cooled and uncooled probes and to the temperature indications of the two kinds of probes.

When the uncooled probes were first used in these runs, they were not equipped with sleeves or caps and it was found that they broke, apparently because of the thermal shock, after penetrations of only 2 to 3 in. H. W. Russell, chief physicist of Battelle Memorial Institute, suggested, on hearing of this difficulty, that the mullite tubes be provided with silica sleeves which, although they could not stand the maximum temperatures encountered, would slow up the rate of heat penetration in the early stages enough to prevent spalling. When this was done, it was found that the probes penetrated well into the bed without breaking; but that the thermocouples were attacked by slag in the region of high temperature, leading to erroneous temperature indications. This difficulty was eliminated by the provision of the cemented caps described in the section on "Test Apparatus." These provisions made the uncooled probes quite massive, causing their indications of temperature to lag in the early stages of each run. The method of correcting for this is described in the next section on "Supplementary Data."

The water-cooled probes used in the preliminary tests were also provided with silica sleeves and caps, but failed because of mechanical breakage, so their use was discontinued. There was, however, some question as to whether gas samples taken with the uncooled probes would be cooled rapidly enough to quench the gas reactions and thus be representative of conditions in the fuel bed. The data in Table 13 show a comparison of gas samples taken from the same levels at No. 2 tuyère, west position, by both the water-cooled and uncooled probes. While these early runs

TABLE 13 COMPARISON OF GAS SAMPLES DRAWN WITH COOLED AND UNCOOLED PROBES AT NO. 2 TUYÈRE, WEST

Height above tuyères, in.	Water-cooled probe					Uncooled probe				
	Run No.	CO_2 per cent	O_2 per cent	CO per cent	Total per cent	Run No.	CO_2 per cent	O_2 per cent	CO per cent	Total per cent
1	4	2.2	15.6	0.3	18.1	3	0.0	15.8	0.0	15.8
	8	3.6	15.4	0.2	19.2	9	2.8	15.9	0.1	18.8
	11	1.7	18.2	0.0	19.7	14	5.2	14.4	0.1	19.7
	15	6.9	12.4	2.5	21.8					
$1^1/_2$						7	0.9	18.4	0.0	19.3
						10	6.9	10.6	...	..
3	2	7.9	6.2	2.9	17.0	3	0.7	19.1	0.7	20.5
	4	6.7	5.1	3.7	15.5	7	8.1	11.8	0.2	20.1
	8	13.8	4.5	3.1	21.4	9	8.1	6.8	4.0	18.9
	11	0.7	20.3	0.3	21.3	10	11.9	5.6	2.1	19.6
	15	13.1	5.8	0.4	19.3	14	14.0	1.2	3.8	19.0
5	11	1.5	18.8	0.4	20.7	7	8.5	11.2	0.0	19.7
	15	5.7	7.7	6.2	19.6	14	12.3	3.1	4.4	19.7
7	11	0.5	19.9	0.2	20.6	7	1.0	18.2	0.2	19.4

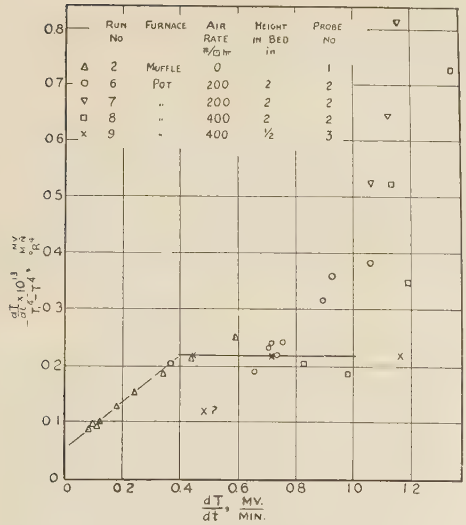

Fig. 20 Data for Correcting Temperature Indications of Capped Probes

rarely penetrated beyond 3 in. into the bed, and the analysis for H_2 and CH_4 in these samples was not completed, the data show that the differences between samples taken by the water-cooled and by the uncooled probes were less than the differences among samples taken by either one. In particular, the samples taken in run No. 9 with the uncooled probe are parallel to those taken in run No. 8 with the cooled probe, those from No. 3 with the uncooled probe to those from No. 11 with the water-cooled probe, and so on.

These results indicated that the water-cooled probe did not quench the gas reactions any more rapidly than the uncooled probe, as might be anticipated from the fact that the top ten inches of the water-cooled probe were uncooled. On the other hand, samples taken with the water-cooled probe and a tip only 2 in. long (run No. 43, when no temperature measurements were made) do not differ consistently from samples taken at the same points with the uncooled probe. Finally, many samples taken with the uncooled probe, from levels more than 4 in. above the tuyères, had analyses showing the presence of up to 10 per cent of combustible gases mixed with 3 per cent of oxygen. These facts indicated that the gas reactions may be rapidly enough quenched even in the uncooled probe to introduce only negligible errors in the gas compositions observed.

Supplementary Data

(a) *Correction of Temperature Indications.* Tables 10 and 11 show that the temperatures, recorded when entering the fuel bed above the tuyères and when passing through the hot layer above the retorts, did not reach steady values. It was recognized that the probes were so heavy as to raise the question of temperature equilibrium between them and their surroundings, so in the earlier tests they were held in position until the rate of rise of tempera-

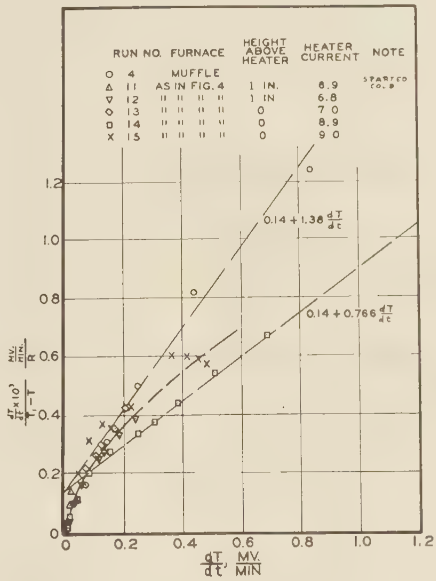

Fig. 21 Data for Correcting Temperature Indications of Uncapped Probes

TABLE 14 RESULTS OF TESTS CALCULATED FROM MOTION-PICTURE FILMS

Set	Film speed: Projected frames per sec	Film speed: Exposed frames per min	Transverse distance as fraction of c to c distance of tuyères	Time of motion as projected, sec	Transverse velocity, fph	Calculated	
					Long fire		
1	12.6	18.0	0.075	6	1.85	Rating, 1000 lb per hr:	
			0.076	14	0.81	Steam	125
						Coal	12.7
2	14.9	18.0	0.106	10	1.33	Burning rate referred to air admission surface	69.8
			−0.057	21	−0.34		
			0.161	28	0.72	Per foot length of tuyères	61.1
3	13.6	18.0	0.034	7	0.67		
			0.068	11	0.85	Fuel bed depth, in.	14
			−0.155	18	−1.18		
			0.106	10	1.45		
Average					0.68 ±0.22	Coal flow rate	0.58
					Short fire		
1	14.8	18.0	0.101	12	1.06		
			−0.089	9	−1.25		
			0.153	11	1.76		
			0.072	13	0.70	Rating, 1000 lb per hr:	
2	13.9	18.0	0.095	14	0.96	Steam	119
			0.095	7	1.83		
			0.057	12	0.65	Coal	12.2
3	14.4	18.0	0.078	8	1.26	Burning rate referred to air admission surface	67.2
			0.070	13	0.70		
			0.039	10	0.51		
			0.068	16	0.55		
			0.072	10	0.94	Per foot length of tuyères	58.8
4	13.9	18.0	0.082	8	1.38		
			−0.028	9	−0.42	Fuel-bed depth, in.	12
			0.037	7	0.52		
Average					0.75 ±0.14	Coal flow rate	0.65

ture fell to 0.2 mv (32 F) per min in the belief that this would assure a discrepancy between the temperature observed and that of the surroundings less than the expected error (150 F). This required 5 min or more and thus shortened the traverses which could be made within the normal life of the probes, so this procedure was abandoned. It became necessary, therefore, to estimate, from the observed data, the temperature which would have been reached by the probe if left for an indefinite time at each of the lower levels.

In order to do this, tests of capped probes were made in one of the pot-type furnaces in the Fuel Laboratory of the Pittsburgh Experiment Station of the U. S. Bureau of Mines, and of the uncapped probes in a special setup in the Coal Research Laboratory. The details of procedure and of analysis of the data are reported elsewhere (25), but the results of the tests are embodied in Figs. 20 and 21, applying to capped and uncapped probes, respectively. These curves give data by the use of which the temperature of the surroundings, referred to as T_1, may be calculated when both the indicated temperature T and its rate of rise at the time of observation $\frac{dT}{dt}$ are known. Thus, suppose the indicated temperature with the capped probe is 1385 F and the rate of rise is 0.63 $\frac{\text{mv}}{\text{min}}$. From Fig. 20, at $\frac{dT}{dt} = 0.63 \frac{\text{mv}}{\text{min}}$, the ratio $\frac{dT}{dt}/(T_1^4 - T^4)$ is 0.233×10^{-13}, giving $T_1^4 - T^4 = 27 \times 10^{12}$. Since $(1385 + 460)^4 = 11.56 \times 10^{12}$, $T_1^4 = 38.6 \times 10^{12}$ and $T_1 = (2490 - 460) = 2030$ F. Fig. 21 is used in the same way except that only the first powers appear and absolute temperatures need not be calculated. It is estimated that corrected temperatures calculated in this way are within ±90 F of the radiant mean temperature at the point with the capped probes, and within ±150 F with uncapped probes.

The temperatures at points, at which a continuous change was observed, have been corrected by the method thus described. These include the points less than 3 to $3^1/_2$ in. above the tuyères and those in the thin (about 2 in. thick) heated skin at the top of the retorts. At all other points, the observed temperatures were averaged, omitting from the average any continuous series which may have appeared at the beginning of the observation period, in which successive readings differed by more than 0.1 mv (15 F). It is estimated that these temperatures are in error by not more than 100 F, and considered probable that, if a consistent error exists, it is such as to make the recorded temperatures too low.

(b) *Estimate of the Flow Velocity of the Fuel.* The magnitude of the transverse component of the fuel flow was estimated from the motion pictures taken at the end of the tests. This component is responsible for the delivery of coked fuel from the retorts to the tuyères and has the direction of the horizontal perpendicular to the center lines of the tuyère stacks and retorts. It is considered positive in the direction from the center line of the retort toward the center line of the next adjacent tuyère stack.

Films taken through the auxiliary door in the rear wall of the furnace were projected through a glass balance case on a screen of tracing paper mounted inside the case, thus making it possible to mark the position on the back of the tracing paper of any prominent object at any time. Using a metronome as a timer, the progress of several bodies of fuel shown in each portion of the film was marked as points at $^1/_2$-sec intervals, while the film was being projected at a known rate. The center lines of the tuyère stacks were also shown on the tracing by passing a line through the average position of the burning lane. The line of dots, representing the progress of each body of fuel, usually sloped downward toward the center lines of the tuyères, although a few sloped away from them. From the slope of the line, making allowance for perspective, the number of dots, representing the time of projection, and the ratio of film speed as projected to that during exposure, the transverse component of the velocity of the fuel could be calculated.

Some results calculated in this way from films taken at a load of 125,000 lb of steam per hr with both long and short fires are shown in Table 14. The films exposed at a lower load were too cloudy to permit calculation. The results have only a low precision, but the values observed are close to those calculated from the dimensions of the bed. Moreover, the observed difference between the flow velocities in the long and short fires is parallel to that calculated.

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BIBLIOGRAPHY

1 Glossary of Technical Terms, Section 3 of Information for Contributors, Symposium on Temperature, American Institute of Physics, November, 1939, p. 3.

2 Ibid., p. 1.

3 "The Dependence of the Resistance of a Layer of Fuel on the Load," by I. W. Arbatsky, *Feuerungstechnik*, vol. 25, 1937, pp. 233–235.

4 "Methods of Representing Distribution of Particle Size," by

J. B. Austin, *Industrial and Engineering Chemistry*, Analytical edition, vol. 11, 1939, pp. 334–339.

5 "Fundamentals of Combustion in Small Underfeed Stokers," by C. A. Barnes, Technical Report No. IV, Bituminous Coal Research, Inc., Washington, D. C., 1938.

6 "The Physics of Coal Carbonization," by S. P. Burke, T. E. W. Schumann, and V. F. Parry, Proceedings, American Gas Association, 1930, pp. 820–855.

7 "Fluid Flow Through Granular Beds," by P. C. Carman, Trans. Institution of Chemical Engineers, vol. 15, 1937, pp. 150–166.

8 "Resistance to Flow of Gases Through Beds of Granular Material," by E. Diepschlag, *Feuerungstechnik*, vol. 23, 1935, pp. 133–136.

9 "Combustion of Fuel on a Traveling Grate," by A. C. Dunningham and E. S. Grumell, *Journal of the Institute of Fuel*, vol. 12, 1938, pp. 87–95.

10 "Analysis of Combustibles in Flue Gas," by R. N. Evans and J. E. Davenport, *Industrial and Engineering Chemistry*, Analytical edition, vol. 7, 1935, pp. 174–178.

11 "Statistical Methods for Research Workers," by R. A. Fisher, sixth edition, Oliver & Boyd, Edinburgh, 1936, chap. 5, pp. 120–175.

12 "Maintaining Proper Porosity of Stoker Fuel Beds," by R. A. Foresman, Proceedings, Eighth Fuel Engineers Meeting, Fuel Engineering Division of Appalachian Coals, Inc., Cincinnati, June 18, 1935.

13 "The Evaluation of Fuel From the Consumers' Viewpoint," by E. S. Grumell, *Journal of the Institute of Fuel*, vol. 5, 1932, pp. 361–370; see especially appendix pp. 366–370.

14 "The Process of Combustion With Chain Grates," by W. Gumz, *Feuerungstechnik*, vol. 24, 1936, pp. 10–12.

15 "The Fundamental Principles Involved in the Separation of Particles by Virtue of Density Difference," by A. A. Hirst, Trans. Institution of Mining Engineers, vol. 85, 1933, pp. 236–268.

16 "The Burning of Bituminous Coal on Large Underfeed Stokers," by Bert Houghton, Proceedings, of the Third International Conference on Bituminous Coal, vol. 2, 1931, pp. 276–293.

17 "The Principles of Underfeed Combustion and the Effect of Preheated Air on Overfeed and Underfeed Fuel Beds," by P. Nicholls and M. G. Eilers, discussion by Bert Houghton, Trans. A.S.M.E., vol. 56, 1934, pp. 335-336.

18 "Pyrolysis of Coal," by Bernard Juettner and H. C. Howard, *Industrial and Engineering Chemistry*, vol. 26, 1934, pp. 1115–1118.

19 "Combustion in the Fuel Bed of Hand-Fired Furnaces," by Henry Kreisinger, F. K. Ovitz, and C. E. Augustine, U. S. Bureau of Mines, Technical Paper No. 137, 1916.

20 "Combustion on Grates," by A. R. Leye, Dissertation, Technische Hochschule, Berlin, 1933. Studentenhaus Drück, Charlottenburg, 1933.

21 "The Rate of Combustion of Coal on a Movable Grate," by R. Loewenstein, *Die Wärme*, vol. 57, 1934, pp. 97–101, 121–125.

22 "Photographic Study of Steam-Boiler Furnaces in Operation," by A. A. Markson and W. H. Dargan, *Mechanical Engineering*, vol. 61, 1939, pp. 729–734.

23 "A Method of Controlling the Temperatures of Traveling-Grate Stokers," by J. D. Maughan, H. B. Spalding, and B. M. Thornton, *Journal of the Institute of Fuel*, vol. 8, 1934, pp. 11–20.

24 "Temperature and Combustion Rates in Fuel Beds," by M. A. Mayers, Trans. A.S.M.E., vol. 59, 1937, pp. 279–288.

25 "Temperature Measurements With Probes of Large Mass in Stoker Fuel Beds," by M. A. Mayers, Paper No. 36 of the Symposium on Temperature of the American Institute of Physics, November, 1939.

26 "Ignition in Beds of Solid Fuel," by M. A. Mayers and H. G. Landau, Proceedings, Symposium on the Combustion of Solid Fuel, Division of Gas and Fuel Chemistry, American Chemical Society, Boston, September 12, 1939, pp. 28–36; also *Industrial and Engineering Chemistry*, April, 1940, pp. 563–568, and June, 1940, p. 877.

27 "Underfeed Combustion, Effect of Preheat, and Distribution of Ash in Fuel Beds," by P. Nicholls, U. S. Bureau of Mines, Bulletin No. 378, 1934.

28 "The Principles of Underfeed Combustion and the Effect of Preheated Air on Overfeed and Underfeed Fuel Beds," by P. Nicholls and M. G. Eilers, Trans. A.S.M.E., vol. 56, 1934, pp. 321–335.

29 Ibid., Fig. 4*A*, p. 324.

30 "The Ignition of Solid Fuel on the Grate," by P. Rosin, H. G. Kayser, and R. Fehling, Bericht D51 der technisch-wissenschaftlichen Sachverständigenausschüsse des Reichskohlenrats, 1935.

31 "Complete Ignition on the Traveling Grate," by P. Rosin and E. Rammler, *Archiv für Wärmewirtschaft*, vol. 13, 1932, no. 5, pp. 113-117.

32 "Temperature History in the Fuel Bed and of the Grate in the Combustion of Coal," by E. Tannner, Veröffentlichungen des Vereins zur Überwachung der Kraftwirtschaft der Ruhrzechen, 1934–1935; also appears as Bericht D60 der technisch-wissenschaftlichen Sachverständigenausschüsse des Reichskohlenrats, 1933.

33 "Underfeed Stokers From the Standpoint of Coal Selection," by J. E. Tobey, Trans. A.S.M.E., vol. 60, 1938, pp. 17–24.

Parallel Versus Individual Operation of Multicyclones

By L. C. WHITON, JR.,[1] PORT CHESTER, N. Y.

In this paper, the author discusses the problem of determining the comparative efficiencies of individual cyclone-type dust collectors and cyclones grouped with interconnecting ducts and hoppers. The test setup and procedure followed are described fully, while the results are completely tabulated. The conclusion is reached that, with a damper-equipped cyclone of the size used in actual practice and operated under identical conditions of resistance and temperature, performance of the individual cyclone is representative of the group in the field.

THE increasing adoption of cyclone-type dust collectors in series for parallel operation indicates the importance of determining whether an individual cyclone or a group of cyclones with interconnecting ducts and hoppers would have the same dust-collection-efficiency characteristics. This is partly due to the fact that extremely accurate collection efficiencies may be determined for an individual cyclone, since a single unit handles a relatively small quantity of gas and, therefore, can be tested for its efficiency, other than by merely sampling the dust in the gas. The possibility of experimental error in the sampling method of determination is necessarily considerable.[2] However, by utilizing an individual cyclone, a far more accurate measure of the efficiency of this cyclone can be made by any one of several methods. For example, a given quantity of dust can be introduced into an otherwise dust-free gas, and the amount collected used for determining collection efficiency.

Another accurate method of test can be arrived at by connecting a single cyclone to a flue, weighing the dust caught in an individual hopper for the collector, and then collecting the escaping dust in a cloth bag and determining the increase in the weight of the bag. By comparing the absolute weight of dust caught and the increase in the weight of the cloth collector, a greater degree of accuracy of the efficiency determination may be attained than by sampling at the inlet and outlet of a cyclone collector, due to the greater experimental error inherent in the sampling method.

Such results have been studied, and field tests on a group of multicyclones of the same design have been shown to correspond accurately with observations made when one cyclone was tested.[3]

Nevertheless, it is important to observe the operation of individual or multiple cyclones under precisely the same conditions, in order to determine whether possible recirculation through certain of the cyclones destroys the efficiency of any of them because of varying static conditions, thus causing an over-all collection with multiple units which would not be true with an individual unit.

[1] Prat-Daniel Corporation.

[2] "Testing Dust Collectors," by J. E. Watson, *Power*, November, 1939, pp. 68–70.

[3] "Dust-Collection Tests at New Power Plant of the Industrial Rayon Corporation," by C. B. McBride, *Combustion*, September, 1939, pp. 36–38.

Contributed by the Fuels Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Tests of Single- and Multiple-Cyclone Operation

To check the matter a test setup was prepared as shown in Fig. 1. The arrangement consisted of three commercial-size cyclones with a common inlet and outlet duct, and a common bin. At the bottom of each cyclone and within the bin, a threaded connection was made so that small individual bins could be attached to the cyclones, although they would still have common inlet and outlet ducts. These individual bins prevented any circulation between cyclones via the bin.

All cyclones were of Thermix design, the important feature of which is that each cyclone is equipped with a vertical damper at the inlet. Although in practice these vertical dampers are interconnected by an outside mechanism so that they are all in the same relative position, it was possible in connection with the second series of tests to be described to place them in variable positions so that one cyclone would handle a considerably greater quantity of gas than another. Thus the worst possible conditions were reproduced; hence, if any circulation were to occur with a resultant drop in collection efficiency, this fact would be noted.

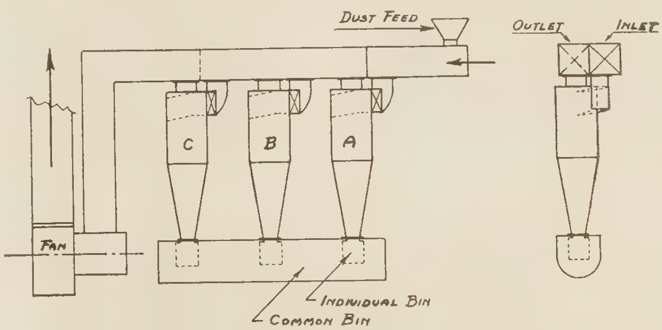

Fig. 1 Test Setup of Three Multicyclones in Parallel

The dust used for testing was collected by an electrostatic precipitator and was unusually fine due to some loss of coarser particles in the gas which escaped therefrom. The analysis of this dust follows: On 100 mesh, 2.86; through 100 on 200 mesh, 14.00; through 200 on 325 mesh, 10.38; through 325 mesh, 72.76.

No attempt was made to use hot gases, since a comparison in operation rather than an absolute determination of cyclone efficiency was to be made.

The dust, of which several barrels were received, was carefully mixed in a large bin in order to obtain a mixture that so far as possible would be uniform.

In the test, air was drawn through the cyclones at a predetermined draft loss by means of an induced-draft fan at the outlet. The dust was fed through a 15-ft duct to the immediate inlet of the cyclones at an approximate rate of 2.5 to 3.5 grains per cu ft of air. The feeding mechanism was a special hopper, equipped with screens and a rotating device, which supplied an even load of dust. The mixture of air and dust was passed through an egg-crate guide in order to even the flow into the inlet duct immediately above the cyclones.

Although, as stated, the dust was carefully mixed, it was felt

that it might still be possible for certain portions in the bin to be coarser than others. In order to diminish the chance of errors occurring from this cause, the tests were run first with a common and then with an individual bin, rather than entire series of tests with a common bin, and then again with an individual bin. After each test, dust settling on the bottom of the 15-ft inlet duct was stirred with an air lance and allowed to pass through the collector.

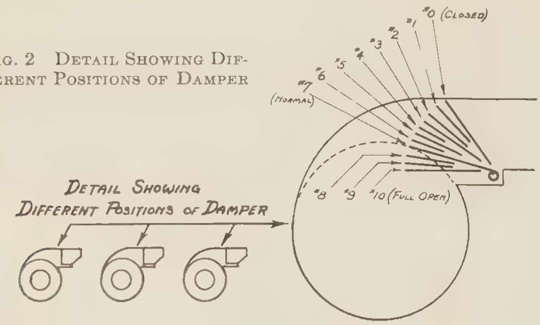

Fig. 2 Detail Showing Different Positions of Damper

As is shown in Fig. 2, the damper in the cyclone could be adjusted to ten different positions. Position 7 was tangential with the cylindrical body of the cyclone and was the maximum opening permissible to obtain good dust collection and similar to the normal design of the cyclone inlet with the damper omitted.

Conditions and Results of Tests

In the first series of tests, the results of which are given in Table 1, all three dampers were set in position 7. The draft loss of the cyclones was maintained at 2 in. There was a slight variation in temperature caused by a near-by furnace which was operating in the test laboratories. Also, in this first series, there was a variation in grain loading, due to the operation of the loader, which was not as uniform as desired. However, it is to be noted that the variation between 2 and 20 grains per cu ft does not noticeably affect the collection efficiency.[4]

The average collection efficiencies, using either the common bin or the individual bin, were within 0.56 per cent, which is within the limit of tolerance for experimental error. Therefore, it should not be assumed that the common bin was necessarily more effective in the operation than the individual bin. However, the results may be considered as conclusive proof that, with the dampers set in such a position as to obtain approximately the same static conditions in each cyclone, there is no decrease in collection efficiency when using the common bin.

The second series of tests was conducted with the dampers of the cyclones set in various positions with the express purpose of creating different static conditions in the different units. It must be pointed out that positions 8, 9, and 10, as well as 1 and 2, are not indicated for the highest percentage of collection possible with this design of cyclone. Therefore, it is to be expected that the collection efficiencies will be somewhat less than in a more normal range from positions 3 to 7. The practical object in having available positions 8, 9, and 10 is to make it possible to pass 25 per cent more gas through the units at position 10, and thus provide the capacity safety factor for temporary operating conditions which is desirable in connection with boiler-plant operation. In other words, without increasing the draft loss, it is thus made possible to reduce the number of cyclones to 80 per cent of what would otherwise be required.

In Table 2, the results are the more interesting in that there was such a wide variance in the quantity of gas handled by each cyclone without any appreciable difference in the collection efficiency. This table actually indicates 0.26 per cent better collection with a common bin, which is due to experimental error.

It is well known that a group of cyclones in parallel can be thrown out of balance statically so that recirculation and reduced efficiency will result. From this series of tests, it can be concluded that, by use of a common bin, considerable variation in positioning of the inlet-control dampers, with resulting variations in cyclone capacity, can take place with no decrease in collection.

Provided the individual cyclone tested is equipped with inlet-control dampers, and is of the size to be used in actual practice and operated under identical conditions of resistance and temperature, it may be concluded that it is representative of the performance of the group in the field.

[4] "Operating Variables of Cyclone Dust Collectors," by L. C. Whiton, Jr., *Chem. & Met. Engr.*, March, 1932, p. 150.

TABLE 1 TEST RESULTS ON THREE CYCLONES IN PARALLEL SERIES WITH DAMPERS SET UNIFORMLY IN POSITION 7

Test no.	Type bin	Cyclone resistance, in. w.g.	Temp, F	Duration of test, min	Total air, cu ft	Weight charged, oz	Loading, grains per cu ft	Weight collected, oz	Efficiency per cent
1	Common	2.0	85	20	69300	336	2.12	306.25	91.2
2	Individual	2.0	79	16	54900	317	2.52	291	91.8
3	Common	2.0	81	15	51600	311.5	2.64	281	90.2
4	Individual	2.0	79	16	54900	316	2.52	280	88.5
5	Common	2.0	85	20	69300	317	2.00	294.75	93.1
6	Individual	2.0	75	$10^1/_2$	35700	317	3.88	290.5	91.7
7	Common	2.0	77	9	30750	317.75	4.52	288.25	90.8
8	Individual	2.0	86	14	48600	314	2.82	282.5	90.0
9	Common	2.0	78	12	41100	316	3.36	291	92.2
10	Individual	1.94	85	14	47700	311.5	2.85	284.25	91.3
11	Common	2.0	77	$17^1/_4$	58950	318	2.36	286.5	90.1
12	Individual	2.0	76	27	92100	318	1.51	289	90.9
Avg	Common	2.0	..	..	...	....	2.833	....	91.26
Avg	Individual	1.99	..	..	...	....	2.683	....	90.7

TABLE 2 TEST RESULTS ON THREE CYCLONES IN PARALLEL SERIES WITH DAMPERS SET IN VARIOUS POSITIONS

Test no.	Type bin	Damper position —in cyclone— *A*	*B*	*C*	Cyclone resistance, in. w.g.	Temp, F	Duration of test, min	Total air, cu ft	Weight charged, oz	Loading, grains per cu ft	Weight collected, oz	Efficiency, per cent
101	Common	3	5	7	1.94	79	$15^1/_2$	52200	316.75	2.65	285	90.0
102	Individual	3	5	7	1.94	78	$14^1/_2$	48700	313	2.81	279.5	89.4
103	Common	7	5	3	1.94	83	$14^1/_2$	49100	316.5	2.82	287	90.7
104	Individual	7	5	3	2.0	82	13	44800	318	3.10	284.5	89.5
105	Common	2	5	8	2.0	76	$15^1/_4$	52000	315	2.65	276	87.7
106	Individual	2	5	8	2.0	77	14	47900	316	2.88	274.5	87.0
107	Common	1	5	9	2.0	81	$15^1/_4$	52400	316	2.64	272	86.1
108	Individual	1	5	9	2.0	82	$14^1/_2$	49900	316	2.77	270	85.5
109	Common	7	1	7	2.06	80	$15^1/_4$	52300	318	2.66	276	86.8
110	Individual	7	1	7	2.0	92	16	56200	318	2.47	282	88.7
Avg	Common	..	..	..	1.988	..	..	...	....	2.684	...	88.28
Avg	Individual	..	..	..	1.988	..	..	...	....	2.806	...	88.02

Discussion

H. H. Bubar.[5] Mr. Whiton's paper presents extremely interesting observations on the performance of an individual cyclone, tested under the conditions stated, as a fair criterion of the field operation of a multiple-cyclone unit.

From these observations it is deduced that the method outlined is more accurate than the commonly used sampling method of testing, because the possibility of experimental error in the sampling method is necessarily considerable. These observations are amplified by the statement that extremely accurate tests can be run on individual cyclones and that the extreme accuracy is partially due to the relatively small gas volume used.

It is believed the accuracy of these and other conclusions made in the discussion, are open to question. Nothing in the paper justifies the conclusions drawn. In the tests as outlined, both in arrangement of apparatus and method of conducting the tests, hypothetical points are established which do not conform to field conditions. No proof is offered that measurement of small gas volumes is more accurate than measurement of large gas volumes.

The error in testing is not experimental and it is not necessarily considerable. It is true that in a laboratory setup a given quantity of dust can be introduced into the gas stream and accurate data obtained. However, in field installations, the dust is already in the gas stream and the relative quantity must be determined. How otherwise than by the sampling method are the initial dust loading and gas volume to be obtained and checked in field installations?

It is stated that field tests on a group of multicyclones of the same design have shown corresponding accuracy with observations made on a single unit. While it is not stated how the field tests were made, this statement appears to be contradictory to the previous statement of "necessarily considerable" errors experienced in testing a multiple unit in the field.

It is stated that it is important to observe the operation of individual or multiple cyclones under precisely similar conditions. The results of the laboratory tests, which do not indicate conditions at all similar to field conditions, either in setup of apparatus or method of testing, are then presented as conclusive proof to the effect that testing an individual cyclone will give the same results as testing a multiple-cyclone installation.

As a further basis for the statement of conclusive results obtained, the screen analysis of the dust used is stated as being "unusually" fine, as the dust was obtained from an electrostatic precipitator. This unusually fine dust is stated to have 72.76 per cent through a 325-mesh screen. Recorded experience with dust from electrostatic precipitators on power plants would indicate this dust as being comparatively coarse, as such dust usually runs close to 90 per cent through a 325-mesh screen and often finer.

It is stated that it is well known that a group of cyclones in parallel can be thrown off balance statically so that recirculation and reduced efficiency will result. If such is the case, then any test of a single cyclone unit should be considered to be inconclusive and the standard sampling method of efficiency determination used to establish the over-all performance of a multicyclone collector.

It is stated that an egg-crate guide was used in order to even the flow into the inlet duct immediately above the cyclones. In field installations of any size, we have yet to find any case of an approximately even flow of inlet gas to dust-collecting apparatus.

It is stated that, although the dust was carefully mixed, it was felt that it might still be possible for certain portions in the bin to be coarser than others. In extracting samples from various parts of flue or breeching on field installations, how often does it occur that the dust in these various samples is of uniform screen analysis?

No draft differentials are given between various points in the apparatus to permit a check of operating conditions or to compute gas volumes accurately.

No dimensions are indicated whereby size of cyclones, ducts, common bins, etc., can be determined so as to establish velocities at different points in the apparatus.

In Fig. 1 a common bin is shown which does not conform in cross-sectional area with bins used in field installations. Because of this difference it is possible that air velocities and turbulences in this common test bin would be much higher than in the bins designed for field conditions. Assuming this possibility, then the draft differential in the test bin would be higher than that in the field installation, resulting in a definite variation between the test and field results.

Also, the results of tests on three cyclones set up in parallel are deemed conclusive proof of what would be secured with, say, fifteen cyclones set up in parallel. This is questionable.

The data submitted do not justify the conclusions drawn in the paper, either in the substitution of tests on a single collector for tests on an entire installation, or in the substitution of the method offered in preference to the sampling method as outlined in the tentative code of the A.S.M.E.

P. H. Hardie.[6] The author has presented interesting data on the controversial subject of whether the performance of a single unit of a multicyclone dust separator can be accepted as representative of the combined units. The data obtained on the experimental setup used would seem to prove that it can. However, such a limited investigation should not be accepted as final proof. Are three cyclones in parallel representative of forty, or even twelve? Did the methods used to create an unbalanced condition produce the same type of disturbance to proper operation as might be encountered in service?

The failure of some installations of multicyclones to duplicate the performance of a single unit tested individually is thought to be due to poor distribution of flow at the entrance to the cyclones or to the disturbances set up in the outlet duct at the points where the escaping gases leave the cyclones. Either, or both, of these conditions could cause unequal pressures in the different units which would result in the flow of dust-laden gas from the bin through some of the units to the outlet duct. This condition is somewhat similar to that which exists when a bin is not properly sealed.

The first group of tests would seem to indicate merely that, with these particular arrangements and dimensions of inlet duct, outlet duct, and bin, there is no difference in performance when operating with the common bin or with individual bins. In this connection it is noted that the flow in the inlet duct was straightened. The results would be more helpful if poorer rather than better distribution at entrance had been produced.

The conditions imposed during the second group of tests were probably more severe than would be encountered in actual practice. The results so obtained are highly informative; but does unequal flow through the different units, when produced by setting the dampers in different positions, reproduce the same type of unbalancing as sometimes exists when the dampers are in the same position?

Even if we accept the author's conclusion that one cyclone can be used to determine the performance of a large number of similar

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[6] Research Engineer, Department, Technical Development and Research, Consolidated Edison Company of New York, Inc., Brooklyn, N. Y. Mem. A.S.M.E.

ones in parallel, there would seem to be good reasons for preferring the over-all test after installation as proposed in the A.S.M.E. Power Test Code for Dust-Separating Apparatus. The apparent advantages of the single-unit test materialize only if the tests can be carried out in the manufacturer's testing laboratory. This means that a large representative sample of the dust must be collected and shipped to the factory. The collection of this dust is in itself almost as much of a job as running the over-all test. Even if a similar steam-generating unit, equipped with an electrostatic precipitator, is available from which a representative sample of dust can easily be obtained, there still remains the problem of reproducing the original finely divided mixture of dust and gas in the laboratory.

Dust particles, especially the very fine ones, tend to stick together once the dust has been collected and allowed to pack. By very fine particles is meant those which are considerably smaller than 44 mu (325 mesh). It is in this range of particle sizes that the efficiency by size of cyclone separators plunges toward zero as the particle size is reduced. Above 44 mu 100 per cent efficiency is easily attained. To the writer's knowledge no satisfactory means has been devised for separating these conglomerate particles and reproducing the original finely divided mixture of suspended dust. When using an elutriator for size analysis, it takes many hours and often days of operation before all of these conglomerate particles are finally broken up. Therefore, the greater accuracy of measurement obtained when testing a single cyclone loses its significance when there is no certainty that the dust-gas mixture being used is truly representative.

H. E. Macomber.[7] The author has proposed the idea that the performance of a dust-collector installation in the field, consisting of a number of collecting units operating in parallel, each of essentially identical size and design, will duplicate the individual performance of any one of the units of the assembly, or of a similar unit outside the assembly. The data presented for the conditions stated in the paper bear support to the idea.

The writer, however, looking at the problem from the viewpoint of the user of dust-collection equipment, wishes to bring the following thoughts into the picture:

1 A shop test or a field test of an individual unit should not supplant a field test of the entire equipment in position, even though the field test may be more difficult of attainment as to a comparable degree of accuracy. That is, the purchaser wants to know the degree of performance he is receiving from the assembled apparatus in his plant, regardless of the performance similar equipment may have shown to be possible in the shop, or which might be expected following the results obtained from tests of an individual unit of like size and design in the field.

2 Conditions under which dust-collection equipment is actually required to operate in some instances may vary considerably from day to day, as well as from the conditions previously set up as the typical basis upon which the guarantee is computed.

3 Further, every dust collector is affected to some extent by variation in the density and size distribution of the dust particles, volumes of gas or air to be passed, arrangement of ducting preceding the collector, etc.

Thus, briefly, the situation as it concerns the ability of a collector to meet a specified value of removal efficiency seems to be that, generally, the lower the value of the design-point efficiency, the greater will be the deviation from that value if the actual dust sizing or consistency as found is such that it contains an increased or decreased fraction of fines.

The point which the writer wishes to emphasize is that, in the purchase of a collector, the field test, which would probably accompany its operation before acceptance apparently assumes greater importance as the guaranteed-performance ability of the collector to be purchased is decreased.

4 Even so, some manufacturers may state a guarantee on the basis of total collection efficiency and pressure loss for a typical condition, without regard to possible variation in field conditions, and some purchasers may buy the equipment with the understanding that the guarantee value of total collection efficiency will be maintained regardless of field conditions. On the other hand, other manufacturers definitely state any guarantee of total collection to be subject to adjustment according to the actual conditions and make-up of the gas- or air-borne dust encountered in the field.

Considering the points here outlined, it is the writer's opinion that, although the performance of an individual cyclone unit was indicative of the performance of multicyclones, when handling caught dust under the stated conditions, he would not, as a user, be satisfied that the results so determined should be considered as applicable to the performance of field-assembly installations in position.

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R. F. O'Mara.[8] Since the subject of this paper is one in which we have been interested and on which we have been carrying a continuing research program over the last 12 years, we feel that the tests as discussed represent a good beginning of such an investigation, but that they are quite academic and too sketchy to justify the broad conclusions which have been drawn. On the whole, however, our many tests both in the laboratory and in the field substantiate the broad conclusions outlined in the paper.

In carrying work from the laboratory to the field or when making tests on an installation in the field on which to base selection of the final equipment, it has been our experience that the same size collector in physical dimensions as is to be used in the final installation must be tested and that, if this is done, the results from a single tube can be reasonably well checked with an installation consisting of a large number of identical small tubes. The following results are taken from check tests on a single tube followed later by actual installations:

Ventilating Air From Clinker Grinders

Preliminary Test. One 9-in. tube; efficiency 93.8 per cent at 2.8 in. pressure drop

Installation. 18 tubes; efficiency 92.4 per cent at 2.9 in. pressure drop

Asphalt-Mixing Plant

Preliminary Test. One 9-in. tube; efficiency 94.7 per cent at 2 in. pressure drop

Installation. 20 tubes; efficiency 94.5 per cent at 3 in. pressure drop

Pulverized-Coal-Fired Boiler

Preliminary Test. One 9-in. tube; efficiency 85 per cent at 2 in. pressure drop

Installation. One hundred and eighty 9-in. tubes; efficiency 85.6 per cent at 2 in. pressure drop

Cement-Kiln Gases

Preliminary Test. One 9-in. tube; efficiency 86.6 per cent at 3.3-in. pressure drop

Installation. Two hundred and eighty-eight $10^1/_4$-in. tubes; efficiency 85.5 per cent at 4 in. pressure drop

Gases From Fuller's-Earth Treating Furnace

Preliminary Test. One 6-in. tube; efficiency 92.8 per cent at 2 in. pressure drop

Installation. Thirty-six 6-in. tubes; efficiency 92.5 per cent at 2 in. pressure drop

[8] Western Precipitation Corporation, Los Angeles, Calif. Mem. A.S.M.E.

In multiple operation of cyclonic separators study of the header design to provide good distribution both of the gases and the suspended solids is equally as important as the hopper construction.

The discussion in the paper regarding sampling technique indicates lack of knowledge or investigation of the refinements in sampling technique that have been developed in other industries, particularly the metallurgical industry, both for ferrous and nonferrous metals. Here, guarantees are required on recovery of gold, silver, and precious metals from smelter gases and 1 per cent more or less in efficiency may mean losses of thousands of dollars a year. Efficiencies are not based on particle size but on a basis of the percentage recovery of the actual values in the escaping dust. Ofttimes these values lie in the finer rather than in the coarser material.

Power-plant engineers could take a leaf from the metallurgical engineers' refinements in sampling techniques for gases carrying suspended matter in the solid or liquid state. The following bibliography is given on the measurement of suspended solids and gases:

BIBLIOGRAPHY

1 "Atmospheric Particulate Matter; Dust With an Apparatus for Its Determination," by Philip Drinker, R. M. Thomson, and S. M. Fitchet, *Journal of Industrial Hygiene*, vol. 5, 1923, pp. 19, 62, and 162.

2 Papers on Smelter Smoke, by W. D. Harkens and R. E. Swain, *Journal, American Chemical Society*, vol. 29, 1907, pp. 970–998.

3 "Determination of Dust in Blast-Furnace Gas," by W. Brady and L. A. Touzalin, *Industrial and Engineering Chemistry*, vol. 3, 1911, pp. 662–670.

4 "Messung von Flugstaub in Rauchgasen," by E. Zimmerman, *Zeit. V.D.I.*, vol. 75, 1931, pp. 481–486.

5 "Ein neuer Apparat zur Bestimmung des Staubs und Wassergehalts in Abgasen," by J. Simon, *Stahl und Eisen*, vol. 25, 1905, p. 1069.

6 "A New Method for the Determination of Dust," by W. Allner, *Braunkohle*, vol. 24, 1925, pp. 378–383.

7 "Determination of SO_2 and SO_3 in Flue Gases," by R. J. Nestell and E. Anderson, *Industrial and Engineering Chemistry*, vol. 8, 1916, pp. 258–260.

8 "Determination of Suspended Matter in Gases by Collection on Filter Paper," by S. H. Katz and G. W. Smith, U. S. Bureau of Mines Reports of Investigations, Serial 2378, 1922.

J. M. DallaValle.[9] The characteristics and limitations of multicyclones have never been treated comprehensively. This has been due in part to their relatively recent introduction in the field of dust collection, and perhaps also to inability to generalize performance in terms of tests carried out with a so-called "standard" dust. Mathematically, the behavior of a particle in a cyclone is easily determined, but anyone having even slight experience with particulate matter knows well that its true dynamical behavior depends upon other considerations than a knowledge of accelerating forces alone. A discussion of some of the factors involved in the development of an ideal cyclone was presented by Lissman[10] several years ago. In particular, Mr. Lissman discusses the design arrangements necessary to reduce turbulence in the dustbin in order to assure a minimum of interference with other cyclones (in parallel) connected to it. This paper emphasizes what is actually achieved in this connection under operating conditions.

The fact that a common dustbin with cyclone control dampers set in various positions does not materially affect the efficiency of dust collection (as compared with individual bins) is heartening to dust engineers who have heretofore possessed no definite information in this connection and in fact have often questioned its use. More striking than this, however, is the fact that changes in the control damper setting in no way appreciably affect the collection efficiency of the cyclone. This, as the author notes, is an important consideration in boiler plants equipped with multicyclone collectors where air volumes handled vary greatly, but draft losses must not be increased.

The writer expresses the hope that the author and others interested in the performance of multicyclones will make available their knowledge of the behavior of other dusts than fly ash, and perhaps decide upon a uniform performance test to be used in the laboratory. The testing of equipment after instead of before installation is not only costly to all parties concerned in event its performance is unsatisfactory, but also confuses the prospective purchaser who as a rule cannot judge the merits of various designs tested in a score of different ways.

[9] In charge, Environmental Sanitation Section, U. S. Public Health Service, Bethesda, Md.

[10] "An Analysis of Mechanical Methods of Dust Collection," by M. A. Lissman, *Chemical and Metallurgical Engineering*, vol. 37, 1930, pp. 630–634.

Author's Closure

Mr. Bubar's observations appear to be mainly intent upon the necessity of dust sampling in the field in preference to testing one cyclone.

It was not the intent in this research to make such comparisons, and it might be stated that the author is highly in favor of field test. It was intended in this research to give the results in a specific instance of testing one cyclone or testing three cyclones under accurate test conditions. If all of the information supplemental to this simple series of observations were included, as Mr. Bubar feels would be of interest, the amount of data presented in this report would have been far beyond its intended scope, or the time allotted for the presentation.

Mr. Macomber's observations also bring out the point concerning field tests compared to single cyclone tests. As observed previously, this was not the intent of the report.

The author thoroughly agrees with the point brought out by Mr. Macomber that an adjustment should be included in guarantees for variations in dust fineness that may be encountered in practice, over that assumed before the installation is installed. The author's company, however, has had some difficulty in getting people to agree to such a plan, and consequently in general has offered lower operating guarantees than would be expected from their own experience, in order to cover such contingencies as mentioned by Mr. Macomber.

Mr. O'Mara's observations appear to bear out the conclusions of the report, and it is interesting to have such confirmation from another manufacturer.

Mr. Hardie has brought out some important points which add considerably to the information contained in the original paper.

He asks perfectly correctly whether it may not be possible that three cyclones with a common hopper operate successfully, whereas 40, or even 12, may not. This is a very just observation, and the author cannot say from experience whether, for example, 40, cyclones would operate the same as the 3 in this test. In commercial practice, however, the number of cyclones connected with one hopper is necessarily limited. In view of the results of this test, therefore, any group connected with the hopper may be considered the unit. Four to nine cyclones are generally used per hopper. In one instance we know of twenty 2-foot cyclones per hopper, but this was unusual and exceptional, and is not to be recommended, especially as the hopper is extremely deep in such an instance because of the necessary angles of its sides.

Mr. Hardie states that tests of some installations of parallel operated cyclones do not always duplicate the performance of single units tested individually. There have been frequent occasions to compare field tests with tests of a single cyclone, and as far as can be observed, the rule is that they show a similar collection, and it is rather the exception that they do not. In such

instances this may be due to duct layout, the settling of dust in the flues, and various other factors which sometimes make it difficult to approach the ideal from the standpoint of field testing, even with the greatest of skill being exercised.

As Mr. Hardie points out, the variable positioning of the cyclone dampers in the second series of tests is a condition more severe than in actual practice, but questions whether or not this reproduces the condition of different static conditions being created in the flue connecting a series of cyclones. He is probably correct that it is not precisely comparable. However, it has been the custom of the author's company to connect the inlets and outlets of the cyclones to large rectangular overhead ducts that are practically plenum chambers themselves. Many static readings have been taken at various parts of such ducts, and it has been found that the static condition is practically uniform throughout. It is, therefore, probable that the eccentric positioning of dampers in the test observations is considerably more severe than is ecountered in practice.

The author agrees with Mr. Hardie that it is preferable to operate a field test in accordance with the A.S.M.E. proposed code. The only advantage of utilizing a single unit test is that considerable information can be obtained by a prospective user, before the final installation is made, by attaching such a unit directly to his flue. Furthermore, cases may present themselves where it is an impossibility to obtain a field test because of the arrangement of ducts, the settling of dust in the flues, and other factors. In such an instance a single unit test might be valuable.

Testing of dust in the manufacturer's plant necessarily requires dust that has been previously collected. It has been found that if dust from the electrostatic precipitator is used, it is generally somewhat finer than the dust in the gas itself, and such dust appears to give about the same results as when a unit is hooked up directly to the flue. It may be that the fact that it is harder to collect, because of its greater fineness, is compensated for by some conglomeration.

In a series of tests run at the Stamford Gas and Electric Company a cyclone was attached to the flue. At 2 in. draft loss and an average of 289 deg an over-all collection of 88.1 per cent was obtained with pulverized-fuel dust. Dust was then taken from the electrostatic precipitator at the same plant, which analyzed 77.9 per cent through 325 mesh, and, at an average of 2.6 in. draft loss, 91.3 per cent was the efficiency with the same cyclone previously used at Stamford. The draft loss was somewhat higher; therefore a better collection should be expected. Furthermore, the gas temperature was atmospheric; hence about 220 deg less than at Stamford, which would further increase efficiency.

A further series of tests was then run in the research plant, using hot flue gas to carry the electrostatic dust. At an average draft loss of 2.1 in., and an average temperature of 433 deg, an average collection of 90.1 per cent was obtained with slightly finer dust, which averaged 80.01 per cent through 325 mesh. These tests are given primarily because they are on the same plant with the same dust, and might further be compared with seven hour tests, extending over 14 consecutive days, at the plant of the Industrial Rayon Corporation,[11] which had 30 cyclones of the same design per boiler. Here the draft loss was less, namely, 1.45 in., the temperature averaged 345 deg, and the collection showed 88.4 per cent. The screen analysis of the dust in the flue was especially fine, however, being an average of 88.6 per cent through a 325-mesh screen. The reason for this was that the boiler which was designed for 90,000 lb of steam per hour was averaging 52,000 lb per hour during these tests.

The author believes, therefore, that although the research reported in the original paper is limited necessarily, the observations elsewhere seem to bear out the conclusions with the particular cyclone tested.

[11] "Dust Collection Tests at New Power Plant of Industrial Rayon Corporation," by C. B. McBride, *Combustion*, September, 1939, pp. 36 38.

Penstocks for the Grand Coulee Dam

By P. J. Bier,[1] Denver, Colo.

The ultimate power development for the Grand Coulee Dam includes eighteen main units of 150,000 hp each and three station-service units of 14,000 hp. Individual penstocks of 18 ft diam were provided for the main units and of 6 ft diam for the station-service units. The penstocks are of welded-plate steel construction, fabricated and X-rayed in accordance with the API-ASME Code. The 18-ft penstocks were fabricated in a plant erected near the dam, and the 6-ft penstocks were produced in the subcontractor's shop. All penstocks were installed in octagonal tunnels provided for that purpose through the dam, and were embedded in reinforced concrete after installation.

General Description

The penstocks for the Grand Coulee Dam on the Columbia Basin project, Washington, were provided for the release of water under pressure for the production of power. The total estimated flow through the penstocks, at a rated head of 330 ft for the turbines, is approximately 82,000 sec-ft, based on the ultimate development of eighteen main power units and three station-service units. The main-unit penstocks are 18 ft in diam, serving turbines of 150,000 hp each, and the station-service penstocks are 6 ft in diam, serving turbines of 14,000 hp each, making a total ultimate capacity of 2,742,000 hp. The generating equipment will be installed in two separate buildings, one on the east bank of the river called the right powerhouse, and one on the west bank of the river called the left powerhouse. The two powerhouses are located at the downstream toe of the dam, and will contain nine main power units each, the left powerhouse including, in addition, the three station-service units.

All penstock intakes will be protected by trash racks arranged in semicircular form around the openings, and reaching up to within 21 ft of the crest of the dam. Hydraulically operated coaster gates will be installed on the face of the dam to shut off the flow from any of the main-unit penstock intakes until the turbines have been installed. The upstream ends of the main-unit penstocks were provided with hemispherical bulkheads to exclude the reservoir water from the penstocks during construction, and until the turbines are installed. After the installation, the intake will be closed with the coaster gate, and dewatered through a drain valve in the bulkhead, which latter is then flame-cut into small pieces and removed through the manhole.

Installation and Contracts

While the initial installation includes only three main units and two station-service units, the penstocks were installed for the full development. The Grand Coulee Dam is being built in two stages and under two separate contracts. The first contract included the foundation to an approximate elevation of 1000, and the second contract calls for the completion of the dam to elevation 1311. As it was not feasible to install the penstocks under the first contract, tunnels were provided which permitted the installation of the penstocks at a later date. These tunnels were of octagonal shape, 24 ft in size for the main-unit penstocks and 12 ft for the station-service penstocks. They were reinforced to resist the loads on the dam due to its dead weight and live water load, and for temperature effect.

The tunnels were started under the first contract, and completed to the upstream face of the dam under the second contract. The contract for the fabrication and installation of the penstocks, exclusive of the installation of the upstream sections containing the bulkheads, was awarded to the Western Pipe and Steel Company of San Francisco, at a bid price of $1,456,624 which included twelve 14-ft-diam pump inlet pipes. This sum, however, did not include the cost of transporting the pipes and pipe materials to the fabricating plant and to the dam. The estimated weight of all pipes furnished under the contract was approximately 8300 tons. The contractor sublet the fabrication of the station-service penstocks and the preparation and rolling of the plates for the main-unit penstocks to the Chicago Bridge and Iron Company.

The 18-ft-diam penstock, because of the impossibility of shipping pipe of this size, made shop fabrication impracticable, and it was necessary to erect a field-fabricating plant near the dam. Erection of this plant was started in April, 1938, and shop fabrication was completed in March, 1940. The installation of the penstocks was completed in May, 1940.

Design of Penstocks

Main-Unit Penstocks

The penstocks are on a slope of $20^1/_2$ deg, starting with elevation 1041 at the upstream face of the dam and dropping down to elevation 938 at the turbine inlet, as shown in Figs. 1 and 2. The penstocks have an inside diameter of 18 ft which is reduced to 15 ft with a reducing bend near the turbine. The 15-ft-diam end section is connected to the turbine-scroll case with a double-acting expansion joint, as shown in Fig. 3, which will permit both axial and lateral displacements of the turbine in relation to the penstock. Axial displacements will be caused by temperature changes in the pipe and turbine, and lateral displacements by deflections of the dam and deformation of the foundation.

The inlet transitions are 30 ft long, and were designed to reduce inlet losses. They were formed in the concrete, and are provided with 30-in. air pipes, to facilitate the escape of entrained air when in operation or filling, and the inflow of free air when draining the penstock. The air pipe will also serve as a vent when the penstock is empty.

Each penstock is 329 ft long from the face of the dam to the scroll case, and is embedded in the tunnel backfill except for 10 ft around the expansion joint, where an access chamber was provided for the maintenance of the joint packing. This chamber can be reached by a passageway from the powerhouse. The penstocks were designed for a maximum head of 420 ft, including a static head of 360 ft and a water-hammer head of 60 ft. The head due to water hammer was computed in accordance with the rigid and elastic water-column theories on a basis of a minimum closure time of 4 sec at full gate, the turbine specifications calling for governors adjustable to a rate of turbine-gate movement of from 4 to 12 sec for a full-closing stroke.

The design of the penstocks was based on the use of firebox-quality steel plate, grade B, A.S.T.M. designation A-89, which has a minimum yield point of 27,000 lb, a minimum tensile

[1] Senior Engineer, Bureau of Reclamation, United States Department of the Interior. Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., Sept. 3–6, 1940, of The American Society of Mechanical Engineers.

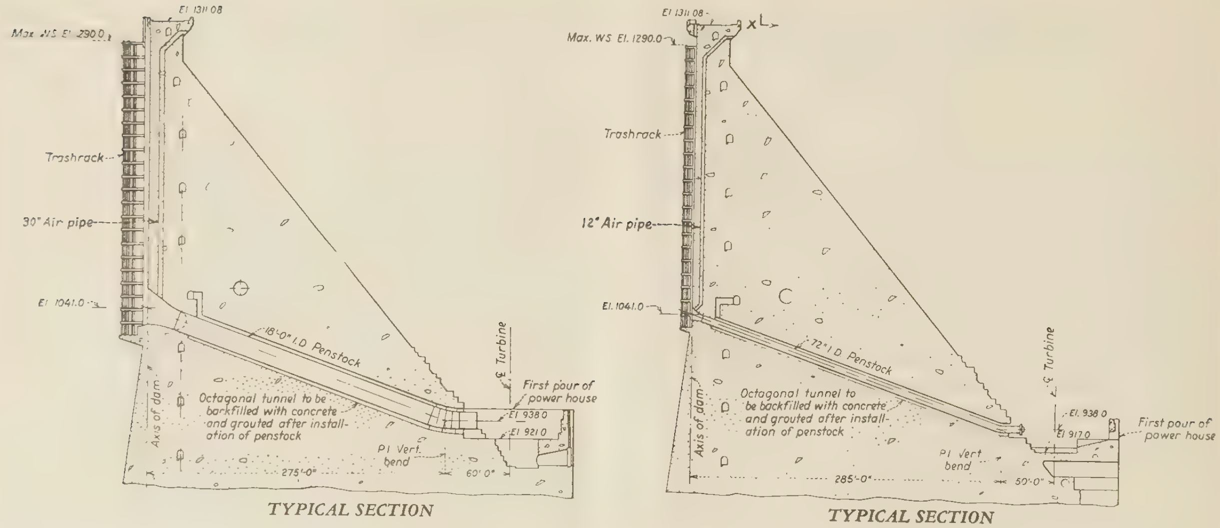

FIG. 1 PROFILES OF DAM AND PENSTOCKS

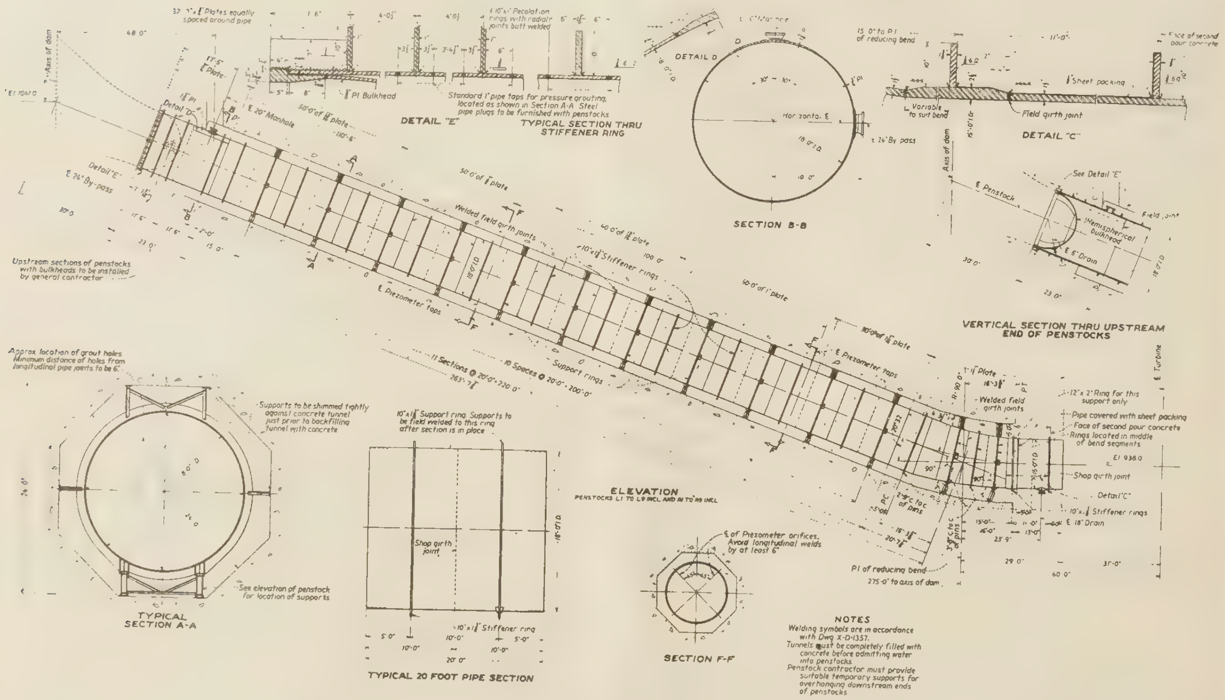

FIG. 2 DESIGN OF 18-FT PENSTOCKS

strength of 50,000 psi, and an elongation in 2 in. of 32 per cent. This steel was selected because of good welding qualities due to a low carbon content (0.20 to 0.22 per cent), which reduces the tendency of air hardening and produces ductile welds. Shocks due to water hammer and surge waves induce impact stresses which may be the cause of failure, especially in penstocks with brittle welds. With the use of this ductile steel, and due to the fact that the penstocks were to be embedded in reinforced concrete, thermal stress relieving of the pipe sections was not considered necessary. Radiographic examination of the welded joints, on the other hand, seemed desirable because it provides an effective check on the quality of welding, and permits an increase in joint efficiency from 80 per cent to 90 per cent, in accordance with the API-ASME Code for Fusion-Welded Unfired Pressure Vessels, which was adopted for this work. As an additional safeguard, a hydrostatic-pressure test was specified for all completed pipe sections.

Several schemes of installation were studied, and the scheme adopted consisted of embedding the penstocks in the tunnel backfill and painting the outside of the pipe and stiffeners to prevent a bond between the steel and concrete, thus permitting the pipe to "breathe" between empty and pressure conditions. A design stress equal to two thirds of the yield point was used for the fully embedded pipe having ample concrete coverage and

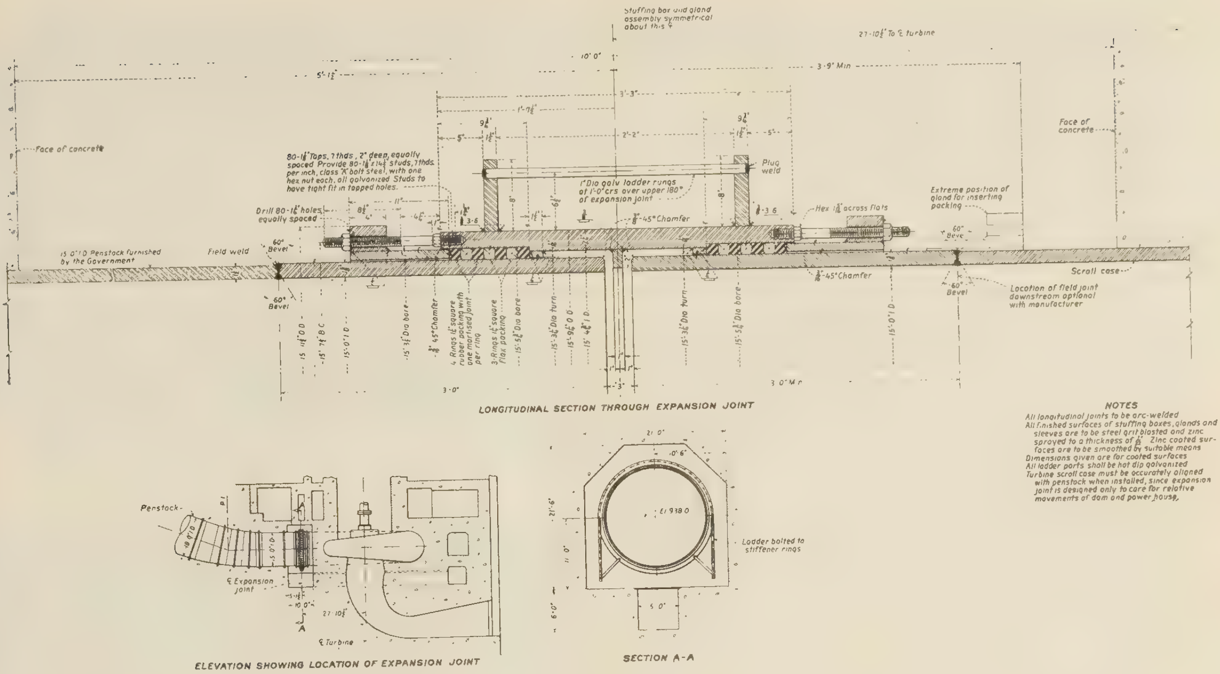

Fig. 3 Expansion Joint 15 Ft Diam

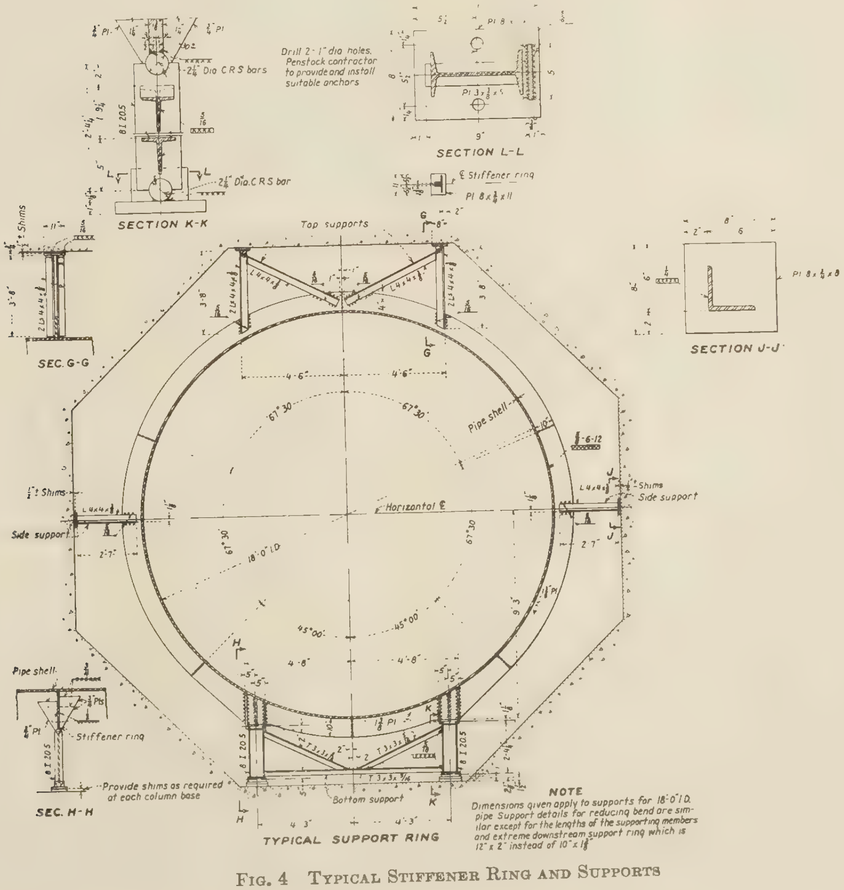

Fig. 4 Typical Stiffener Ring and Supports

reinforcement, and one half of the yield point for the pipe near the downstream toe of the dam where the concrete cover was not considered sufficient to aid the shell in resisting the internal pressure. The shell thickness was computed at $^3/_4$ in. at the upper end increasing by $^1/_{16}$-in. increments to $1^1/_{16}$ in. at the lower end of the tangent, and to $1^1/_2$ in. in the bend and lower end section.

Each penstock was provided with three percolation rings near the upstream end to retard the seepage of water along the outside of the shell. Single stiffener rings, $1^3/_8$ in. thick × 10 in. high, were welded onto the shell every 10 ft to maintain the circularity of the pipe section during fabrication, handling, and installation. Structural-steel supports, as shown in Fig. 4, were provided at every other stiffener ring, at the bottom and the top, the former to support the weight of the empty pipe during installation and the latter to protect the pipe against uplift

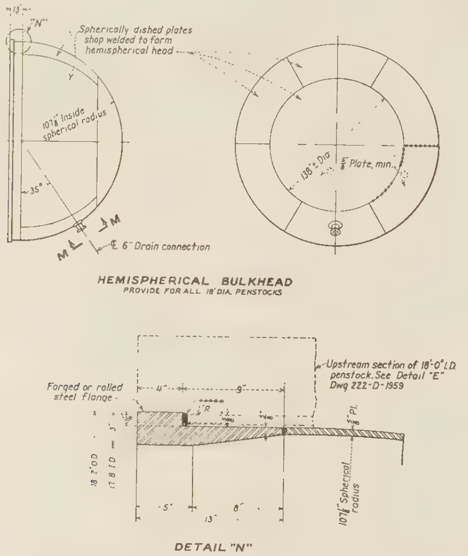

Fig. 5 Hemispherical Bulkhead for 18-Ft Penstocks

during concreting in the tunnel. Lateral struts were provided at the supporting points to maintain the pipe in alignment while concreting. The lower supports were pinned at top and bottom to permit temperature movements in the line during erection.

The reducing bend was designed with a long radius and small deflection angles to reduce turbulence and head losses. A 24-in-diam by-pass pipe was connected to the upstream end of each penstock to provide means for filling the penstock, and to make it possible to operate the coaster gates under balanced pressure. The by-pass lines take water from the reservoir through a grated inlet and are controlled with double valves located in valve chambers near the 1050-ft gallery. The penstock is made accessible with a 20-in-diam manhole, placed on top of the pipe near the upstream end, which can be reached through a shaft from the gallery. Each penstock can be drained through an 18-in. drain at the downstream end of the pipe. The manhole and by-pass openings were reinforced to compensate for the loss in metal cut from the shell. The reinforcement consists of a plate ring welded to the shell, the nozzle being made from a pipe nipple and welding-neck flange. The manhole opening was streamlined with a plug welded from light-gage steel provided with drain holes to prevent unbalanced pressures. As the manhole is located in an open shaft, the pipe shell around the concrete will be subjected to secondary bending stresses; therefore, it was increased in thickness by inserting a $1^5/_8$-in. plate for the length of the course and for one sixth of the circumference.

The bulkheads placed in the upstream sections are hemispherical, and were welded from spherically formed plates and a central closing plate, as shown in Fig. 5. Special flanges were welded to the bulkhead shells for attachment to the upstream-pipe sections. These flanges were designed to resist the bending stresses due to the deformation of the shell when under pressure. The pipe shell opposite the flanges was stiffened with a number of ribs placed radially along the outside diameter. The hemispherical shape was selected as being the most efficient from the standpoint of stress, also because ellipsoidal heads for this diameter were not available, and could not be transported in one piece. The bulkheads were designed for a static head of 270 ft, requiring a $^1/_2$-in. shell, which was increased to $^5/_8$ in. to allow for corrosion and to provide additional safety in view of the importance of this water stop.

Two sets of piezometric orifices of four each, located 100 ft apart, were provided for the purpose of performing flow and pressure tests. The stainless-steel orifices were placed at quarter points of the shell. Individual lines of $^3/_4$-in. extra-heavy steel pipe were run from each orifice to the terminal box in the powerhouse, with one static-pressure line from the reservoir to the terminal. Grout holes consisting of 1-in. pipe taps, were provided in the penstocks, for the purpose of back-grouting and consolidating the concrete backfill. They were placed on the inside of the pipe at horizontal and vertical centers, on each side of the stiffeners and percolation rings.

Station-Service Penstocks

The station-service penstocks are similar in design to the main-unit penstocks. The intake is located at elevation 1041, and the penstocks slope toward the turbines on an angle of about 20 deg, as shown in Fig. 1. Each turbine will be protected with a ring-seal gate. Penstock No. S3, which is to serve the future turbine, is provided with a welded bulkhead at the downstream end. A bulkhead gate will be used on the upstream face to close any of the three intakes when the turbines and ring-seal gates are installed. The three intakes were closed with temporary timber bulkheads before the reservoir level reached the intake elevation during construction. Between the lower ends of the penstocks and the ring-seal gates, double-acting expansion joints are installed, for similar reasons as for the main-unit penstocks.

The penstocks have a uniform diameter of 6 ft and are 332 ft long, from the face of the dam to the ring-seal gate. They were designed for a total head, including water hammer, of 400 ft, which resulted in plate thicknesses of $^3/_8$ in. in the tunnel and $^9/_{16}$ in. at the downstream end near the toe of the dam. The penstocks were fabricated in the subcontractor's Birmingham plant, in 20- and 40-ft lengths, ready for installation in the tunnels. Each penstock was provided with a manhole-and-drain connection, also with piezometer orifices, grout holes, percolation rings, stiffeners, and supports, similar to the 18-ft-diam penstocks.

FABRICATION OF MAIN-UNIT PENSTOCKS

Field-Fabricating Plant

The field-fabricating plant was located near the construction railroad in a settlement called Electric City, located about $3^1/_2$ miles from the dam. The plant consisted of a storage yard for incoming pipe materials, a welding shop, and a yard for testing, painting, and the storage of completed pipe sections. The welding shop was of frame and corrugated-steel construction, 70 ft wide, 260 ft long, and 45 ft high under the roof trusses. Adjoining the shop building on one side was an annex containing transformers and shop equipment, also offices for the shop superintendent and the government inspectors; on the other side of the building was an office for the X-ray technicians, and a de-

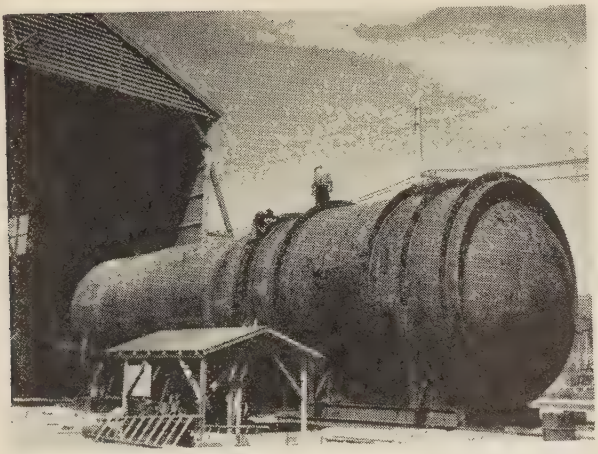

FIG. 6 TESTING MACHINE FOR MAIN-UNIT PENSTOCKS

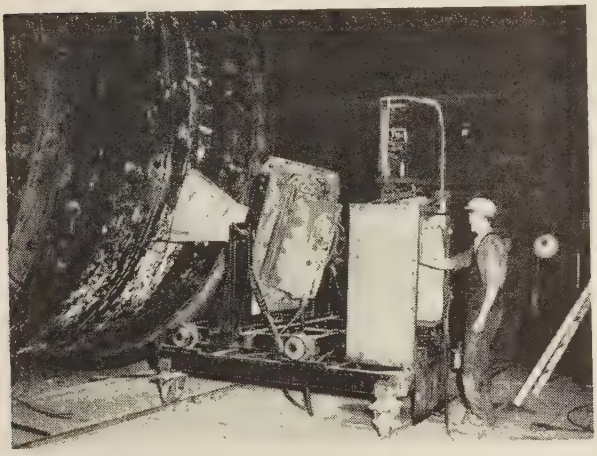

FIG. 7 PORTABLE X-RAY APPARATUS USED IN FABRICATING PLANT

veloping room for the X-ray films. The entire plant, including yards and shop, was served by a 35-ton gantry crane having a span of 66 ft 6 in. and a runway of about 1100 ft.

SHOP EQUIPMENT

The shop equipment consisted of four setting-up bases for the fitting and welding of the 18-ft pipe shell and stiffeners. Each base consisted of a series of concrete piers arranged in a circle and provided either with level steel platforms or with steel brackets or telescoping pipes on each pier, depending upon the fitting work to be done. A bulldozer, supported on a structural-steel traveler, was used to hold the two halves of the pipe or the pipe and stiffener together when tack-welding preparatory to the final welding. A motor-driven trolley and a roller base were installed for the movement and rolling of the pipe, during the welding of the longitudinal and circumferential joints, respectively. For the automatic-welding work, a welding machine was used which was supported from a structural-steel tower. A specially built testing machine, as shown in Fig. 6, was provided for the hydrostatic-pressure tests. The X-ray apparatus used is shown in Fig. 7; it was of portable construction with a capacity of 200 kv at 8 ma. A number of adjustable, rounding-out spiders, made of round bars with a central plate, were used for the pipe sections during construction and until the stiffeners were welded on. The auxiliary equipment in the shop annex included an air compressor and receiver, a lathe, planer, milling machine, drill press, and a Riehle testing machine.

SHOP PROCEDURE

The 18-ft penstocks were fabricated in sections or erection lengths of 20 ft, weighing up to 28 tons each, except the upstream-end sections containing the hemispherical bulkheads, which were produced in 23-ft lengths, weighing about 37 tons each. The pipes were made from two 120-in-wide plates which were beveled for welding, and rolled in the subcontractor's Chicago shop, then shipped to the field-fabricating plant for completion. The fabrication procedure for a 20-ft pipe section was as follows:

1 Two rolled plates were set up and aligned on one of the fitting bases and tack-welded into a cylinder.

2 A rounding-out spider was inserted into the 10-ft tack-welded pipe course, which then was taken to the trolley under the automatic welding machine, and welded along the longitudinal seam.

3 The stiffener rings, having been flame-cut in six segments from $1^3/_8$-in. plates and assembled on a special platform, were fitted to the 10-ft pipe course. Intermittent fillet welds, 6 in. long and 12 in. on centers, alternated on both sides, were used between the pipe and the stiffener ring.

4 Two 10-ft pipe courses were tack-welded together, then placed on the roller base, and completely welded on the automatic machine.

5 The welded seams were radiographed in 16-in. sections, marked along the joint. If no welding defects were disclosed in the films, the pipe section was ready for the pressure test.

6 The pipe section, after passing the visual inspection and X-ray examination, was placed in the testing machine, and subjected to a hydrostatic-pressure test of about 150 per cent of the design pressure, stressing the shell to approximately 21,000 psi. The test pressure was applied and relieved three successive times, and held a sufficient length of time to permit a thorough inspection of all joints for signs of leakage or failure. This pressure test proved its value as cracks from 4 to $9^1/_2$ in. long were discovered in five pipe sections, which apparently eluded detection by X-raying. Four of the cracks were near the junction of the longitudinal and girth seams where stress concentrations due to welding are more prevalent.

7 After the pressure test, the pipe section, with the exception of areas within 10 in. of the joints to be welded, was cleaned on the outside of all rust, dirt, grease, and loose scale, and painted with two coats of coal-tar paint applied cold. After the paint was dry, the section was ready for transport to the dam, and erection in the tunnels.

WELDING PROCESS

The contractor used the single-pass process of welding for all machine welds in the pipe shell, for plate thicknesses up to and including $1^1/_{16}$ in.; for all heavier plates, manual welding was used. Single-V welding grooves were used for the automatic welds, and double-V grooves for the manual welds. In the automatic welding process, the joint was welded in one pass on the outside of the shell, the welding head remaining stationary while the pipe was moved along the trolley for the longitudinal seam or rotated on the roller base for the circumferential seam. During welding, a water-cooled grooved copper backing-up bar was pressed against the underside of the shell opposite the welding head by means of a bracket attached to a pipe post. The flux was deposited ahead of the welding operation and a coiled, bare electrode was fed in automatically, the welding speed and electric current having been regulated to suit the plate thickness. The lower layer of the flux covering the joint was fused into solid slag which protected the deposited weld metal against

the atmosphere, and effected some annealing of the weld. Preparatory to making a longitudinal-seam weld, small plates of the same thickness as the shell were welded to both ends of the pipe course to form extensions of the welding seam. The starting plate, about 6 in. square, was in one piece and the finishing plate in two pieces, each $4^1/_2$ in. wide $\times$ $7^1/_2$ in. long, both beveled for welding and tacked together at the bottom of the groove. By this means, the tendency to crack, especially at the finishing ends of the longitudinal joints, was practically eliminated. After

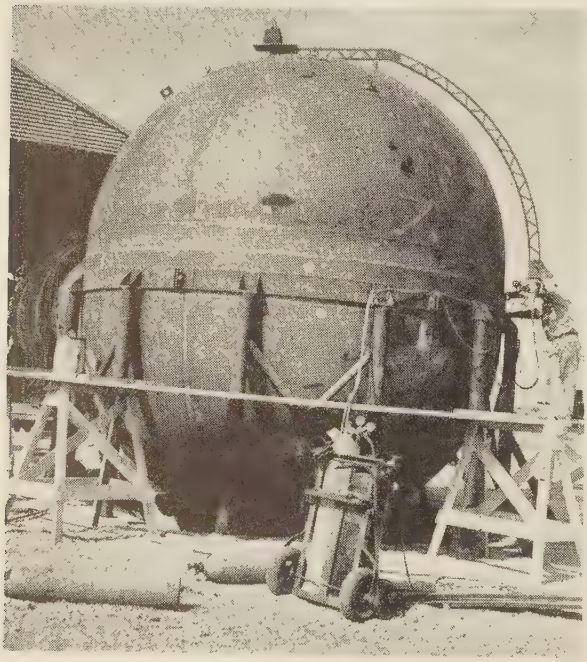

Fig. 8 Two Hemispherical Bulkheads Welded Together for Pressure Test and Being Separated by Flame-Cutting

welding, the extension plates were removed by flame-cutting, and the welding edges restored by grinding. The undersides of the single-pass welds were chipped to sound metal, and back-welded by hand, producing a double-welded butt joint as required in the specifications.

Before the automatic- and manual-welding processes were authorized, the contractor was required to furnish proof of their adequacy. Accordingly, two $^3/_4$-in. plates and two $1^1/_{16}$-in. plates were welded up by the contractor with the single-pass process, and two $1^1/_2$-in. plates with the manual process. Test specimens machined from the $^3/_4$-in. and $1^1/_{16}$-in. plates showed, for the weld metal, yield points and tensile strengths well in excess of the plate values, although the elongations, at 30 per cent, were somewhat lower. Similar results were obtained with test specimens machined from the $1^1/_2$-in. manually welded plates, which indicated average elongations of 28 per cent and strength values also well in excess of the parent plate.

The hemispherical bulkheads were pressure-tested by welding two bulkheads into a closed vessel, which was facilitated by the fact that the bulkhead flanges were rolled in pairs requiring only the welding of a head to each end of the double flange. A hydrostatic pressure of 240 psi, or about double the design pressure, was applied and relieved three successive times similar to the pipe tests. After testing, the two flanges were separated by flame-cutting, as shown in Fig. 8, and the bulkheads were welded into the upstream pipe sections of each 18-ft penstock, using a heavy butt weld between the end of the pipe and the underside of the flange.

Qualification of Welders and Weld Inspection

All welders engaged on manual welding were tested and qualified in accordance with the API-ASME Code. The contractor was required to furnish welded test plates both for the qualification of welders and for routine tests on production welding by the manual and automatic processes. For plates from $^3/_4$ in. to $1^1/_{16}$ in., inclusive, one test plate was furnished for each 200 ft of welded seam, and for the $1^1/_2$-in. plates, one test plate for each 100 ft of welded seam. The test specimens were machined and tested by the contractor, under the supervision of government inspectors. The tests included reduced-section-tension, free-bend, reverse-bend, and nick-break tests, and in some cases also all-weld-metal tension tests.

All shop and field fabrication and welding work was performed in the presence of government inspectors stationed at the plant, and all completed pipe sections were inspected and approved by the inspectors before being released for installation in the tunnels. The radiographic work was done by the contractor, using cellulose-acetate films of the slow-burning type, $4^1/_2$ in. wide $\times$ 17 in. long, with an effective length of 16 in., requiring 43 exposures for each girth joint. The exposure time was 20 sec for the $^3/_4$-in. plate, and $1^3/_4$ min for the $1^1/_2$-in. plate, requiring currents of 140 kv at 5 ma, and 175 kv at 7 ma, respectively. The developed films were examined by the contractor's X-ray technician and the government's chief welding inspector, for defects such as slag inclusions, cracks, porosity, or unfused areas. Defective sections of welds were traced on narrow strips of paper, located on the seam, and marked with yellow crayon for chipping. The chipped areas were rewelded, and reradiographed to prove the quality of the repair welding.

Installation of Penstocks

The specifications for the construction of the dam provided that the upstream sections of the penstocks be installed first to form a water stop for protection during the installation of the lower sections. Therefore, the upstream sections were installed and embedded by the dam contractor during concreting of the dam. Fig. 9 shows several of these with their hemispherical bulkheads welded in place preparatory to embedment in the dam.

The pipe sections for the right or east-side powerhouse were transported from the fabricating plant by the construction railroad, as shown in Fig. 10, to the government warehouse where they were unloaded onto a special 60-ton, 20-wheel trailer drawn by a truck in front, and on the downgrades braked by a truck in the rear. They were then hauled to a point on the riverbank where the trailer was run onto a barge, which was moved across the tail bay to the base of the powerhouse by a winch on the barge. The pipe was transferred from the barge to a tunnel trolley by a barge derrick, as shown in Fig. 11. In the tunnel, the pipe section and the trolley were pulled into place on a 6-ft gage track by an electric hoist placed at the downstream end of each tunnel. The trolley was equipped with jacks and rollers with which the section was aligned, centered, and set to grade from the bench marks previously established by government field parties. Before the tunnel trolley was withdrawn, the pipe section was placed on structural-steel supports which were hinged at their connections with the pipe and with base plates. The bases were shimmed to the proper elevation, and anchored against lateral movement, after which sufficient tack-welding was completed.

The pipe sections for the left or west-side powerhouse were transported by the construction railroad directly into the power-

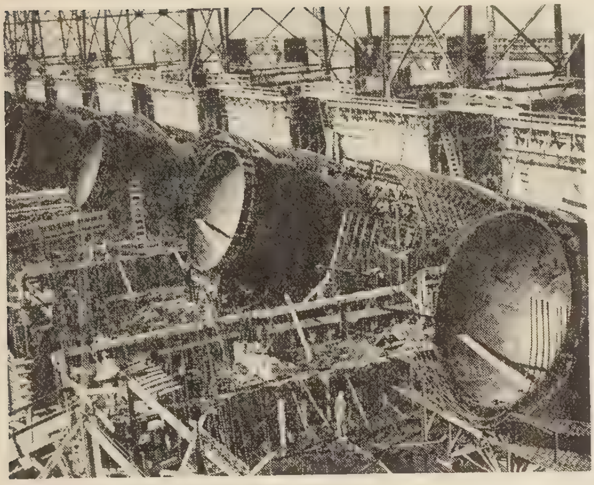

Fig. 9 Upstream Sections of 18-Ft-Diam Penstocks in Place in Dam

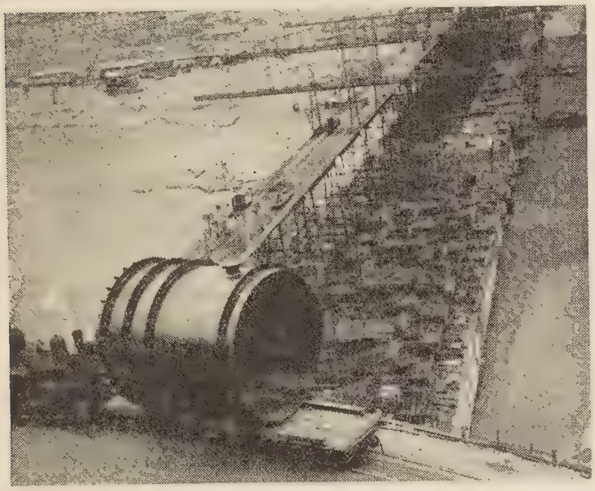

Fig. 10 Railroad Transport of Main-Unit Penstock Section

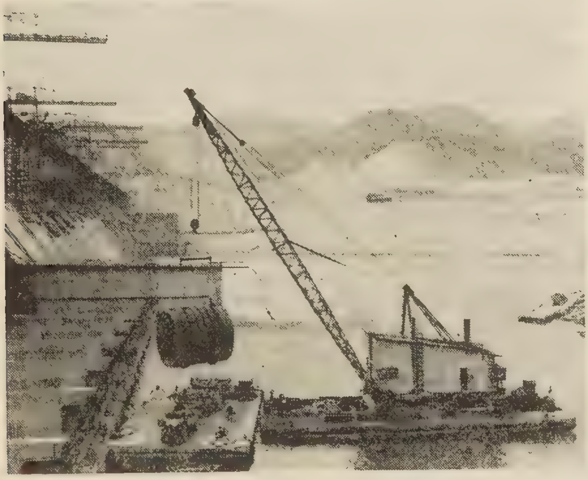

Fig. 11 Hoisting Main-Unit Penstock Section From Transport Barge by Barge Derrick

Fig. 12 Welding of Erection Joints for 18-Ft-Diam Penstocks in Tunnel

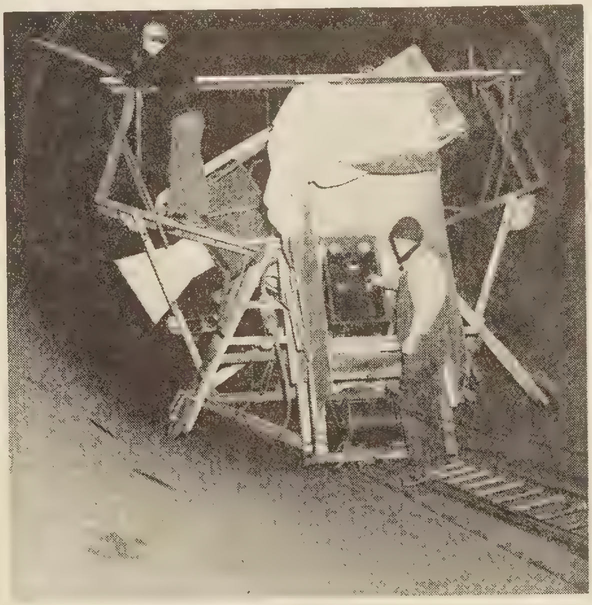

Fig. 13 X-Raying Tunnel Welds in 18-Ft-Diam Penstocks

house itself, where they were lifted off the railroad cars by an electric overhead traveling crane which moved them to their respective tunnel trolleys, at which point they were routed in the same manner as the sections in the east powerhouse.

All tunnel joints were double-V butt welds, no butt straps being used at any seams. For the welding work on the inside of the pipe a portable steel scaffold as shown in Fig. 12 was used. With the exception of the final bead, each was cleaned and peened, the average peening time being 1 hr per bead around the circumference.

All the tunnel welds were radiographed in a manner similar to the shop welds, using a portable X-ray machine mounted on

a rubber-tired trolley, as shown in Fig. 13. Generally speaking, all welding in each tunnel was completed before moving an X-ray machine into it. However, as the work approached completion, X-ray equipment and welding equipment were used in the same tunnel, particular care being taken to protect the workmen from scattered radiation. In all, 51,000 ft of welding were X-rayed, including the shop welds; and a total of 60,000 ft of X-ray film was used. There were approximately 300,000 lb of electrodes used.

After the completion of welding and X-raying, the paint coating on the outside of the pipe was repaired, the top and side supports were welded to the stiffeners, the piezometer piping was installed, and the reinforcing steel placed before the concrete backfill was started. After the backfill was placed, it was pressure-grouted from the inside of the pipe, using tapped grout holes for the purpose. Additional grout holes along the penstock were drilled as required. The penstocks were painted on the inside with two coats of coal-tar paint, applied cold, which completed the installation.

The station-service penstocks were installed first, followed by the main-unit penstocks for the right powerhouse, and then the left powerhouse units were installed. Installation started May 22, 1939, and exactly one year later, on May 22, 1940, all installation had been completed, six months ahead of the scheduled completion date.

Organization

The penstocks were designed under the direction of the author. All penstock designs are made under the general direction of W. C. Beatty, mechanical engineer, and L. N. McClellan, chief electrical engineer. All engineering designs are under the general direction of J. L. Savage, chief designing engineer, and all engineering and construction work is under the general direction of S. O. Harper, chief engineer, Denver, Colo. All activities of the Bureau are under the general charge of John C. Page, commissioner, Washington, D. C.

Discussion

C. L. Barker.[2] This paper presents in an interesting manner the design, manufacture, and installation of the huge pipes which will carry the flow of the Columbia through the Grand Coulee Dam.

Hydraulic model tests were made of the entrance but no tests were made on models of the penstocks. It might have been wise to make tests of the linning surface of the penstocks, and perhaps of the bends and the expansion joints. If the head loss through the penstock could have been reduced 1 ft, the power saved would have been 9000 hp. Tests of this type would also have been of value in comparing model results with the prototype. Piezometer connections were made in the penstock to measure head loss. This makes it possible to build a model and compare the results between model and prototype. Too frequently, tests of this type are not made.

The radiographic method of material examination is interesting and important. For many years, the only tests which could be applied to materials were tests of strength. These were applied to specimens and destroyed the material. The hardness test was one of the first tests of material which did not destroy; the radiographic examination is the second method. Its use in industry has spread rapidly in the last 3 years.

The five failures which developed in the course of the hydrostatic tests are important. The author states that after the five failures had occurred, due to the hydrostatic tests, those portions of film covering the failed sections were re-examined to see if the difficulty lay in studying the film or in the method. In every case the film showed no sign of crack or failure. It is rather safe to presume in every case that these failures, since they failed to show in the film, must have been hairline cracks, the plane of which was in line with the X-ray beam.

Messrs. Townsend and Abbott, of the Bell Laboratories, have reported studies[3] on hairline cracks. This type of crack is common in castings as well as in welding plates. Their report states: "It was found that the limiting cross-sectional area of a crack that could be distinguished in $^1/_2$-in. steel plate (the thickness most frequently encountered in telephone work) under the best radiographic conditions was 0.000035 sq in., representing a crack having a depth of 0.007 in. and a width of 0.005 in. Because of the importance of the orientation of these very fine shrinkage cracks, upon their detection by X-ray methods, it is sometimes desirable and often imperative to take two or more radiographs at different angles, in order that the defects presenting too small a difference in thickness for detection in one of them may be detected in the other, where the path of the radiation through the defect may have happened to coincide more nearly with its longitudinal axis."

This method, which by the way has been used in Germany, suggests the necessity for a slightly different method of radiographic examination which would eliminate the difficulty referred to. It should be possible to build an X-ray machine, using two tubes so spaced that the radiation paths would cross through the weld or section to be examined, at an angle. The radiation would be received on a film, perhaps slightly wider than the present film, but in the same manner. This would give a stereoscopic effect and make possible determination of the location of the defect in the weld or casting. It would also eliminate the possibility of failing to record the hairline cracks due to accidental alignment with the radiation path.

Author's Closure

The discussion presented by Mr. Barker is of timely interest as it touches upon such important features as model tests as an aid to a more efficient design and the use of the proper X-ray technique as a nondestructive test of the welding procedure.

While extensive hydraulic model experiments were necessary for the Boulder penstocks because of the complexity of the system, they were not considered of any particular value in the design of the Grand Coulee penstocks, as the latter consist only of straight runs of pipe without branch outlets or complicated fittings. The reducing bends at the lower ends of the penstocks were the only feature where head losses could be reduced by improved design. These bends were designed with small deflection angles producing a fairly smooth interior, using an R/D ratio of $5^1/_2$ which, on the basis of numerous bend loss experiments, is considered to be about the most favorable ratio, resulting in the lowest hydraulic loss. A model experiment for this specific installation could not have furnished any additional information unless it were performed at a much higher Reynolds number than that used for all bend experiments made to date. Considering the diversity of results between Thoma's experiments at a Reynolds number of 225,000 and Gregorig's experiments at a Reynolds number of 750,000, the conclusion may be drawn that bend loss values obtained with small-scale equipment at such low Reynolds' numbers are not representative of bend losses for large installations having Reynolds' numbers correspondingly higher.

The estimated friction loss through the penstock for a flow of 4500 cfs, at the rated head of 330 ft, is only 1.1 ft. This cannot

[2] Assistant Professor of Hydraulic Engineering, State College of Washington, Pullman, Wash.

[3] "Some Applications of X Rays to Industrial Problems," by J. R. Townsend and L. E. Abbott, *Metal Progress*, vol. 29, Feb., 1936, pp. 64–70 and 86.

be reduced unless the velocity is reduced by increasing the diameter which, however, has been determined from an economic study evaluating hydraulic losses and the cost of construction. The head loss in the bend, according to Thoma's experiments, is estimated at 0.3 ft. This, added to the friction loss in the pipe, would bring the total hydraulic loss to 1.4 ft, which is equivalent to 665 hp per unit. Any economies in head which may be possible by a different design of the bend would be so small as not to effect a worth-while saving.

It is well realized that the cracks which were discovered during the hydrostatic-pressure tests could have been detected by a double X-ray exposure from two different angles. Such procedure, however, would have greatly increased the already high cost of radiographic inspection on this job. In order to get the full benefit of this method of inspection, it would be necessary to use two exposures for every foot of weld, which would nearly double the cost. By checking the pipe sections with an excess pressure test at two thirds of the yield point of the metal, any deficiencies in X-ray examination were detectable, as proved by the cracks discovered during the tests. It may be of interest to mention here that this double-exposure examination was used on some of the repair welds for both the Boulder and Grand Coulee penstocks. The method was useful in such cases to prove the quality of the repair welds, also to disclose the depth of the defect below the surface. This made it possible to determine from which side the defect should be chipped out, reducing thereby the amount of chipping and rewelding necessary in the repair work.

Fig. 1 General Arrangement of Grand Coulee Dam, Power Plant, and Pumping Unit

Turbines for Grand Coulee Dam

By JAMES J. BURNARD,[1] DENVER, COLO.

The Grand Coulee power plant, when fully equipped, will be the largest hydroelectric power plant in existence, containing eighteen 150,000-hp main generating units and three 14,000-hp station-service units, or a total capacity of 2,742,000 hp. The turbines are of the vertical-shaft single-runner Francis type with spiral casing, supplied with water through individual plate-steel penstocks connecting with the upstream face of the Grand Coulee Dam. This paper describes in some detail the mechanical features of the main hydraulic turbines for this plant.

THE Grand Coulee power plant is located at the downstream toe of the Grand Coulee Dam on the Columbia River, about 94 miles north and west of Spokane, in the State of Washington. It consists of two powerhouses, one at each end of the dam, and is designed for the ultimate installation of eighteen 150,000-hp main generating units and three 14,000-hp station-service units.

Nine 150,000-hp turbines, driving 60-cycle main-unit generators of 108,000 kva each, will be installed in each powerhouse and, in addition, space is provided in the left or west powerhouse for three 14,000-hp turbines, driving 60-cycle service generators of 12,500 kva each, and for a control bay. The initial development includes the completion of the left powerhouse and the installation of three 150,000-hp main units, two 14,000-hp service units, and common station facilities.

The general arrangement of the project is shown in Fig. 1. Water is supplied to the turbines through a concrete and steel trash-rack structure at the upstream face of the dam. An individual 18-ft-diam welded-steel penstock with an 18-ft to 15-ft reducing bend at the turbine connection is provided for each main generating unit. A coaster gate, mounted at the upper end of each penstock and operated by a vertical oil-actuated hydraulic hoist, controls the water to the penstock and turbine. Each station-service unit is supplied by an individual 6-ft-diam welded-steel penstock with a shutoff gate of the ring-seal design at the inlet to the scroll case. The inlets to all penstocks are at elevation 1041, and the center lines of the turbine distributors are at elevation 938.[2]

Hydraulic Conditions

The Columbia Basin project, when fully developed, will reclaim 1,200,000 acres of land, regulate the flow of the Columbia River, and develop electrical energy to be used for pumping for irrigation and for industrial purposes. Waters of the upper Columbia River are impounded from a drainage area of 74,100 sq miles into a reservoir 151 miles long, having a total capacity of approximately 10,000,000 acre-ft. The upper 80 ft of the reservoir, containing approximately 5,000,000 acre-ft, are available for power production and for the regulation of the river flow for the improvement of navigation and benefit of future power developments downstream from the dam.

The power generated from six of the proposed eighteen 150,000-hp units will be used during the high-water season to pump water into a balancing reservoir in the Grand Coulee where, by means of a system of canals and laterals, it will be distributed for irrigation purposes. High-water periods occur at such times that secondary power will take care of pumping needs. Therefore, all primary and some secondary power will be available for power

[1] Engineer, Bureau of Reclamation, United States Department of the Interior.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., Sept. 3–6, 1940, of The American Society of Mechanical Engineers.

[2] "Penstocks for the Grand Coulee Dam," by P. J. Bier, published on page 219 of this issue of the Transactions.

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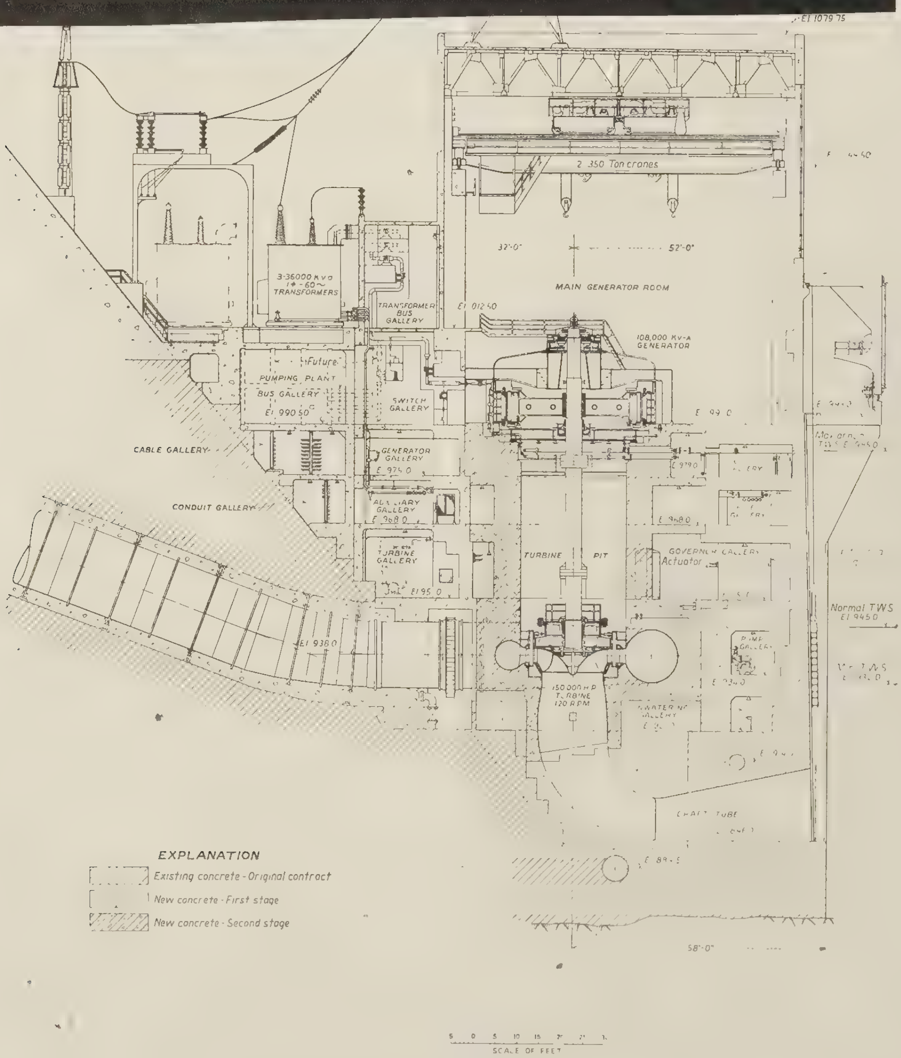

Fig. 2. Typical Cross Section Through the Power Plant

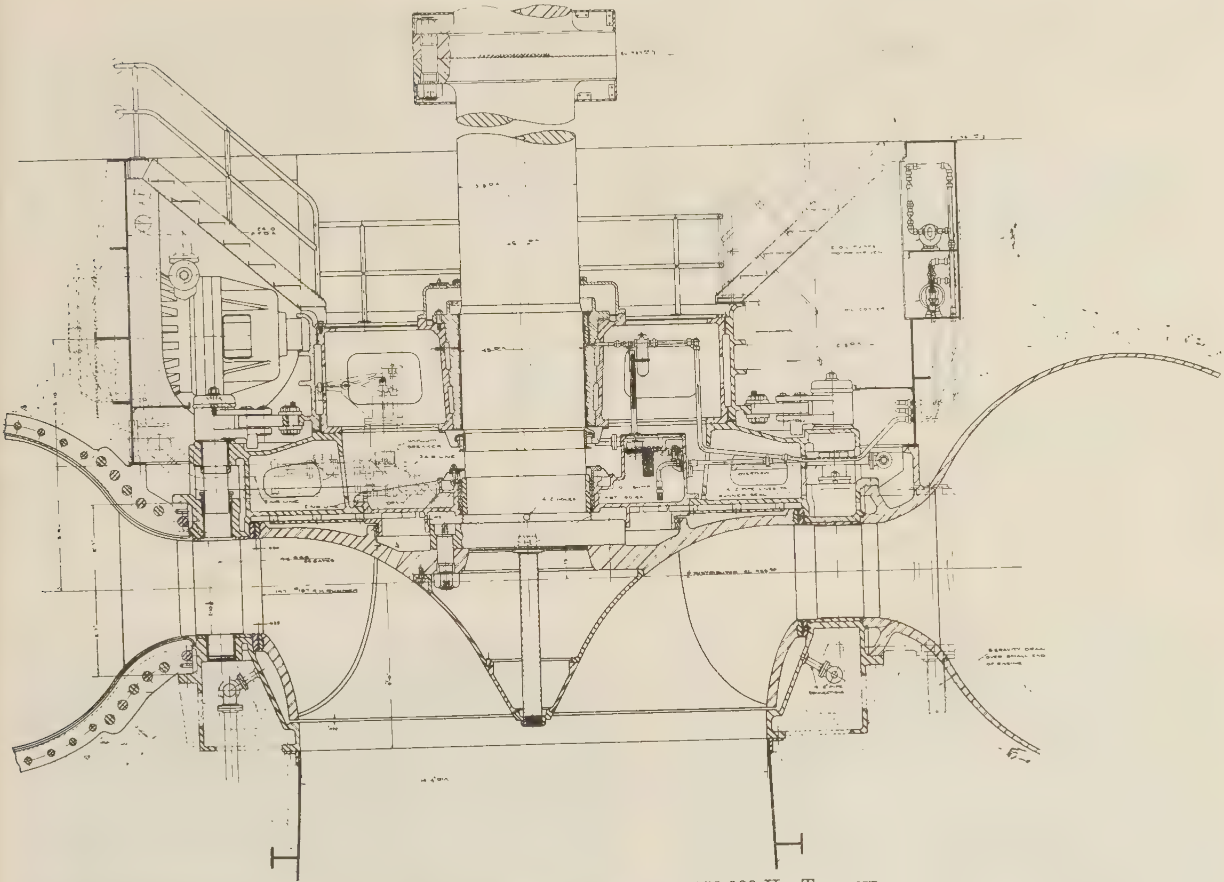

Fig. 5 Cross Section Through 150,000-Hp Turbine

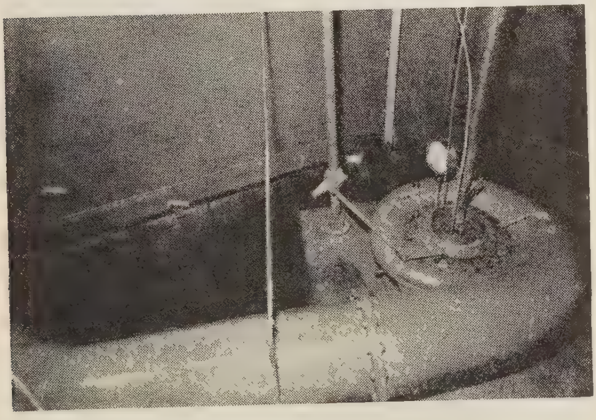

Fig. 4 Turbine Model in Manufacturer's Laboratory

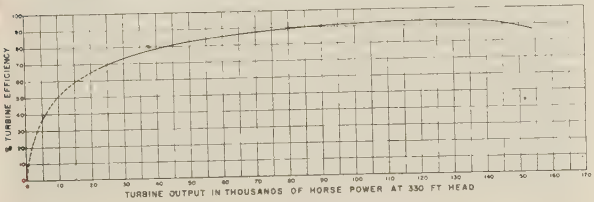

Fig. 3 Curve Showing Expected Performance of 150,000-Hp Turbine at the Rated Head

920,000 kw of which 800,000 kw will be used for the generation of firm continuous power, and the balance for secondary power for pumping for irrigation and for stand-by service. Load conditions for this power require the installation of sufficient units to bring the ultimate turbine installation to 2,700,000 hp.

The turbines selected for the Grand Coulee power plant are of the vertical-shaft single-runner Francis type with spiral casing. The three initial 150,000-hp turbines operate at a speed of 120 rpm and, with respect to horsepower capacity, are the largest hydraulic prime movers in existence. As shown in Fig. 3, they are designed for a maximum efficiency of about 93 per cent at a power output of 125,000 hp under the designed head of 330 ft. A model of the turbine is shown in Fig. 4. This model is homologous with the prototype (except for being of left-hand rotation to suit the manufacturer's laboratory). The homologous parts include the runner, casing, wicket gates, and draft tube. The runner diameter is 18.9 in. as compared with 197 in. for the prototype. Tests on the model were made under heads varying from 25 to 50 ft.

The arrangement and setting of the 150,000-hp turbine is shown in Fig. 5. A cast-steel spiral casing, cast integral with the speed ring, is embedded in the concrete substructure of the powerhouse, with the center line of the distributor at elevation 938. A plate-steel pit liner, to which the servomotors are bolted, lines the turbine pit and extends from the top of the casing to elevation 951. The upper and lower covers of the turbine are bolted to the speed ring, and contain the stationary wearing rings and plates and the bearings for the wicket gates which control the water to the runner. The upper cover also supports the single main

bearing, stuffing box, and gate-shifting ring. The runner, of cast steel, is attached to the main shaft by means of a bolted, flanged connection. The draft-tube liner is bolted to the lower cover and extends to a point 22 ft below the center line of the turbine distributor.

Hydraulic Design

The velocity of the water entering the spiral casing is 29 fps, or approximately 20 per cent of the spouting velocity under the designed head, Fig. 6. This velocity remains approximately con-

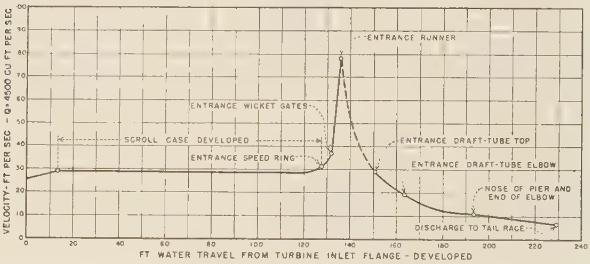

Fig. 6 Curve Showing Velocity of Water Through 150,000-Hp Turbine

stant throughout the length of the casing, as the cross-sectional area of the casing is decreased proportionately to the increment of flow into the speed ring. The speed ring forming the throat of the spiral casing is spanned by fourteen fixed vanes, so designed as properly to direct the water to twenty-four movable wicket-type vanes or gates. The water, after passing through these wicket gates, enters the runner with a velocity of approximately 53 per cent of the spouting velocity under the designed head. The runner has nineteen vanes the curved surfaces of which, by changing the direction of the water, convert the energy of the water into power and produce a final discharge velocity at the throat of the draft tube of approximately 29 fps. The discharge of water from this point to the tailrace is through a flattened elbow type of draft tube where the decrease in velocity is such as to produce a residual velocity at the end of the draft tube of 6.8 fps. This residual velocity represents a final rejection 0.2 per cent of the total available energy.

Leakage of water between the crown of the runner and the turbine cover plate, and around the lower shroud of the runner, is minimized through the use of two plate-steel wearing rings which act as seals, Fig. 5. Means are provided, should the unit be operated as a synchronous condenser with the turbine gates closed and the water depressed below the runner, to supply the runner-seal chambers with penstock water. The essential use of this water is to act as a cooling medium to prevent heating and consequent expansion and possible seizing of the runner wearing rings to the stationary wearing rings located in the upper and lower covers of the turbine.

Hydraulic losses in the seal chambers are materially reduced by the elimination of all projections and pockets. A plate-steel baffle is provided on the upper cover just above the runner to direct any leakage water through the seal clearances to cored holes in the runner hub.

If future operation of the units as synchronous condensers is found desirable, provision has been made for the installation of float-operated air valves which will admit compressed air below the runner. The level of the water in the draft tube will be depressed to a point about 3 ft below the bottom of the runner, and thus materially decrease the power required to drive the runner.

Provision has been made for the admission of atmospheric air to the turbine cover plate under conditions of small gate openings to act as a draft-tube vacuum breaker and to remove water from the runner. This is accomplished by means of a poppet-type valve, automatically operated from the gate-shifting ring. An 8-in. pipe line connects the valve with the downstream face of the powerhouse so as to prevent the objectionable noise from an air inlet inside the turbine pit.

The penstock and scroll case are initially filled by means of a 24-in. by-pass, located at the upstream end of the penstock. The entrained air is discharged through a 30-in. pipe, embedded in the concrete, terminating at elevation 1302 at the downstream face of the dam.

Mechanical Design

The physical dimensions of the spiral casings for the Grand Coulee turbines are the largest ever attempted for cast-steel construction. In order to transport a casing of this size from the manufacturer's shop to the project, it was necessary to sectionalize it into fourteen parts with radial flanged bolted joints. The average section is 15 ft 6 in. × 14 ft 6 in. × 7 ft 6 in. and weighs approximately 41,000 lb. The total weight of the castings for the 14 casing sections of the turbine is 582,000 lb. The heaviest section weighs 59,000 lb. Most of the casing sections, the runner, and many other parts, will have to be shipped on special cars. A view of the casing assembled at the contractor's plant is given

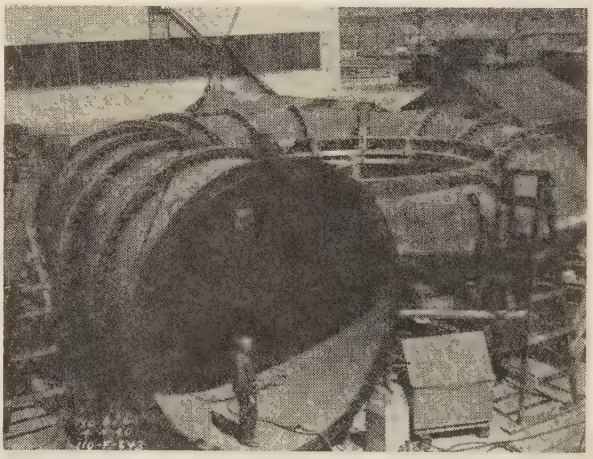

Fig. 7 Spiral Casing Assembled at Manufacturer's Plant

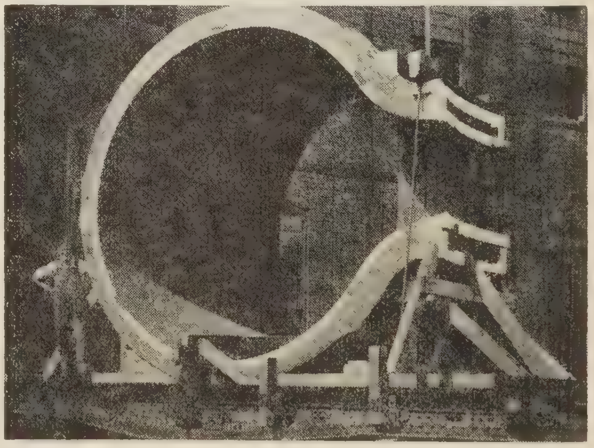

Fig. 8 Casting for Casing Section No. 7; Weight 38,000 Lb

in Fig. 7. A view of one of the sections being transported is shown in Fig. 8.

Owing to the large number of bolted sections, the support and anchorage of the casing, while filled with water under pressure during erection and embedding in concrete, presented a very interesting problem. After much study of various methods, it was finally agreed that the support by means of screw jacks bearing against steps cast on the flanges of the various sections would prove the most suitable. Turnbuckle rods, attached at one end to ears welded on the flanges and at the other end to loop bars embedded in the surrounding concrete, are prestressed to 12,000 psi to prevent any possible movement of the casing during the pouring operation. This method of support and anchorage is shown in Fig. 9.

All of the essential parts of the turbines are made of cast steel with an allowable design stress not to exceed 10,000 psi under normal operating conditions. Under emergency conditions, stresses up to one third the yield point of the material were allowed. The flanges of the scroll case, however, are an exception, having an allowable bending stress of 15,000 psi. It was found that, if the bending stresses were kept below the 10,000 psi, required by the specifications, the flange thicknesses would be so great that unsound castings would probably result.

Turbine Runner

The turbine runner is of the Francis type of cast steel made in one piece and is 197 in. diam. It has sufficient strength to support its own weight plus the weight of the turbine shaft, up to the connection with the generator shaft, with the runner resting on a finished ledge in the lower cover or foundation ring. It is designed and constructed to withstand safely the stresses resulting from operation at a runaway speed of 220 rpm under conditions of maximum head, with the turbine gates wide open and with no load on the generators.

The runner is provided with two steel wearing rings, one on the lower shroud and one on the crown. The wearing rings are shrunk on and machined on the outside diameter for a close running fit with stationary wearing rings in the top and bottom covers. Cored passages in the hub reduce the downward thrust of the water on the top of the runner, and the water passages are finished to a smooth surface to minimize friction and cavitation. A cast-steel runner tip with a removable cast-steel cap will allow inspection of the runner and shaft connection. The finished runner, complete with wearing rings attached, is to be statically balanced in the contractor's shop before shipment to the project. The runner casting being transported to the manufacturer's shop is shown in Fig. 10.

Shaft for the Main Turbine

The turbine shaft is made in two sections of forged heat-treated open-hearth carbon steel with coupling flanges forged integrally with the shaft. The material of the shaft has an ultimate strength of 70,000 psi and a yield point of 35,000 psi and, under normal operating conditions, is stressed to not more than 4700 psi. It is capable of operating at any speed up to a full runaway speed of 220 rpm without vibration or objectionable distortion. It is 44 in. in diam at the coupling end and extends 43 ft above the center line of the turbine distributor, where it connects to the generator shaft. The flanges at the ends of the shaft form a male and female bolted coupling with body-bound forced-fit coupling bolts. Axial holes are provided in the bolts to permit the use of a cooling medium for shrinking them for assembly and disassembly. A bolt cover made in halves encloses the nuts on each side of the coupling.

A 6-in-diam hole is provided throughout the length of the shaft for inspection purposes, with four 2-in-diam radial holes adjacent to the lower flange connecting with the central hole for

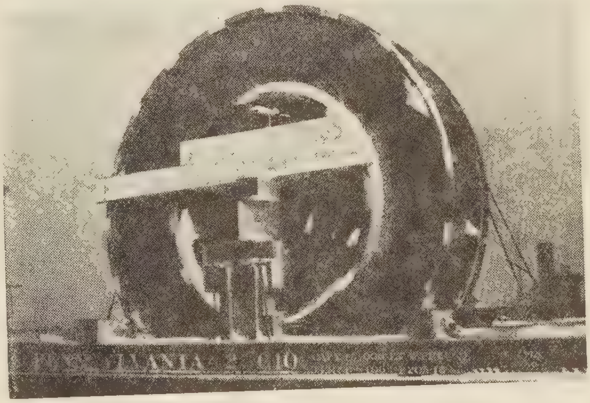

Fig. 10 Runner Casting for 150,000-Hp Turbine

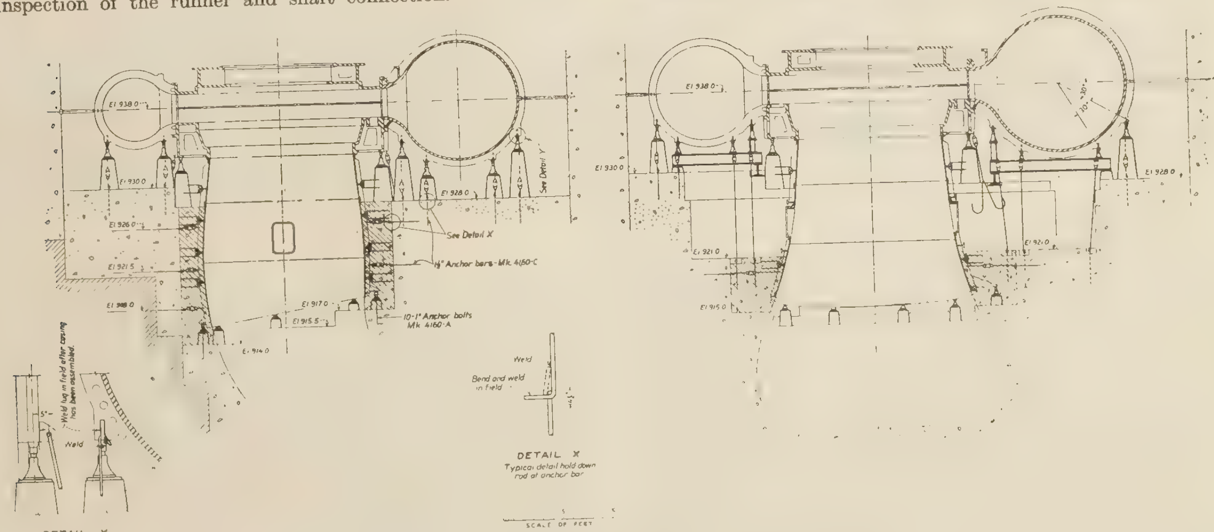

Fig. 9 Arrangement of Supports and Anchors for Turbine Casing

the purpose of admitting air from the cover plate to the tip of the runner. A renewable stainless-steel sleeve in halves is secured to the shaft, by means of shrink links, where it passes through the packing box.

Guide Bearing

The main-shaft bearing is of the babbitt, oil-lubricated type, 45 in. diam, with a length of $41^1/_2$ in., and consists of a semisteel shell, made in two sections, bolted to a bearing support. The bearing support is of cast steel in two pieces and is bolted to the upper cover. It will support the main bearing and act as a guide for the gate-operating ring. A pressure switch located in the oil groove in the bearing directly opposite the oil inlet will sound an alarm on low oil pressure, and a mercury-contact-type thermal relay with bulb located in the hot-oil discharge from the bearing oil pan will sound an alarm upon excessive temperature rise. A cast-steel oil deflector provided with an oil collector is secured to the shaft immediately below the bearing and drains the oil into the turbine oil sump which is formed in one side of the packing-box support.

Two duplicate lubricating-oil pumps, for circulating the oil through the bearing, are mounted in an alcove in the pit liner. The main pump is driven by a 60-cycle 440-v a-c motor, and the stand-by pump is driven by a 250-v d-c motor. Normally, the oil will be circulated by the a-c motor pumping unit, the d-c motor pumping unit being arranged to start automatically and supply oil to the bearing upon failure of the oil pressure.

Casing and Speed Ring

The casing is of spiral form, made of cast steel in fourteen sections, and is cast integrally with the speed ring which forms a tie across its throat. The sections are joined together by means of flanged, bolted joints of ample strength and rigidity, Fig. 7.

The speed ring containing fourteen fixed vanes is designed to support the weight of the superimposed parts and also to resist the upward thrust due to a pressure of 175 psi in the casing with no superimposed load.

The casing is substantially circular in cross section to prevent deformation under pressure and is designed so that all internal parts of the turbine may be removed from above, Fig. 5. It has an inlet diameter of 15 ft and extends upstream to the face of a finished flange a distance of 20 ft from the center line of the turbine, where it connects to an expansion joint. A 24 × 36-in. manhole with a hinged cover is provided in the casing for access to the interior.

The expansion joint made of plate steel connects the turbine casing to the penstock and consists of two separate pipe sections and an outside ring which form a packing box at the joint between the sections. The packing box is provided with alternate rings of rubber and flax packing held in place by steel glands.[2]

Covers

The top and bottom covers are each made of cast steel in two pieces to facilitate shipment and handling. The top cover is designed to support the weight of the main-shaft-bearing housing, the stuffing box, and the gate-shifting ring, Fig. 5. It contains two bronze-bushed bearings for each wicket-gate stem. Provision is made to carry the weight of the wicket gates and levers and any unbalanced hydraulic thrust by means of thrust bearings in the upper cover. The lower and inner faces of the top cover adjacent to the gates and runner are provided with finished surfaces for attaching renewable plate-steel wearing plates and rings.

An adjustable packing box, which may be repacked without disturbing the main bearing, is provided where the main shaft passes through the top cover. The packing box is lubricated by means of water supplied from the penstock through a reducing valve in an amount sufficient to effect lubrication without leakage into the turbine pit. Packing boxes are also provided where the wicket-gate stems pass through the top cover.

The bottom cover is combined with the foundation ring. It is bolted to the spiral casing and draft-tube liner and contains the lower bronze-bushed bearings for the wicket-gate stems. The upper and inner faces of the bottom cover are provided with finished surfaces for attaching renewable plate-steel wearing plates and rings, as in the top cover.

Wearing Rings and Plates

Each runner wearing ring is made from three carbon-steel bars with the ends welded together to form a ring. The ring is shrunk on the runner and in addition is secured by means of fillister-head screws with heads countersunk flush with the outside diameter. The stationary wearing rings in the top and bottom covers are also of carbon steel, but in addition each wearing ring has two inserts, of hard brass, calked into dovetail slots in the ring. The hard-brass inserts were decided upon as it was desired that two metals of different characteristics be used in the event of accidental contact between the rotating and stationary rings. The stationary rings are machined on the outside diameter and sufficient stock is left on each ring seat in the top and bottom covers to machine to a true circle by means of a boring rig during assembly at the project.

Gates and Operating Mechanism

The turbine gates are of the balanced-wicket type of cast steel with stems cast integral. Each stem is provided with three bronze-bushed bearings, one located in the bottom cover and two in the upper cover, one below and one above the stuffing box. The stem is bored throughout its entire length to supply grease to the lower bearing.

Each gate has a cast-steel lever keyed to its upper end which, by means of a steel link, is connected to the gate-shifting ring. The link is connected to the lever by a semisteel shear pin designed to be the weakest element in the gate mechanism. This pin is strong enough to withstand the maximum operating forces but will break and protect the rest of the mechanism from injury in the event that one or more of the gates becomes blocked.

The gates are so designed that in case of breakage of the shear pin the movement of the gate is limited so as to prevent interference with the operation of other gates or the runner. Each gate is also provided with a thrust collar to carry any upward thrust due to hydraulic pressure, and is suspended in mid-position between the upper and lower covers by means of a thrust washer.

The gate-shifting ring is of cast steel in one piece, of rigid design, and is guided by renewable bronze guide strips on the top cover and bearing housing. It is connected at diametrically opposite points, through adjustable forged-steel connecting rods, to the servomotors.

Pit Liner

The pit liner is of welded construction, made of $^1/_2$-in. steel plate, with an inside diameter of 24 ft. The bottom of the liner is bolted to a flange on the spiral casing and the liner extends up from the top of the casing to the governor gallery floor at elevation 951. The liner is designed to withstand hydrostatic pressure with tail water at elevation 985 without severe distortion. It has rigid circular flanges for mounting the servomotors. Alcoves are formed in the liner for the main-bearing oil pumps and instruments. Two checkered steel-plate walkways are provided in the turbine pit, one outside of the gate stems and one on top of the bearing housing. Steps leading to the turbine pit are mounted on the shifting ring, Fig. 5.

Servomotors

The turbine is provided with two oil-pressure-operated double-acting hydraulic cylinders or servomotors which are mounted on heavy circular flanges in the walls of the pit liner. They have semisteel cylinders and cast-steel heads and stuffing boxes. The servomotors have a combined capacity sufficient to exert a torque in the shifting ring of 2,900,000 ft-lb, with an oil pressure of 250 psi. Under this oil pressure and with an adequate supply of oil, the servomotors are capable, under maximum operating-head conditions, of moving the turbine gates a full opening or closing stroke in 4 sec. The servomotors are provided with adjustable by-pass connections, whereby the rate of closure may be retarded from slightly below the speed-no-load position for maximum head to the fully closed position, so as to minimize pressure rises in the penstock. Provision is made on the piston rods for locking the gates positively in the opened or closed position, or for limiting the movement of the gates.

Draft Tube

The draft tube is formed in the concrete substructure of the powerhouse, the upper part being lined to a point 22 ft below the center line of the turbine distributor. The liner is made of $^3/_4$-in-thick steel plate with welded joints and is heavily reinforced on the outside by means of suitable ribs. These ribs also provide means for anchoring the liner to the concrete of the powerhouse by the use of turnbuckle rods. Jack pads, attached to the lower edge of the liner, are provided for supporting it during the pouring of concrete. The top of the liner is bolted to the foundation ring by means of a flanged connection. Two 24 × 36-in. manholes with cast-steel hinged covers, opening outward into the access passageways, are provided diametrically opposite each other to permit access to the draft tube and to the under part of the turbine runner.

The liner was assembled in the shop and match-marked, then shipped completely knocked down for welding in the field.

The draft tube is unwatered by gravity through valves provided in the draft-tube dividing piers. These valves connect through a common unwatering header to a sump, from which the water is pumped by a motor-driven, deep-well type of pump to the tailrace. During the unwatering operation, the downstream end of the draft tube is closed by means of steel stop logs.

Governors

Governors for the 150,000-hp units are of the oil-pressure actuator type with motor-driven speed-responsive elements actuated from 3-phase permanent-magnet generators connected to the tops of the generator shafts. They control the gates of the turbines by means of the servomotors connected to the gate-shifting ring. They are adjustable to provide variable rates of actions for completely opening or closing the turbine gates in from 4 to 12 sec. The speed-responsive elements are sensitive to turbine-speed variations of 0.01 per cent. The speed change from no load to full load is adjustable from 0 to 6 per cent.

Each governor is complete with an oil-pressure system, including two pressure-controlled 40-hp motor-driven pumps and a sump tank located within the actuator, and a pressure tank located adjacent to the actuator. The actuator is located at one end of and forms a part of the control board in the governor gallery at elevation 951, Fig. 11. The pressure tank is located just back of the actuator and connected to it by means of a 6-in. pipe below the floor level. The system is designed to operate with an oil pressure ranging from 250 to 300 psi.

The actuators are equipped with the following features which may be operated manually at the actuator or electrically by remote control from the control board:

(*a*) A gate-limit-control device which will operate two adjustable 1.5-amp 250-v d-c ungrounded limit switches. Each switch is independent and adjustable for circuit opening or closing over the range of travel of the gate-limit device.

(*b*) A speed-level-controlling device. Speed controls are from 85 per cent of rated speed at no load and zero speed droop to 115 per cent of rated speed at rated load and maximum speed droop.

(*c*) A device for opening and closing the turbine gates at the normal rate of movement. This device will be used for manual starting and stopping of the turbine and for automatic shutdown by means of automatic protective features incorporated in the main generator, governor equipment, or transformers.

(*d*) A device for controlling the speed droop of the turbine. The amount of speed droop is adjustable from zero to 5 per cent.

In addition, the following devices are furnished:

(*e*) An electrically operated speed indicator mounted on the actuator column. This speed indicator is driven from a magneto-type generator on the generator shaft. In addition to indicating turbine speed, it will also indicate when rotation starts and stops.

(*f*) Two gate-limit and gate position indicators of the dual type, one mounted on the actuator and the other mounted on the benchboard in the main control room. These instruments indicate the position of the governor gate-limit device and the position of the turbine gates.

(*g*) An overspeed switch, mounted on and forming a part of the governor-drive generator, arranged to shut down the turbine and sound an alarm upon overspeed.

(*h*) A combination automatic and hand-operated air valve for controlling the operation of the generator brakes. The air valve is controlled by means of a low-speed switch, located in the housing of the permanent-magnet generator, so adjusted that the brakes cannot be applied until the turbine gates are fully closed and the speed of the unit has been reduced to 30 rpm. Brake application is intermittent with the time periods adjustable, for a selected number of cycles. After that, the brakes are applied constantly until the unit is brought to a stop.

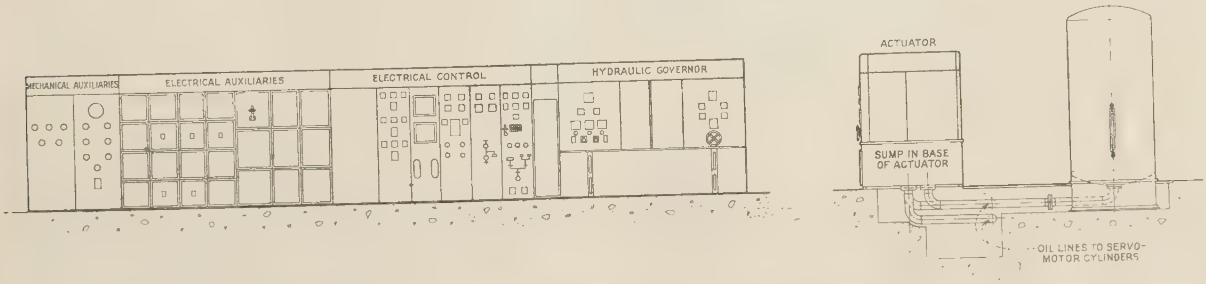

Fig. 11 Arrangement of Control Boards and Governor

(*i*) Manual control of the turbine gates at the actuator by means of oil pressure from the governor oil-pressure system.

(*j*) An airplane-type flexible-steel restoring cable, operating over sheaves and enclosed in a metal conduit, connecting the governor-restoring mechanism and the turbine gate-operating mechanism. This eliminates lost motion between the servomotor piston travel and the governor pilot valve.

The two oil pumps in the governor system have a combined capacity per minute of 3 times the total oil volume of the servomotors. They will supply an adequate quantity of oil to the servomotors so as to operate the turbine gates through a complete closing or opening stroke in 4 sec with an oil pressure of 250 psi, and with an operating head on the turbine of 355 ft. They are arranged to start and stop at predetermined oil pressures in the pressure tank.

The two pumps are interconnected so that they can be operated independently, or together. When operating together, the interconnection and automatic control is such that either pumping unit may be used for normal operation with the other unit serving as a stand-by unit, arranged to start automatically either on failure of the electric-power supply to the operating pump or upon the oil pressure falling below a predetermined amount.

The pressure tank has a total volume of 20 times the volume of the servomotor cylinders and will supply five complete servomotor strokes of the turbine with a drop in pressure from 300 to 250 psi without the operation of the pumps. Float valves are provided in the bottom of the pressure tank to close automatically in case the oil level drops, so as to prevent air, from the pressure tank, entering the governing system.

The oil sump tank for the governor system is located in the base of the actuator, thus eliminating the necessity of having sump tanks below the governor floor.

Acceptance Tests

After the turbines have been installed in the powerhouse and placed in satisfactory operation, they will be tested to determine whether or not the contractor's guarantees of horsepower, water discharge, and efficiency have been fulfilled. These tests will be conducted in accordance with the testing code for hydraulic turbines recommended by The American Society of Mechanical Engineers.

Pumping Plant

Some of the power developed at the Grand Coulee Dam is to be used for the operation of pumps for irrigation purposes; therefore, a short description of the pumping plant may be of interest.

All irrigation water for the Columbia Basin project will be pumped approximately 295 ft from the reservoir behind the Grand Coulee Dam into a balancing reservoir formed in the Grand Coulee by the construction of earth dams at each end. The Grand Coulee Dam will raise the water from a minimum of 275 ft to a maximum of 355 ft, and the pumps will lift it the remaining distance to the balancing reservoir. Ordinarily, pumping will be against a 295-ft head, i.e., from a full storage reservoir behind the dam to a full balancing reservoir in the Grand Coulee.

The pumping plant is located along the shoreline of the reservoir just upstream from the left abutment of the dam, as shown in Fig. 1.

Twelve pumping units are proposed for the ultimate installation, each unit consisting of a single-stage vertical-shaft centrifugal pump having a capacity of 1600 cfs when operating under a total head of 295 ft, direct-connected to a 65,000-hp synchronous motor. The motor capacity is of such size that, with a full reservoir, one main generating unit in the power plant will have sufficient capacity to operate two pumping units.

The water is supplied to each pump through an individual 14-ft-diam welded plate-steel intake pipe which has a bellmouth inlet at elevation 1191.75 in the upstream face of the pumping-plant dam. The center lines of the pump casings are located at elevation 1203 and from this point the water is discharged through 12-ft-diam welded plate-steel discharge pipes about 800 ft long into a canal leading to the balancing reservoir in the Grand Coulee 1.7 miles away.

Through a canal of approximately 15,000 cfs capacity, water will be carried about 10 miles from the balancing reservoir to other canals from which it will be distributed through numerous laterals for irrigation purposes.

Acknowledgments

The 150,000-hp turbines for the Grand Coulee power plant were designed by the Newport News Shipbuilding and Dry Dock Company, Newport News, Va., and the governors were designed by the Woodward Governor Company, Rockford, Ill., in accordance with specifications prepared by the United States Bureau of Reclamation.

The author is indebted to the engineers of the Bureau of Reclamation for valuable aid in connection with the preparation of this paper.

The mechanical installations at the Grand Coulee power plant were designed under the supervision of the author. All mechanical installations are designed under the general direction of I. A. Winter, senior engineer, and L. N. McClellan, chief electrical and mechanical engineer.

All engineering designs prepared by the bureau are under the general direction of J. L. Savage, chief designing engineer; all engineering and construction work is under the direction of S. O. Harper, chief engineer, with headquarters at Denver, Colo.; and all activities of the bureau are under the general charge of J. C. Page, Commissioner of Reclamation, with headquarters at Washington, D. C.

Discussion

R. V. Terry.[3] This paper has been written primarily from the viewpoint of the designer of the power plant. Perhaps a few remarks on these turbines from the viewpoint of the turbine designer and manufacturer would be of interest.

Among the first problems to be attacked was that of splitting up the parts, especially the cast-steel casing, so that they could be handled by rail shipments. The final design evolved for the casing provided for splitting the combined casing and speed ring into 14 pieces, which included two inlet rings. Longitudinal joints were avoided, all joints being placed at right angles with the flow. This reduced the loading of joints per unit length and facilitated manufacture. The angular spacing of the segments varied from 18 deg at the large end to a maximum of 63 deg near the smaller end. The 14 speed-ring vanes are irregularly spaced to suit each casing segment. Scale models were made of the largest pieces, of the special available railroad cars, and of the composite railroad clearances, as a final check on the drawings. High-tensile-strength bolt material, heat-treated, was used for the joints to reduce flange dimensions, and the bolts were prestressed to 23,000 psi, as determined by micrometer measurements.

Due to its size and weight, the casing was assembled on a special foundation outside the shop, similar to that used in the field for boring, for assembly with other parts to be embedded in concrete, and for the hydrostatic test. Boring was accomplished with a portable mill previously employed for battleship-turret work. The complete assembly for the hydrostatic test weighed

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about 786 tons, including 375 tons of water. During the shop hydrostatic test of the first unit, the only one tested to date, deflection readings were taken in mils at 48 points. The casing joints were practically droptight. Leaks were found through the castings in only a few places.

A somewhat new detail of a rectangular-shaped water-sealing groove was employed for the round rubber cord at the joints. The width of the groove is slightly less than the diameter of the cord, creating a pinch which holds the cord in place without clamps while placing the adjoining part. This type of groove seems to be superior to the older type. A model test indicated that the flange could be backed off nearly $^1/_8$ in. before the cord would blow at 230-psi test pressure. This type, incidentally, would be equally effective for pressure in either direction.

The shop space available, as well as the shipping schedule, did not permit of making a complete shop assembly of all embedded and nonembedded parts at one time. However, separate assemblies were made so that every part was fitted to its mating part or parts and thus proper fitting in the field is assured.

The servomotors are carried by the walls of the pit liner. However, the pit liner is not designed to carry the servomotor reactions without the help of the concrete backing. Alternate bolts of the servomotor flange serve as foundation bolts and extend well back into the concrete.

It will be noted from Fig. 5 of the paper that a considerable radial clearance is allowed between the speed-ring vanes and the wicket gates and between the gates and the runner vanes. The former clearance is necessary in this case due to the irregular spacing of the speed-ring vanes previously mentioned, there being 14 vanes and 24 wicket gates. The latter clearance, between gates and runner with the gates fully open, is about $7^3/_8$ in. or 3.75 per cent of the runner diameter. Ample clearance at that point is, of course, required to reduce mutual interference of a hydraulic nature, particularly for turbines which must be operated over a wide range in head.

The runners are 197 in. nominal diam, the approximate limit for transportation in one piece, and each runner weighs about 125,000 lb. Such runners are given an accurate check for static balance by supporting them on a hardened spherical point at the axis, slightly above the center of gravity of the runner.

Provision is made for the possible admission of free air at three different points, Fig. 5 of the paper. First, an 8-in. valve is provided, cam-operated from the gate mechanism, to admit air at the lower gate openings through ports in the runner crown, just downstream from the runner vanes. With an initial normal tailwater level 7 ft above the center line of the runner, it is not expected that free air can be admitted through that valve. Later when the tailwater level drops to its minimum value, 6 ft below the center line of the runner, that valve will become effective. In the meantime use will be made of the 3-in. air line which terminates at the center of the runner cone, usually the point of lowest pressure when the water leaves the runner with a whirl at low- or high-gate openings, such as to create a vortex disturbance in the draft tube. The 3-in. air line takes free air from the turbine pit through a Maxim silencer, a check valve, and a gate valve. The system will be effective and take air only when required, that is, when a vortex exists at the tip of the runner cone. The amount of air may be partially controlled by throttling the gate valve. A third air line, 2 in. diam, is provided through the crown plate to the space above the runner, just outside of the intermediate seal, where a low pressure is expected to occur. Any of the three schemes of air admission may be used as the conditions of tailwater level or other operating conditions may dictate. Although the various conditions were studied during the running of the model tests, it is difficult to predetermine the exact adjustments needed for the field.

TABLE 1 COMPARISON OF BOULDER DAM AND GRAND COULEE TURBINES

Plant	Boulder	Grand Coulee
Rated unit, bhp	115000	150000
Rpm	180	120
Rated head, ft	480	330
Specific speed	27	33
Rated discharge, cfs	2400	4500
Inlet diameter of casing, ft	10	15
Number of casing sections	6	14
Weight of casing, lb	450000	582000
Maximum operating head, ft	590	355
Casing test pressure, psi	500	230
Diameter of wicket-gate circle, ft	17	19
Nominal diameter of runner, in	171	197
Height of wicket gates, ins	19	$34^3/_8$
Diameter top of draft tube, ft	11	14.33
Pit diameter, ft	22.5	24.0
Diameter of shaft, in	36	44
Length of shaft from center line of turbine to face of generator coupling, ft	26	43
Diameter of shaft coupling, in	$61^3/_4$	75
Size of main bearing, in	$36^1/_4 \times 29^1/_2$	$45 \times 41^1/$
Servomotor capacity, ft-lb	340000	400000
Operating ring torque, ft-lb	2100000	3180000
Total turbine thrust, lb	590000	925000

The complete shipping weight of one turbine is expected to be about 820 tons, including the draft-tube liner, the 15-ft expansion joint at the casing inlet, and the intermediate turbine shaft but not including the governor and governor piping.

Table 1 of this discussion gives a comparison of the principal data and dimensions of the Grand Coulee and Boulder Dam turbines.

It has given the writer unusual pleasure to be associated with the design and manufacture of these turbines which are of unprecedented size and horsepower capacity, being over 30 per cent more powerful than the most powerful hydraulic turbines now in operation. His company received most excellent cooperation from the engineers and inspection personnel of the Bureau of Reclamation.

Author's Closure

The discussion presented by Mr. Terry contains many interesting features not covered by the author and should add much to the value of the paper.

He is correct in his statement that the pit liner is not designed to take the full servomotor reaction but is aided by foundation bolts extending into the surrounding concrete.

He also properly points out that the 8-in. valve for admitting atmospheric air through ports in the runner crown may not become effective until the tailwater level drops to its minimum value and in the meantime use will be made of other available means of air admission.

Table 1 of the discussion gives an interesting comparison between the Boulder Dam and Grand Coulee turbines. The particular Boulder Dam turbines used in this comparison, however, were not designed to operate solely at 60 cycles, 180 rpm. The specifications issued by the Bureau of Reclamation for these turbines called for units which could operate satisfactorily at either 50 cycles, 150 rpm per minute, or 60 cycles, 180 rpm with no major change in the apparatus except the installation of runners suited to the particular speed. The author believes a better comparison with the Grand Coulee turbines may be obtained by the use of data pertaining to other Boulder Dam turbines which were designed for and are now operating at 60 cycles, 180 rpm.

Using these data for the same items as given in Mr. Terry's comparison, the following values for the Boulder Dam turbines will obtain:

Rated unit, bhp	115,000
Rpm	180
Rated head, ft	480
Specific speed	27
Rated discharge, cfs	2,340
Inlet diameter of casing, ft	10
Number of casing sections	5
Weight of casing, lb	381,000

Maximum operating head, ft	590
Casing test pressure, psi	500
Diameter of wicket-gate circle, ft	15.625
Nominal diameter of runner, in	$163^7/_8$
Height of wicket gates, in	21
Diameter top of draft tube, ft	10.83
Pit diameter, ft	20.75
Diameter of shaft, in	36
Length of shaft from center line of turbine to face of generator coupling, ft	26
Diameter of coupling between turbine and intermediate shafts, in	$61^3/_4$
Size of main bearing, in	$36^1/_2 \times 29$
Servomotor capacity, ft-lb	324,000
Operating ring torque, ft-lb	2,320,000
Total turbine thrust, lb	600,000

In conclusion the author wishes to thank Mr. Terry for his very interesting discussion. He also wishes to express his pleasure in being associated with him and his company in the design of the Grand Coulee turbines.

Economic Draft-Tube Proportions

By A. R. DAWSON,[1] TORONTO, CANADA

The purpose of this paper is to illustrate a new method whereby the most efficient and also the most economical proportions of a draft tube for a water-turbine installation may be found. A study is made of the excavation and concrete costs, as the ratios of draft-tube width to runner diameter and draft-tube depth to runner diameter increase. These costs are compared to the actual capitalized value of the horsepower saved due to the increased efficiency, as either ratio is increased. The comparison is made by means of computed curves and includes the ordinary range of power costs, as well as a range of load factors from 50 to 100 per cent, at which the unit might be expected to operate. The paper develops a method to overcome the seemingly prevalent practice of many turbine manufacturers of installing a certain definite size and shape of draft tube with a certain size runner without regard to the various factors which should enter into the selection.

THE tendency in the design of hydraulic-turbine draft tubes has been to produce draft tubes of highest possible efficiency. This tendency on the part of manufacturers to improve the performance of their product is entirely understandable and is appreciated by the purchasers. While manufacturers and users have undoubtedly given some thought to the economics of draft tubes, the literature dealing with this phase of the subject is but meager. It must be recognized that draft tubes can be designed and built with very high efficiency if no limitations on cost are imposed. On the other hand, there are many instances in which the conditions existing at a given power-plant site make it economically impossible to use draft tubes of the highest efficiency. This paper suggests one method of procedure by which engineers may estimate how far it is feasible and economical to go in the design of a draft tube for a given installation.

The method proposed herein is one which is well known to engineers and has been applied to many other problems, of which may be mentioned the selection of the proper diameter for hydraulic-power-plant penstocks. In essence, the method adds together the cost of construction and the capitalized value of maintenance and of the power lost through inefficient operation and other causes and, by selecting that design which shows the lowest total capitalized cost, indicates the most economical design.

In the case of draft tubes, there is no one design which is generally accepted by all manufacturers, nor is there one design of draft tube which is best for all types of hydraulic turbines. As a generalization, however, it may be stated that the longer the draft tube, the more efficient it is likely to be. In the usual design of power plants, in which the turbine is set with a vertical shaft, the draft tube may be thought of as being divided into two lengths: (1) The vertical section from the outlet of the turbine runner vertically downward to the elbow; (2) the horizontal portion of the draft tube from the elbow to the discharge end. It is not the author's intention to suggest that the data which will be used here, and which will serve to relate these two lengths of the draft-tube and machine efficiency are generally applicable, but they will serve to supply data from which an illustrative example can be worked out. The unit for which this study is made is shown in Fig. 1, which represents a proposed unit for a hydroelectric power plant operating under a head of 68 ft and generating 16,000 hp at 115.4 rpm. The diameter of the Francis-type runner is 12 ft and the specific speed is 74.8. The top of the draft tube is assumed to be 12 ft in diam.

From studies which have been made available, curves have been drawn to show the relationship of the turbine efficiency to the proportions of the draft tube. Two curves are available, the first, Fig. 2, showing the relation of the efficiency change to the ratio of the diameter of the turbine and the vertical distance from the bottom of the runner to the bottom of the tube, the latter distance being hereafter called the depth of the tube. The second curve, Fig. 3, shows the relations between the efficiency change, and the ratio of the diameter of the runner to the width of the draft tube at exit. For both curves, a dimensionless parameter was used and efficiency drop was chosen for plotting. It should be noted that the efficiency drop was used rather than actual efficiency, i.e., this is the loss of efficiency occasioned by too short or too narrow a draft tube, as compared with a deep or a wide one. These data have been obtained from the average of three different analyses, due to W. J. Rheingans on Francis-type runners, to F. Nagler on all types of runners, and to the N.E.L.A. Hydraulic Power Committee tests on straight draft tubes. The actual data were not directly available to the author, but were supplied through the courtesy of Mr. Nagler.

Effect of Variations in Draft-Tube Depths

The curve Fig. 2, giving the relation between the ratio $A = \dfrac{\text{Draft-tube depth}}{\text{Runner diameter}}$ and the loss in efficiency, is based on data from various sizes and shapes of turbines and draft tubes and has been assumed to be generally applicable to the unit being discussed. This curve shows that as A increases from 1 to 7, the loss in efficiency decreases from 4.65 to 0.05 per cent, i.e., there is an increase in turbine efficiency of 4.6 per cent. Applying this result to a 16,000-hp turbine operating at 100 per cent load factor, shows that the actual power lost by varying the depth of draft tube over this range changes from 744 hp to 8 hp, and in Table 1 is shown the power loss corresponding to intermediate values of A.

TABLE 1 POWER LOSS CORRESPONDING TO INTERMEDIATE VALUES OF A[a]

Ratio A	Depth of tube, ft	Efficiency loss Fig. 2, per cent	Horsepower loss 100 per cent load factor	Revenue deficiency at $10 per hp
1	12	4.65	744	$7440
2	24	2.32	371	3710
3	36	1.28	205	2050
4	48	0.78	125	1250
5	60	0.50	80	800
6	72	0.25	40	400
7	84	0.05	8	80

[a] Ratio $A = \dfrac{\text{Draft-tube depth}}{\text{Runner diameter}}$.

Based on a price of $10 per hp per annum, the revenue deficiency ranges from $7440 to $80, as is indicated in Table 1. If these sums are capitalized on a 5 per cent basis, they correspond to values ranging from $148,800 to $1600.

[1] Senior student in Mechanical Engineering, University of Toronto. Student Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

Although the efficiency increases as the draft tube is made deeper, it is obvious that for each installation there is a point at which extra cost of excavation and construction more than offsets the saving due to the decreased loss of horsepower. To find this point, an estimate of the costs of excavation and concreting must be made. The method of analysis will be illustrated by applying it to the plant shown in Fig. 1, which corresponds to $A = \frac{25}{12} = 2.08$. The depth of draft tube is shown as 25 ft, but the excavation should be carried lower, and it is estimated that 3 ft would be suitable.

In order to arrive at the actual cost, the volume of the excavation and of concrete to be placed should be determined with care, by means of drawings showing quite closely the finished dimensions of the work. However, in this case the method of procedure is being described and certain assumptions have been made so as to enable this to be done; the author does not suggest that these assumptions are particularly exact.

The horizontal length of tube is 37 ft from the center line of the turbine and it is estimated that the excavation would have to be carried 10 ft to the left, making 47 ft horizontal distance in all. The width of the tube is shown as 42 ft, made up of two 15-ft widths for the tube openings, a 5-ft division wall, and 3.5 ft of concrete between the tubes and the rock face. The author has followed the practice of certain designers in assuming that the rock must be blasted away in front of the tube, at an angle of 45 deg to the level of the top of the tube, this clearance being necessary to facilitate removal of the rock.

The total excavation would therefore be $(47 \times 42 \times 25) + (3 \times 47 \times 42) + (^1/_2 \times 25 \times 25 \times 42) = 68{,}397$ cu ft = 2533 cu yd. Rock excavation has been assumed to cost \$4 per cu yd,[2] considering average difficulties involved, including reasonable location of the plant, watering costs, and the cost of labor and transportation. Further, a calculation for this case indicates that the concrete placed is approximately 75 per cent of the volume of rock excavated, and the price of concrete in place has been taken at \$22 per cu yd,[2] including reinforcement, aggregates, transportation, labor, and form work.

For the value of $A = 2.08$, therefore, the cost of the tube is readily found, but it should be pointed out that design and engineering costs are not included. In the same way the construction costs for ratios A from 1 to 7 have been calculated and are set down in Table 2.

To compare the relative merits of the different depths of tube, the method of total capitalized cost is used to combine the cost of the tube and the revenue deficiency, because the sum of the effects of these two factors should be a minimum for the best tube. In calculating the total capitalized costs the following assumptions have been made:

1 That the useful life of the tube is 20 years and that it will be replaced at 20-year periods to perpetuity. This time may be thought too short, but experience shows that, for one reason or another, the tubes are replaced in from 20 to 30 years. Assuming a straight-line depreciation of 5 per cent, the capitalized value of the annual payments to a fund to provide replacement in 20 years would be equal to the first cost.

2 That annual repair and maintenance are 2 per cent of the original cost. The capitalized value of these annual payments to perpetuity at 5 per cent amounts to $^2/_5$ = 40 per cent of the construction cost.

3 That the capitalized value of the annual power loss, that is, the revenue deficiency, is 5 per cent to perpetuity, which would amount to 20 times the values in the last column of Table 1.

Based on the 16,000-hp turbine, taken as an illustration, running at 100 per cent load factor and with power selling at \$10 per hp per year, the values shown in Table 2 are obtained.

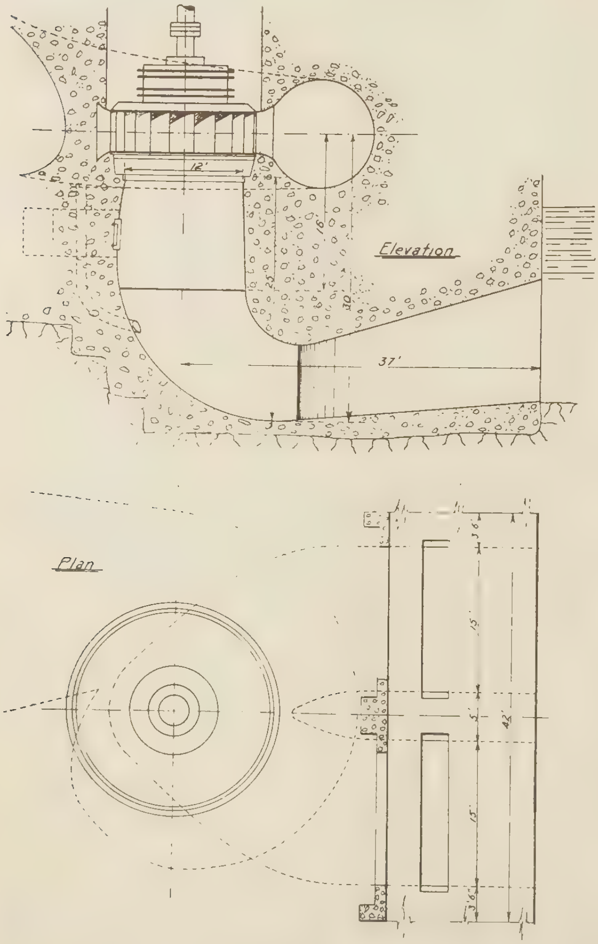

Fig. 1 Proposed Hydroelectric Unit for Which Economic Study Was Made

TABLE 2 CONSTRUCTION COSTS FOR RATIOS OF A[a] FROM 1 TO 7

Ratio A	1	2	3	4	5	6	7
				\$1000 units			
Construction cost..	24.6	49.5	79.0	112.8	151.3	194.3	221.5
Depreciation fund..	24.6	49.5	79.0	112.8	151.3	194.3	221.5
Repair and maintenance fund.....	9.8	19.8	31.6	45.1	60.5	77.7	88.6
Capitalized value of power loss, Table 1...............	148.8	74.2	41.0	25.0	16.0	8.0	1.6
Total capitalized cost............	208.0	193.0	231.0	296.0	379.0	474.0	533.0

[a] Ratio $A = \frac{\text{Draft-tube depth}}{\text{Runner diameter}}$.

Now, the selling price of power in Canada falls within the limits of \$10 to \$40 per hp per year, with very few exceptions, so

[2] The values of concrete and excavation costs are averages for existing plants of the Hydro-Electric Power Commission of Ontario, Canada.

(a) Chats Falls plant on Ottawa River, Ontario, for eight 28,000-hp turbines under 53-ft head, concrete \$15 per cu yd; excavation \$2.50 per cu yd.

(b) Ragged Rapids plant on Musquash River, Ontario, for two 5000-hp turbines under 38-ft head, concrete \$25 per cu yd; excavation \$6 per cu yd.

(c) Alexander plant on Nipigon River, Ontario, three 18,000-hp turbines under 60-ft head, concrete \$24 per cu yd; excavation \$4 per cu yd.

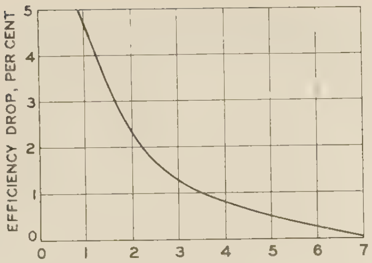

Fig. 2 Curve Showing Decrease in Efficiency Drop as Ratio A Increases

$$\left(\text{Ratio } A = \frac{\text{Draft-tube depth}}{\text{Runner diameter}}\right)$$

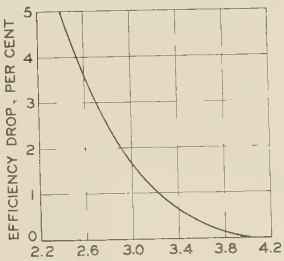

Fig. 3 Curve Showing Decrease in Efficiency Drop as Ratio B Increases

$$\left(\text{Ratio } B = \frac{\text{Draft-tube width}}{\text{Runner diameter}}\right)$$

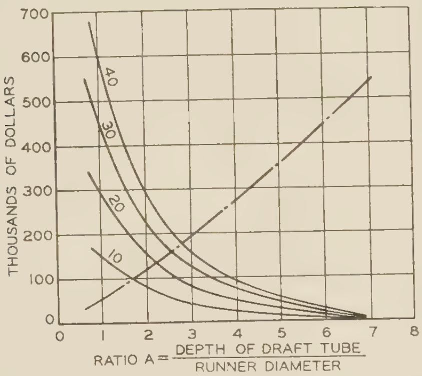

Fig. 4 Solid Lines Show Capitalized Value of Annual Revenue Deficiency at 100 Per Cent Load Factor and at \$10, \$20, \$30, and \$40 per Hp. Dotted Curve Is Total First Cost Plus Capitalized Value of Depreciation and Maintenance

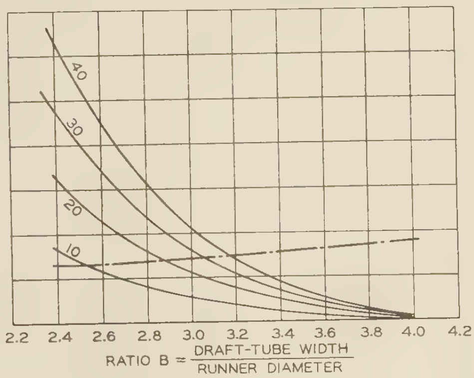

Fig. 6 Solid Lines Show Capitalized Value of Annual Revenue Deficiency at 100 Per Cent Load Factor and at \$10, \$20, \$30, and \$40 per Hp. Dotted Curve Is Total First Cost Plus Capitalized Value of Depreciation and Maintenance

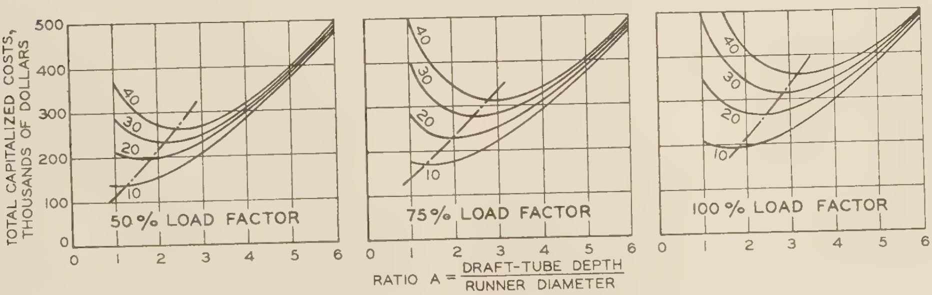

Fig. 5 Total Capitalized Costs for Three Load Factors and for Power at \$10, \$20, \$30, and \$40. Dotted Curves Show Best Tubes

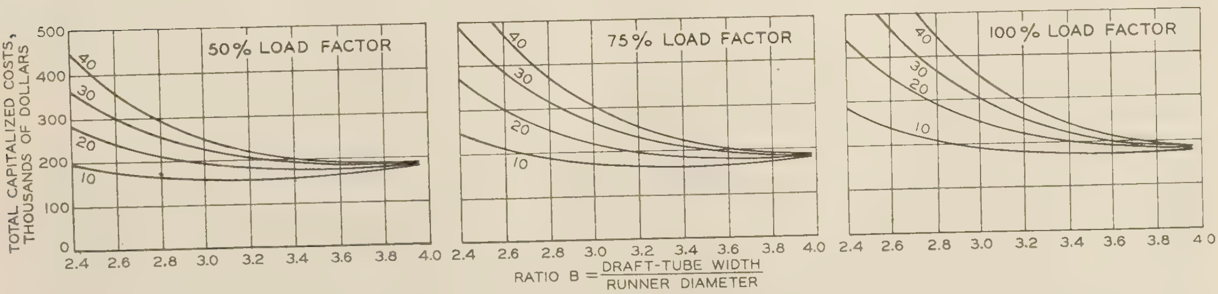

Fig. 7 Total Capitalized Costs for Three Load Factors and for Power at \$10, \$20, \$30, and \$40

that, by repeating the calculations for various power prices, a whole family of curves may be obtained in the same manner. For illustration purposes only, the values of $10, $20, $30, and $40 per hp per year have been dealt with in this paper. Also, since turbine units rarely run continuously at 100 per cent load factor, the entire procedure may be repeated for any desired load factor and, in this case, the calculations have been made for 100, 75, and 50 per cent load factors.

The curves, Fig. 4, show how the capitalized values of the annual horsepower losses or revenue deficiencies at varying horsepower values and 100 per cent load factor fall as the depth of draft tube increases. In the same figure, the total first cost and capitalized value of the depreciation and maintenance funds have been plotted on the dotted curve, which slopes in the opposite direction to the others. The curves in Fig. 5 show the sum of the items mentioned, giving the total capitalized costs for the four prices of power assumed and for 100, 75, and 50 per cent load factors.

Effect of Variation in Draft-Tube Width

This study is based on the results shown in Fig. 3, obtained from the same source as was used in Fig. 2, and shows the variation of efficiency drop with draft-tube width. From this curve, the results shown in Table 3 are obtained in a similar way to Table 1.

TABLE 3 POWER LOSS DUE TO VARIATION IN B^a VALUES

Ratio B	Draft-tube width, ft	Efficiency loss Fig. 3, per cent	Horsepower loss at 100 per cent load factor	Revenue deficiency at $10 per hp
2.4	28.8	5.00	800	8000
2.6	31.2	3.50	560	5600
2.8	33.6	2.50	400	4000
3.0	36.0	1.70	272	2720
3.2	38.4	1.12	179	1780
3.4	40.8	0.670	107	1070
3.6	43.2	0.350	56	560
3.8	45.6	0.150	24	240
4.0	48.0	0.025	4	40

^{a} Ratio $B = \dfrac{\text{Draft-tube width}}{\text{Runner diameter}}$.

In this phase of the calculation, the shape and dimensions of the sectional elevation at the top of Fig. 1 are assumed to be constant, the width of the tube alone changing. Proceeding in the same way as before, the rock excavation will be, for the case shown in plan in Fig. 1, i.e., $B = 2.5$: —$[(47 \times 28) + (^1/_2 \times 25 \times 25)] \times 42 = 68{,}397$ cu ft or 2533 cu yd.

Using the same excavation and concrete prices as before, the construction cost has been arrived at and tables similar to Table 2 made, and from these the results shown in Fig. 6 are obtained. Then in Fig. 7 the total capitalized costs for three load factors and for power at $10, $20, $30, and $40 have been plotted from which the most economical tube may be selected.

Conclusions

From the curves of Fig. 5, it is seen that the ratio $A = 1.8$, corresponding to 21.6 ft depth, would be the most economical if the unit is to be run at 100 per cent load factor most of the time and if the selling price of 1 hp is $10 per year. However, if it runs at 100 per cent load factor and the value of power is $40 per hp per year, then it would be more economical to use a depth of draft tube corresponding to a ratio A of 3.2 (i.e., depth $= 12 \times 3.2 = 38.4$ ft). Again, the ratio of 2.5 gives the most economical draft tube, if the unit is to be run at 75 per cent load factor and the value of 1 hp is $30 per year, while for 50 per cent load factor, the ratio 2.5 corresponds to power at $40 per hp per year.

Similarly, for this proposed unit, the ratio of draft-tube width to runner diameter is $B = 2.5$. Now, from Fig. 7, it is seen that the minimum points for the curves at $10 per hp per year are at a ratio B of 3.6, 3.4, and 3.2 for 100, 75, and 50 per cent load factors, respectively, giving the corresponding optimum widths of 43.2 ft, 40.8 ft, and 38.4 ft. For values of power of $20 per hp per year, or greater, the curves all approach a minimum at a ratio of $B = 4$ corresponding to a width of 48 ft. Thus, any width of draft tube greater than 48 ft for this turbine would not give additional efficiency or economy. Of course this is not the rule with any draft tube, since for other units with different horsepower ratings, first costs, or load factors, these curves might give definite minimum values such as were found previously with regard to the change of depth of draft tube. In this case, the most economic width could be fixed definitely.

There are many factors which affect the shape of these curves, both for variation in depth and variation in width but, in general, the curves take the same form, although the minimum points cover quite a wide range of ratios. For example, taking the unit studied in this paper as a basis, these facts become evident:

1 For an installation of greater rated power than 16,000 hp, the minimum points would move to the right, i.e., the more economic tubes would be both deeper and wider.

2 For higher selling prices of power, the minimum points would move to the right.

3 For higher interest rates than 5 per cent, the minimum points on the curves would move slightly to the left, but would not be changed greatly.

4 For higher costs of excavation and concrete, the minimum points would move to the left.

5 For lower load factors, the minimum points would move to the left.

6 For higher maintenance and repair values and for a shorter estimated life of tube, the minimum points would move to the left.

For a proposed installation, the rated horsepower is known, the costs of excavation and concrete can be closely approximated from boring tests and from the known location of the proposed unit or plant, and the selling price of power is usually well defined, as is also the existing rate of interest. Thus, the only factors which have to be assumed before being able to complete the foregoing calculations are the expected life of the tube, which in most cases is the same as the expected life of the entire plant, the expected annual repair and maintenance costs, and the expected load factor at which the unit will be operated.

It can be readily seen that these calculations and curves would be most valuable where (*a*) the limiting depth of the tube is fixed due to some irregularity of the rock formation from which excavation must be made for the draft tube, in which case the most economical width could be found; or (*b*) the width of the tube is fixed, as is more usual, due to limitations on the possible over-all length of the plant, in which case the most economical depth could be found.

Acknowledgments

Grateful acknowledgment is made to Mr. F. Nagler of the Canadian Allis-Chalmers, Limited, whose suggestions formed the basis for this paper, and also to Professor R. W. Angus, whose technical advice was greatly appreciated.

Discussion

F. Nagler.[3] This paper is commendable, in that it presents a type and completeness of analysis which the writer believes has not previously appeared in any of our technical publications.

The method of capitalizing efficiency losses seems to be slightly intricate, but Table 1 brings the results down to a basis that is definitely conclusive. Certainly, this method of arriving at a

[3] Chief Engineer, Canadian Allis-Chalmers, Limited, Toronto, Canada. Life Mem. A.S.M.E.

conclusion as to economical proportions of draft tube seems definitely more sound than the all too prevalent method of accepting blindly a proposal drawing that is usually an arbitrary picture based on precedent of the manufacturer. It may be admitted that a given draft tube, showing up well with a particular runner, may thereafter be especially fostered by the manufacturer and sold indiscriminately to the buying public. It is far from certain that such a tube will necessarily work well with even a different type of runner by the same manufacturer and even less so with a diverse design from another builder.

Naturally, the significant basis of this paper is found in the efficiency-variation curves, Figs. 2 and 3. It seems to the writer that the former should be definitely asymptotic, although Fig. 3 might not necessarily be so. Probably these curves are more open to criticism than anything else in the paper, as some such curves inherently form the basis of this kind of analysis. If the writer recalls correctly, the N.E.L.A. at one time sponsored or at least considered correlating the data from all manufacturers to arrive at some conclusion along this line. There seems to be no doubt whatsoever that there is a definite relationship between each of the three factors, tube height, tube length, and tube width, and efficiency loss. The losses must increase as the vertical height of a tube is decreased, all other things being equal. Whether the exact shape in this paper is correct or not is probably secondary to the rather definite method the author presents for arriving at economical proportions.

It might be in order to suggest that there is probably a complicated relationship between horizontal length and vertical height. This indicates the possibility of dealing with total length, which would be the combination of these two dimensions. The Ryburg-Schworstädt draft tube is a striking example of a tube which has a major portion of its diffusion section in its horizontal run. The writer doubts, however, that even such an extreme example detracts very definitely from the indication of the author's curve, Fig. 2, as it is extremely likely that its efficiency would be increased if its vertical run were longer. Some studies as to the combination of vertical and horizontal lengths seem to be in order.

Fig. 5 is of interest in showing graphically and rather strikingly the effect on draft-tube proportions that is exerted by value of power, and similarly illustrates the striking effect of load factor. This showing seems unique in indicating so concretely a definite difference in draft-tube proportions for changes in factors that usually enter not at all in fixing draft-tube proportions. It would be of interest, but probably very disappointing, if inquiry were made as to how many American plants had their draft-tube proportions influenced by such factors.

The author's conclusions are rather striking in their emphasis on the economical elasticity of draft tubes. The reason for conclusion No. 1, however, is not quite evident and further comment by the author on this feature would be of interest.

Arnold Pfau.[4] It may be said that the author's draft-tube investigations start with the assumption that the top of the draft tube is at a fixed elevation, so that costs due to depth of excavation are dependent only upon the selection of this depth.

The writer would point out that this top elevation, i.e., the setting of the turbine above the lowest tail-water level, depends upon and is materially affected by several factors, such as elevation above sea level and above all on the velocity of the water entering the draft tube. As is well known, the latter is related to the specific speed of a unit.

The selection of the type of turbine thus affects the depth costs materially. This is especially pronounced with the propeller-type turbine, having inherently very high entrance velocities of water into the draft tube and, thereby, requiring at the start a lower elevation of the top of the draft tube to minimize cavitation.

It seems that the use of Kaplan-type turbines for high heads has been somewhat overadvertised by our European competitors under Kaplan license. We find that an unbiased investigation as to economical costs of a Kaplan type and a Francis type discloses that, at certain heads and capacities, a Francis-type hydroelectric installation proves more economical as to over-all costs than a Kaplan type, for the following reasons:

1 For higher and higher heads, particularly exceeding 100 ft, the specific speed of a Kaplan type must be moderated, so that in fact its higher speed does not offer a material saving of generator costs over that of a Francis unit.

2 Likewise, the setting above (or better said, below) lowest tail-water level becomes very expensive due to excessive excavation.

3 The relatively higher percentage of overspeed above that of a Francis type involves a more costly generator.

4 The vertical thrust of the Kaplan type involves a more expensive thrust bearing.

5 For reasons of stability of speed, the inherent flywheel effect of a Kaplan-type hydroelectric unit must be materially greater than that of a Francis type. It is thus evident that all these factors will lead to a dividing line where the over-all costs point unmistakably to the selection of one or other type unit.

This paper as such is of value since it points out that, in the design of the draft tube, its costs must be weighed against the actual gain by reason of greater output due to higher efficiency. It is a case parallel with other design factors. For instance, a turbine runner with 0.25 per cent improvement in efficiency but with a greatly shortened life, due to cavitation, may prove less economical than a runner moderately less efficient but involving no outage and subsequent repair costs.

F. Schmidt.[5] The fundamental principles of the most economical draft tube must be applied to each individual case. One set of curves based on a given draft-tube design for a low-specific-speed Francis runner and prevailing construction conditions may not apply to a high-speed propeller-type runner under the same field conditions. Therefore, for each hydro project, a careful study of the draft tube must be made to determine the most economical design.

J. D. Scoville.[6] The principle that turbine efficiency increases with draft-tube depth and width is generally recognized. There are limitations, however, which make it difficult to use this fact in determining the economic proportions of draft tubes.

1 In a great many cases the draft-tube width is fixed by the scroll proportions and is not subject to an analysis such as the author's. The draft-tube depth is likewise controlled in some cases by the character of the material which must be excavated.

2 It is possible to offset deficiency in draft-tube depth by variations in design. It should be remembered that the tailrace excavation is to be considered as well as that for the draft tube. Fig. 8 of this discussion shows two draft tubes having identical area curves but different proportions. The width of both of them is the same. Draft tube B shows better efficiency than A, having more vertical height but a steeper upward slope of the horizontal leg. This might mean enough saving due to decreased tailrace excavation to offset the greater depth at the elbow. Deficiency in draft-tube depth can likewise be made up cheaply by

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[6] Assistant Chief Engineer, S. Morgan Smith Company, York, Pa. Mem. A.S.M.E.

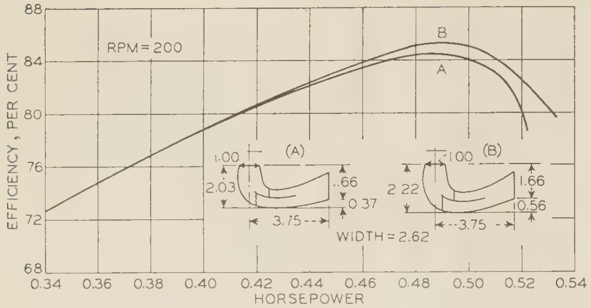

Fig. 8 Two Draft Tubes Having Identical Area Curves but Different Proportions

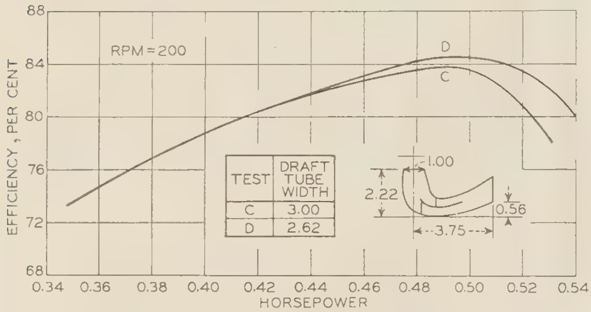

Fig. 9 Efficiency Curves Where Splitter Is Used in Horizontal Leg of Draft Tube

increased horizontal length, especially if this part of the tube is sloped upward.

3 The curve in Fig. 3 of the paper, showing increased efficiency with greater draft-tube width, might be correct for one manufacturer but is not right for all. By the use of a splitter in the horizontal leg of a draft tube, it is possible to offset the loss in efficiency due to a narrow tube. Fig. 9 of this discussion illustrates this graphically. In this case, it will be seen that the narrower of the two tubes shows the better efficiency. The area curves of these two draft tubes were identical.

The writer feels that the rigid application of economic principles to draft-tube design is too complicated to be feasible.

W. M. White.[7] We are indeed indebted to the author for having presented this paper on the importance of draft-tube design as regards efficiency and financial return on hydroelectric developments.

[7] Manager and Chief Engineer, Hydraulic Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

The writer does not agree with some of the conclusions which may be drawn from the data presented in the paper, but it is valuable to have this paper on record as a medium through which we may all offer contributions that will lead to a clarification of the all-important subject of depth of excavation for draft-tube design.

Author's Closure

In answer to Mr. Nagler's query, concerning the reason for conclusion No. 1: "For an installation of greater rated power than 16,000, the minimum points would move to the right;" it probably would have been clearer to have added, "provided all other factors affecting the curves remained constant." It is evident that, if the calculations had been made for a unit greater than 16,000-hp capacity, then the power loss, the revenue deficiency, and consequently, the capitalized value of the revenue deficiency would also be correspondingly greater. However, the total first cost and the capitalized value of the depreciation and maintenance would remain the same with the other factors remaining constant. Thus, when added together, the minimum points of the curves would move to the right of their former position showing that, with increased capacity, the more economic draft tubes would be both wider and deeper.

With regard to Mr. Scoville's limitation No. 1; the calculations would be of the greatest value where one of the dimensions of the tube was fixed due to some outside factor, such as certain required scroll proportions. In this case, the most economic depth could be found as discussed in the first part of the paper.

Limitation No. 2 is easily taken care of since the principles are flexible enough to allow the inclusion of various costs such as tailrace excavation which would not occur in every case. No doubt the actual calculations in practice would vary somewhat with each design but the method would remain basically the same.

As for Mr. Scoville's limitation No. 3, it must be pointed out that the theory put forth in this paper must be attempted only after the general type of draft tube has been decided upon. The shape of the efficiency-drop curves, as shown in Figs. 2 and 3 of the paper, is not changed materially over a wide range of tubes, varying from the straight-vertical type to the sharp-elbow type. However, these experimental data were obtained prior to 1927, at which time the use of a horizontal splitter was comparatively unknown on this continent. It is quite possible that a splitter might change the shape of these curves but this in no way detracts from the usefulness of the principles set forth in the paper. It is quite conceivable that another pair of average curves could be drawn for various shaped draft tubes containing splitter plates. In this case the most economic draft-tube proportions could be found as before, but based on the new experimental curves. This is just another example of one of the many fields which remain to be investigated in this study of economic draft-tube design.

Some Performance Characteristics of Deep-Well Turbine Pumps

By R. G. FOLSOM,[1] BERKELEY, CALIF.

Test data are presented on a comparable basis to indicate the relative performance of a series of deep-well turbine and propeller pumps. The range of performance characteristics obtainable with typical semiopen and closed impellers is shown and the losses introduced by axial adjustment of impeller-bowl positions are briefly discussed.

THE centrifugal pump in all its many and varied forms has reached its high degree of perfection principally through trial-and-error methods of development. Although analytical analysis has been of value in the development of specific units, this process is restricted in application, due to the complicated flow conditions encountered in the centrifugal-pump impeller and case. The trial-and-error procedure does not insure the production of pumps of maximum possible efficiency but, from a commercial viewpoint, satisfactory results have been obtained through its application.

Since modern centrifugal pumps attain very high efficiencies, further research and development work will be of the most careful and painstaking type in order further to increase the performance. These investigations will include the study of isolated phenomena which control the flow through various portions of the pump. Such investigations are being pursued in many laboratories. Recent publications demonstrate that, in the past, various factors have been neglected which may have an appreciable effect on the measured performance of centrifugal pumps.

Detailed studies of various pump phenomena are being made at the pump testing laboratory of the University of California.[2] Certain results obtained from these investigations will be presented and briefly discussed in this paper. The work of this laboratory deals principally with vertical-shaft units, particularly the deep-well turbine and the propeller pump. Both of these types are examples of specialized centrifugal pumps, but the results obtained from the investigations will apply equally well to all centrifugal pumps. Some data were obtained on normal horizontal-shaft centrifugal pumps which are under investigation in the laboratory on a limited scale.

Comparison of Over-All Performance

The deep-well turbine pump has been developed to the stage where extremely high efficiencies are obtained for the design restriction of the diameter of the hole in which the unit is to be placed. Although pumps of this class appear to be essentially the same upon superficial inspection of the types of impeller flow passages, areas, and other features, the laboratory experiments demonstrate that the performances of units from various manufacturers vary over a wide range. In order to compare units of different design, shape, and size, use will be made of the term "specific speed," which is defined by the equation

$$n_{sq} = \frac{\text{rpm}\sqrt{\text{gpm}}}{H^{3/4}}$$

where

H = total head per stage developed by pump, expressed in feet of fluid being pumped.

The specific speed expresses the requirements for dynamically similar operation of all geometrically similar centrifugal pumps. In practice, the term "specific speed" has been found to be useful as an indication of the type of impeller required to meet certain pumping conditions. The specific speed always refers to the point of maximum efficiency of the pump-performance characteristic. The plotting of maximum efficiency versus the specific speed for many pumps throughout a wide range of specific speeds, and drawing the maximum envelope for these points will give a curve which indicates the maximum efficiency obtainable by a pump of a given specific speed. Such a curve was prepared by Hollander in 1937 for optimum efficiency of single-stage

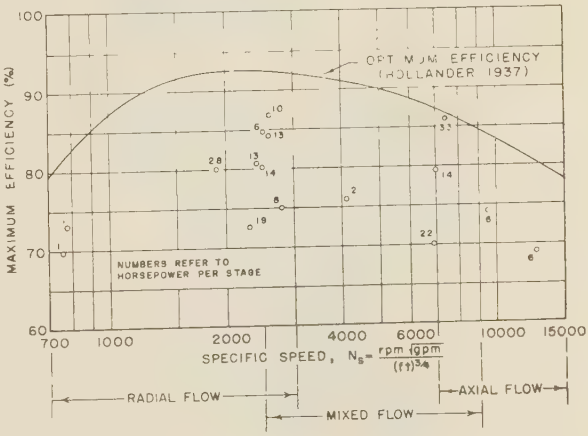

Fig. 1 Curve for Optimum Efficiency of Single-Stage Pumps; Points of Maximum Efficiency of Deep-Well Turbine Pumps

pumps and published by Daugherty,[3] and is reproduced in Fig. 1 of this paper. This particular curve applies to single-stage pumps of 12-in. nominal size. Due to hydraulic leakage loss and mechanical friction, larger pumps would be expected to show higher efficiencies and smaller pumps lower efficiencies.

The maximum efficiency of deep-well turbine units investigated by the pump testing laboratory is also plotted in Fig. 1. In order to indicate the relative size of the various units, the number alongside the plotted point refers to the horsepower per stage of the pump. It will be noted that the relative performance of commercial deep-well turbine pumps varies greatly; in fact, this limited series of tests indicates 15 points or more in efficiency as

[1] Assistant Professor of Mechanical Engineering University of California. Mem. A.S.M.E.

[2] "University of California Pump-Testing Laboratory," by R. G. Folsom, *Mechanical Engineering*, vol. 60, 1938, pp. 301–305.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., Sept. 3–6, 1940, of The American Society of Mechanical Engineers.

[3] "Hydraulics," by R. L. Daugherty, McGraw-Hill Book Company, Inc., New York, N. Y., 1937.

the difference between the best and poorest units. This spread indicates that the deep-well turbine is far from being a standardized product in which the performance of units from different manufacturers is very similar.

The efficiencies of deep-well turbine pumps are generally lower than those of normal horizontal-shaft units, as all hydraulic losses between pump entrance and immediately beyond the discharge elbow and mechanical losses of the drive shaft are charged against the pump. The laboratory tests were made on pumps with riser columns about 10 ft long, thus having small losses as compared with the usual field installation. A slight increase in the efficiency percentage can be realized as the pumps are multistaged to four or five stages. No consideration is given to this feature in this paper.

Fig. 2 Impeller for Radial-Flow Deep-Well Turbine Pump

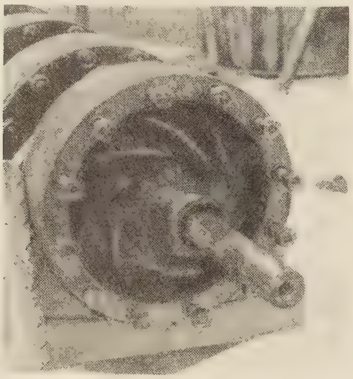

Fig. 3 Type of Semiopen Impeller for Mixed-Flow Pump Unit

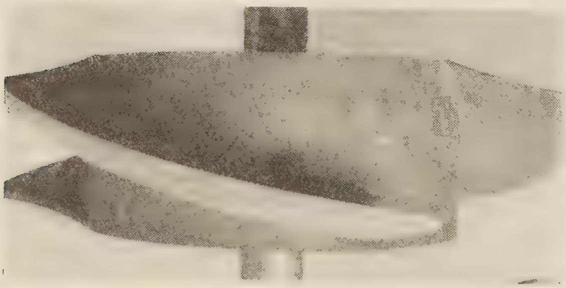

Fig. 4 Axial-Flow Unit or Propeller Pump

Deep-Well Turbine Pump Losses

There are many ways of classifying centrifugal pumps but, on the basis of the impeller shape and design, there are three principal types, namely, radial-flow, mixed-flow, and axial-flow units. The three characteristic types are illustrated, respectively, in Figs. 2, 3, and 4. These different types of impellers are produced with a variety of shapes and construction features. The radial- and mixed-flow impellers are used extensively in deep-well turbine pumps. The two types of construction usually adopted are the closed impeller, as illustrated by the radial-flow unit Fig. 2, and the semiopen impeller, as illustrated by the mixed-flow unit, Fig. 3. The axial-flow unit Fig. 4, which is sometimes referred to as a propeller pump, may have blades of widely varying shapes and areas. The principal impeller features other than

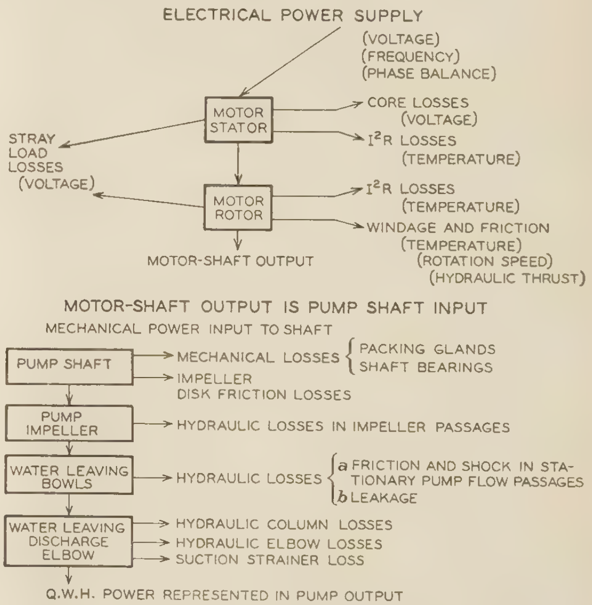

Fig. 5 Energy Flow Sheet and Losses for Typical Deep-Well Turbine Pump Installation
(Values in parentheses indicate variables controlling magnitude of losses.)

blade shape and passages are the methods of attaching the impeller to the rotating shaft, and the method of sealing the suction intake from portions of the pump under the approximate impeller discharge pressure. The methods of attaching the impeller to the shaft will not be discussed in this paper, as this is a purely mechanical problem and in no way affects the hydraulic performance of the unit so long as the relative position of impeller and shaft is fixed. Sealing methods vary widely and have a large influence on the hydraulic performance of the pump.

A centrifugal pump consists essentially of an impeller rotating in a housing with sealing glands on a rotating shaft to prevent external leakage into or from the pump, and with seals inside the pump between the high- and low-pressure areas. In general, the deep-well turbine pump has these features of the normal centrifugal pump but, in addition, it is located at the lower end of a flow column which surrounds the drive shaft, powered at the surface. In order to meet these conditions, special arrangements of impeller and bowl flow passages, guide and thrust bearings,

must be made. The chart, Fig. 5, indicates energy flow and losses for a typical deep-well turbine unit.

Axial Adjustment

The relative position of the impellers with respect to the pump bowls will be adjusted by the shaft nut at the motor coupling. For gear-head or other drive, provision is made for vertical adjustment of the shaft. This adjustment may be of primary importance in many types of deep-well turbines since it controls the leakage quantity which is a short-circuit loss in the pump, and is a direct loss in the performance of the unit as a whole.

The radial clearance between the wearing ring and the bowl is the principal control of the leakage quantity of a normal closed impeller. For this type, axial adjustment of the impellers has little or no effect on the pump performance. Some closed impellers are made without the usual "skirt" or wearing ring, but include a sealing surface which depends upon axial adjustment to control the leakage. The semiopen impeller depends upon a close fit between the blades and bowl to reduce leakage to a minimum.

In so far as possible, the results of all experiments have been expressed in dimensionless form. For example, the discharge rate

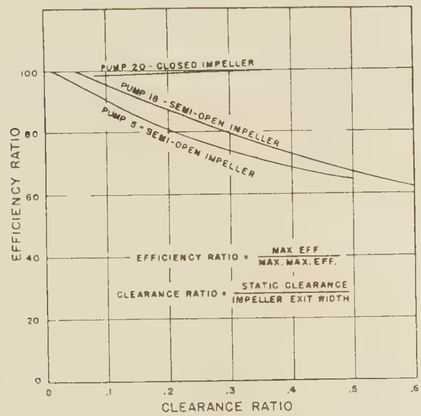

Fig. 6 Typical Clearance and Efficiency Results of Semiopen and Closed Impellers

at any point of operation is expressed as a ratio of actual discharge at that point with the discharge corresponding to the point of maximum efficiency. In a similar manner, clearances of impellers (the clearance being the axial distance between the lower portion of the impeller and the bowl) are expressed as a ratio of clearance between the bottom of the impeller and bowl to width of impeller flow passage at the outlet. Fig. 6 shows typical results of clearance and efficiency of semiopen and closed impellers. This graph demonstrates the relatively small change in performance of a normal closed impeller as compared with that of a semiopen impeller. Closed impellers with special seals for vertical-shaft adjustments have characteristics similar to all semiopen impellers. The work of this laboratory has shown that, when a normal closed impeller is adjusted with reasonable care to the middle of the adjustment range, the variations in performance for different adjustments are less than the error in the usual performance tests.

Fig. 7 illustrates the typical performance curves for semiopen impellers with various clearance ratios. Summing up the characteristics of deep-well turbine pumps, as affected by variations in axial adjustment, we find (1) the head, discharge, hydraulic thrust, power, and efficiency of normal deep-well pumps with normal closed impellers are not measurably affected by axial clearance within the possible range of axial adjustment; (2) at

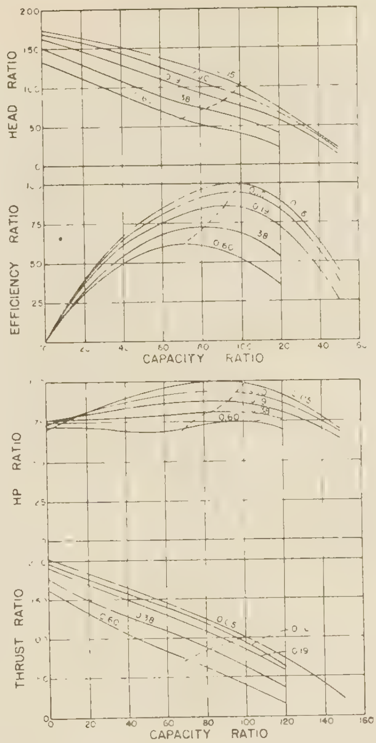

Fig. 7 Performance Curves for Semiopen Impellers With Various Clearance Ratios

(Figures on curves refer to clearance ratio; dotted lines represent conditions at maximum efficiency.)

constant head, the efficiency, power input, hydraulic thrust, and discharge of deep-well pumps with semiopen impellers decrease with increase of axial clearance; (3) closed impellers, equipped with sealing devices subject to axial adjustment, exhibit performance characteristics similar to those of semiopen impellers.

The foregoing discussion is restricted to new pumps and in no way considers the application of specific designs to conditions where wear, due to abrasive particles in the water, is of importance. These data are presented to call attention to the fact that correct adjustment of some types of design is necessary to obtain and to maintain the highest possible efficiency. These conclusions apply equally well to horizontal-shaft or other centrifugal pumps having impellers similar to those described.

Conclusions

The performance of similar commercial deep-well turbine pumps varies over a considerable range. Further tests may indicate a greater range of divergence.

A clear understanding and a careful study of losses will be necessary for a further improvement in performance of present high-efficiency units.

Initial and maintained high efficiency of centrifugal pumps requires careful axial adjustment for certain types of impeller construction. No data are presented regarding the change in performance with wear, a factor which may be of primary importance in some installations.

Acknowledgment

The author wishes to acknowledge assistance furnished by the personnel of Works Progress Administration Official Project No. 65-1-08-113, under the sponsorship of the department of mechanical engineering, University of California.

Discussion

J. W. Daily.[4] It has been observed in the hydraulic machinery laboratory of the California Institute of Technology that some pump designs of the horizontal-shaft, medium-specific-speed type show definite improvements in head and efficiency when subjected to increased submergence. For a few pumps, this improvement persists even after inlet heads of 50 ft and above are reached. The important conclusion from such results is that many pumps designed for and installed with relatively large submergence margins must actually operate under cavitating conditions. Consequently, in tests made for research purposes, adequate representation of pump performance cannot be made without reference to the submergence.

The author describes tests of deep-well turbine pumps and compares them on the basis of over-all performance. No information is given as to the method of testing, but an earlier article by the author[5] indicates that variable-submergence tests are possible. However, the absence of any consideration of inlet pressures here leads to the conclusion that the tests are made at practically constant submergence. Referring to the plotted points of the author's Fig. 1, it is noted that there is a lack of system to the variation in maximum efficiency (for a particular specific-speed range). This allows some speculation as to existence of a behavior in deep-well pumps similar to that observed in the radial-flow, horizontal-shaft type. It would be particularly interesting to learn whether or not the performance of units with semiopen impellers of medium and high specific speeds would improve with increased pressure on the suction side. Submergence is likely to be important not only when operating with normal clearances, but also when the clearances are large. The writer recognizes that, practically, the physical setting dictates the submergence but, as the author has indicated, improvement in design dictates painstaking investigation for optimum results. It is felt that a quantity of such information on a variety of designs should be useful to the designer.

J. M. Hait.[6] In my opinion, this paper, concerning the results of tests on deep-well turbine pumps, represents the most accurate and unbiased report on relative performances which has yet been published. The general results agree very well with similar tests which we have made with entirely different laboratory equipment. There are a few comments the writer would like to make concerning the presentation of these test results:

In Fig. 1 of the paper, the maximum efficiencies of several types of deep-well turbines have been plotted against specific speeds. An optimum-efficiency curve is shown, which, in effect, indicates the degree of development of the pump tested. As far as we can determine, the pump used for the optimum-efficiency curve was a 12-in. horizontal centrifugal pump. The question arises as to whether it is the best basis for comparison, in determining degree of development of relatively small deep-well turbine pumps, since the radial space for conversion of velocity head is definitely restricted in the case of the latter, while the discharge from the impeller of the horizontal-type pump is at the plane of the impeller rather than normal to it.

The number of stages of the various units which were tested and plotted is not noted, but it is stated in the paper that a slight increase in efficiency may be realized by multistaging to 4 or 5 stages. It has been our experience that the additional efficiency by multiple staging varies with the class of pumps, that is, whether they are straight centrifugal or mixed flow and that, in the average case, pumps with 8 stages or above perform with an efficiency approximately 6 points higher than a single-stage pump of the same type. Thus, it seems that the number of stages tested is an important item when efficiencies are to be compared.

In showing the clearance between the semiopen impellers and the bowl, a clearance ratio is used. The writer feels that it would also be of value if the reader were to know the absolute clearances involved, in which he is dealing, and as to whether these clearances are those only obtainable by expert operators in a laboratory or whether they can be easily obtained by an average operator with field settings of the ordinary range of magnitude between 200 and 400 ft. The change in length due to the torque imposed on the shaft in combination with the hydraulic thrust, and the upward reaction of the water being turned as it enters the impeller create a change in the shaft length, complicating the field adjustment materially. Accordingly, the writer feels it to be of importance, in considering this study, to know the order of magnitude of the clearances involved.

In the summary of the effects of axial adjustments on the performance, it is stated (3) "closed impellers, equipped with sealing devices subject to axial adjustment, exhibit performance characteristics similar to those of semiopen impellers." It should be noted that this statement is not true when the closed impeller has both a close radial-skirt clearance and a seal at the bottom of the skirt with the customary axial adjustability.

M. Mull.[7] This discussion will be made from a commercial standpoint, pertaining to the issues discussed in this paper, and their influence upon the recommendation and application of different types of deep-well turbine and vertical wet-pit pumps as manufactured by our company. The writer's comments are based on his observations and experiences with this type of pumping equipment throughout the Northwest during the last 15 years.

The deep-well turbine can truly be called a western product, as it had its inception on the Pacific Coast. For this reason its development can easily be studied from many of the older installations, to determine the result of the various ideas on design and construction. It is true that many design and development methods have been employed to obtain the highly efficient deep-well turbine and other vertical modifications offered on the market today. As stated by the author, "the trial-and-error procedure does not insure the production of pumps of maximum possible efficiency," and it may be years before these actual results are accomplished from a truly commercial standpoint. Accuracy and long life in pumping equipment are achieved, particularly by eliminating so-called special features, using only conservative and time-proved designs.

It is not possible today, with modern manufacturing methods, to develop new designs under the regular schedule of production, because standard current modifications are used in order to speed up production and fill orders on definite delivery schedules. Development work is done on certain-size pumps which are found to be lacking in performance with competitive units of the same characteristics. Competitive bidding on the larger government and municipal specifications is responsible for keeping definite records of performance on the various sizes offered. Many of the individual points of design are constantly under investigation and upon the results are based new development programs from year to year. This development schedule is studied and a definite program laid out for each year, in order to keep abreast of competition.

Today it is possible to offer vertical close-coupled wet-pit-type

[4] Manager, Hydraulic Machinery Laboratory, California Institute of Technology, Pasadena, Calif. Jun. A.S.M.E.

[5] "University of California Pump-Testing Laboratory," by R. G. Folsom, *Mechanical Engineering*, vol. 60, 1938, pp. 301–305.

[6] Chief Engineer, Peerless Pump Division, Food Machinery Corporation, Los Angeles, Calif. Mem. A.S.M.E.

[7] Sales Engineer, Fairbanks, Morse & Company, Seattle, Wash.

pumps for application in the high-head field, with over-all efficiencies exceeding those of horizontal split-case-type centrifugal pumps within their range. This is even more evident where it is necessary to use multistage horizontal units. Fewer hydraulic losses are experienced in the completed installation using the vertical deep-well turbine pump, as compared with the use of horizontal multistage or series-connected pumps with all of the appurtenance piping.

The term "specific speed," which is applied in a nontechnical manner in the commercial field, is a watchword in the selection of pumps which will give satisfactory performance for the operating conditions specified. Propeller, axial-flow, mixed-flow and deep-well turbine pumps, as well as all other centrifugal pumps, have a specific-speed range in which they are designed to operate with quiet smooth performance and without cavitation, and at a minimum of submergence. It is always desirable to select pumps, particularly in the low-head field, which come well within their specific-speed range. However, there are times, in the application of such pumps, when it is necessary to sacrifice the most efficient pump, which is within the correct limits of specific speed for that one modification, because of the high first cost of the equipment. This is found to be true on extremely low-head jobs, where the best performance will often be found at extremely low revolutions, thus causing high cost of the electric motor. (Specific speed must not be confused with pump rpm.)

The author has expressed in an enlightening and graphic manner, the losses to be accounted for in the deep-well turbine pump. The Hydraulic Institute has also made these points very clear, and through this source standardization along this line has been effected to a large degree.

The observations, outlined in the paper, on the subject of "axial adjustment," are important, as they pertain to actual results from operation of deep-well turbine pumps. It has been found, with this type of pumping equipment, that enclosed impellers with a long "skirt" and arranged for a vertical adjustment of ample proportions, will give better efficiency in the hands of the average operator, than will other types of semiopen impellers.

The conclusion, offered in this connection, is not only in accordance with those found in the field, but substantiates the fundamental reason for this type of design. Where highly efficient well-built deep-well turbines are selected for applications requiring continuous service, the enclosed-type impeller should be specified. The construction throughout the bowl assembly should be of a high order of precision, with bronze bearings and renewable wearing rings both in the bowl casting and in the impeller skirt.

For deep-well turbines, which are required to change capacity throughout a small range, as is often specified in cases where the capacity of a well may be less at one season of the year than at another, the semiopen impeller is desirable. With this adjustment, the vane of the impeller is lifted from the seat in the bowl, thus allowing a greater clearance as described by the author. However, this effect to reduce the capacity is always a costly one as it decreased the efficiency of the pump.

The vertical deep-well pump, employing good standard forms of design which have been proved in service for the last decade, has now been developed to a point where it can compete in efficiency and cost with the horizontal centrifugal pump. It has found a very definite application for certain types of pumping requirements. The deep-well turbine should be carefully applied. When a pumping problem is under consideration, a definite analysis should be made to aid in arriving at an unbiased choice between the horizontal centrifugal pump and the vertical deep-well turbine. Many manufacturers of only one type of pumping equipment will almost invariably recommend and try to sell their products regardless of the correct application involved, which practice should be carefully guarded against by all engineers.

Author's Closure

The problem of the correct magnitude of submergence to obtain satisfactory centrifugal-pump performance is always present but becomes particularly troublesome for large-specific-speed units. Mr. Daily has called attention to this important variable, the discussion of which was omitted from the paper. Of equal importance are the fluid velocities, in magnitude and direction, at the impeller inlet. For the tests reported at the large specific speeds, values larger than 5000, the submergence varied from about 3 to 4 ft, and the other units had submergences from about 5 to 12 ft. All tests were conducted with the pump installed approximately in the middle of the laboratory 8 × 8-ft test pit.

Submergence has much less influence on multistage pumps than single-stage units as the first stage is the only one of the multistage construction that is affected to an appreciable extent by the pressures and velocities at inlet. Thus with the normal multistage deep-well turbine pump, the small change in performance at increases in inlet head becomes insignificant. With single-stage large-specific-speed units, the submergence is one of the limiting factors restricting application. On the basis of unreported tests, increases in efficiency of less than one per cent would be expected for these units with relatively large increases in inlet heads.

Results have not indicated appreciably different behavior with respect to inlet heads for semiopen impellers with different clearances. All semiopen impeller pumps tested were multistage and thus the phenomena would not be apparent, as it occurs in the first stage only.

Mr. Hait is correct in his conclusion that the optimum efficiency applies to 12-in. horizontal pumps. This curve was selected to indicate degree of perfection as it has been previously published, and sufficient data have not been made available to the author by deep-well turbine manufacturers to allow him to develop a generalized curve representing all companies. Some such curve should be used when making a comparison as the expected maximum efficiency of any type of centrifugal pump varies with the size and specific speed.

Mr. Hait indicates increases in efficiency through multistaging somewhat in excess of usual values and that appreciable increases occur for a larger number of stages. This increase in efficiency is a function of the pump design, size, and relative distribution of losses. Thus, if bowls only are considered, a different value of efficiency increase will exist than if the complete pump with riser column discharge elbow and suction piece are included. The number of stages for the units tested are the following:

Specific speed	Stages	Specific speed	Stages
760	6	2600	5
780	6	2600	4
1900	2	2800	2
2300	1	4100	10
2400	2	6900	1
2500	2	7200	1
2500	7	7400	1
		9500	1
		12700	1

Axial adjustment of deep-well turbine pumps is considered in some detail in a recent publication.[8]

One series of tests with an impeller having radial-skirt clear-

[8] "The Axial Adjustment of Deep-Well Turbine Pumps," by Morrough P. O'Brien and Richard G. Folsom, University of California Press, Publications in Engineering, vol. 4, no. 2, 1940, pp. 19–26.

ance and axial adjustability showed performance changes with axial adjustment similar to the semiopen impeller characteristics except that the magnitude of the performance change was much reduced.

Mr. Mull's remarks from the field standpoint are gladly received, although the author would like to point out that some of the conclusions are open to considerable argument. It is almost impossible to specify any one type of construction as a standard because of the wide variety of conditions to which these pumps are applied.

Centrifugal-Pump Performance as Affected by Design Features

By R. T. KNAPP,[1] PASADENA, CALIF.

This paper presents some of the results of a study of Grand Coulee pumping-plant characteristics. The research program was conducted for the Bureau of Reclamation by the California Institute of Technology in its hydraulic machinery laboratory (1)[2] and has been in progress since January, 1938. While the principal object was to determine the operating features for pumping units to be installed at the Grand Coulee project, the results obtained are somewhat more generally applicable than might be expected. It is believed by the author that investigations of a somewhat similar nature offer the most reliable means for securing the characteristics desired in hydraulic units, both pump and turbine, for practically any given set of conditions.

Need for Information

The need for a thorough study of the Grand Coulee pumping plant arises basically from the tremendous size of the units proposed, i.e., 1600 cfs capacity with a motor of approximately 65,000 hp for each pump. Full knowledge of the pump characteristics is required, due to the great range of operating head, from 295 to 367 ft, which is caused by the variation of the inlet head, from + 80 ft to + 5 ft. Also, the probable operating cycle makes it desirable to have as high a capacity as possible when operating against the high head. In addition to the matter of the proper relationship between the capacity and head over the operating range, the following items were considered to be important for satisfactory pump operation:

(*a*) Freedom from cavitation over the entire operating range

(*b*) Low radial forces due to hydraulic unbalance.

(*c*) Freedom from unstable regions within the operating range.

(*d*) Constant-speed operation.

(*e*) Satisfactory transient performance which will permit simple shutdown procedure.

(*f*) Suitable characteristics when operating as a turbine to provide the possibility of utilizing units for peak-load power development.

Furthermore, to obtain the lowest-cost unit, including the motor, it was necessary to determine the maximum permissible operating speed for which units could be obtained that could also satisfy the foregoing requirements.

Model and Prototype Pumps

The pumps contemplated for installation at Grand Coulee are unprecedented in size and power requirement. They are to be installed vertically and will be of the single-stage single-suction type. Each unit is expected to have a capacity of about 1600 cfs. The normal head against which it is to deliver is about 295 ft. Approximately 65,000 hp will be required. It is estimated that the pump for this duty will have a discharge nozzle of 8 to 10 ft diam, an impeller of from 12 to 17 ft diam, with an eye dimension of from 6 to 10 ft. The width of the impeller at discharge will be in the neighborhood of 20 to 36 in. The speed range is from 150 to 200 rpm or possibly slightly higher.

Although the present study is not a "model" study, but rather an investigation of the possible characteristics of the machines, the units tested in the laboratory may be thought of as models in order to visualize a size comparison. On this basis the model ratio would range from $12^1/_2$ to 15. The studies were all made at or near the full prototype head. The capacities varied from 7 to 10 cfs at the operating point. The horsepower requirements fell within the range of 290 to 400. All of the units had maximum efficiencies in the vicinity of 90 per cent. The discharge-nozzle diameters were 8 in. The impellers varied from $12^1/_4$ to $14^1/_2$ in. diam, with eyes of from 6 to 8 in. and with discharge widths of from $1^1/_2$ to $2^1/_2$ in. Testing speeds fell between 2100 and 2600 rpm.

It will thus be realized that these test pumps are comparatively large machines, therefore, accurate passages and vane angles may be expected. Furthermore, the large size and high efficiency of these units permit drawing direct conclusions concerning the performance of prototypes. To reduce the number of variables, several cases were designed to operate with the same impeller, thus making it possible to ascertain clearly the characteristic-performance differences between such case types as single-volute, double-volute, and fixed-vane-diffusor constructions.

Presentation of Data

In order to make the results from the different units directly comparable, the characteristic curves have been plotted on a percentage basis. The normal operating head at Grand Coulee is 295 ft. This has been taken as 100 per cent. The capacity at this head is therefore designated as 100 per cent. The maximum efficiency of each unit has been used as the reference value for that unit, and has been plotted as 100 per cent. It should be noted that the maximum-efficiency point will not coincide necessarily with the 100 per cent capacity and head point. Whenever plotted, torques and horsepowers have had, as a 100 per cent reference, the corresponding values at 100 per cent capacity and head. For example, since the prototype-head range is from 295 to 367 ft, this system gives an operating-head range of from 100 to 125 per cent.

Comparison of Normal Operating Characteristics

Capacity-Head and Efficiency Characteristics. During the course of this program, several series of experiments were made in which a single impeller was tested in two or three different cases. In order to establish a basis for the discussion of the results, a brief résumé of the respective functions of the impeller and the case of a centrifugal pump seems desirable.

The impeller adds energy to the fluid flowing through it. At the discharge from the impeller, this added energy is in two forms: (*a*) an increase in pressure, and (*b*) an increase in velocity. The case has two functions: (*a*) to collect the fluid as it discharges

[1] Associate Professor of Hydraulic Engineering, California Institute of Technology. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Hydraulic Division and presented at the Fall Meeting, Spokane, Wash., September 3–6, 1940, of The American Society of Mechanical Engineers.

around the impeller periphery, and (b) to transform a large part of the velocity into pressure with as little loss of energy as possible.

If an impeller could be tested alone under such conditions that, for all rates of flow, the discharge would be uniform around the periphery, its basic operating characteristics would be determined. A perfect case would be one which would have no losses over the entire operating range; therefore the combination of the impeller, operating in such a case, would have the identical performance characteristics which were obtained from the impeller operating alone. Since no real case is without losses and, furthermore, since no real case is equally efficient over the entire operating range, the performance of the unit as a whole is always lower than that of the impeller alone. The deviation will be least in the zone in which the case characteristics match the impeller

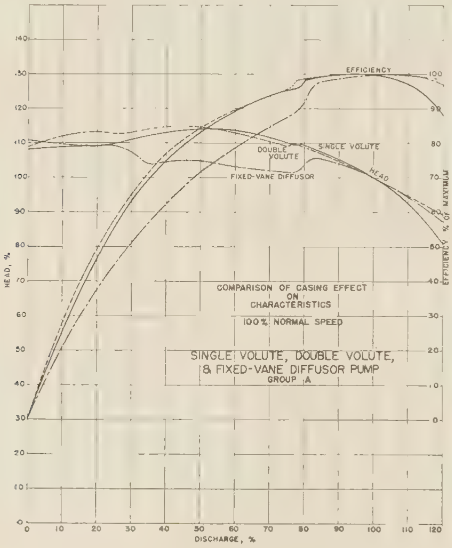

Fig. 1 Comparison of Casing Effect on Pump Characteristics; Group A, 100 Per Cent Normal Speed

characteristics to best advantage, and will increase on both sides of this zone. For a good pump, the case must match the impeller within the high-efficiency zone of the latter.

The case will affect the over-all performance of the pump in two ways: (a) through energy losses in the case itself, and (b) in additional energy losses induced in the impeller. Fundamentally, the case can affect the impeller performance only in one way, i.e., by varying the pressure distribution around the periphery of the impeller and thus producing nonuniform discharge. If the discharge is not uniform from all parts of the impeller periphery, it follows that there must be pulsating flow in the impeller passages, nonuniform entrance conditions at the eye, and presumably increased losses, both in the impeller and in the case. In this simplified picture, the secondary effects of the impeller shrouds, the circulation existing between them, and the casing walls and leakage losses to the suction sides are disregarded.

From this discussion, it will be realized that a comparison of the characteristics of different units, made up of various types of cases operating with the same impeller, resolves itself into a comparison of the relative matching of these cases to the impeller and of casing losses, both intrinsic and induced in the impeller. Fig. 1 shows such a comparison for a series of units designated as group A. The first unit was designed as a single-volute pump to operate at a prototype speed of 150 rpm. The double-volute case was then constructed, using the same design methods. It was anticipated that, if everything worked out satisfactorily, the performance of the double-volute pump would be the same as that of the single-volute unit. The fixed-vane-diffusor case was designed around the same impeller.

If the curves for the single- and double-volute cases are compared, a striking difference is observed in the high-capacity region. The head curve for the double-volute case does not fall off as rapidly as that for the single-volute pump, and the efficiency also remains higher. The same is true to a lesser extent in the low-capacity region. However, the maximum efficiency is about the same. Since these maximum efficiencies are high, both cases are very satisfactory in the region of the design point, but the double-volute case apparently matches the impeller characteristics better in the low- and high-capacity regions. It must be remembered that, because of the two passages, the double-volute case has a lower effective hydraulic radius and, therefore, a higher skin-friction loss. For this reason, the wide region of high efficiency is all the more surprising.

The fixed-vane-diffusor case, operating with the same impeller, shows the same high maximum efficiency observed in the other two cases. However, the characteristic curves are quite different in shape. The maximum-efficiency point comes at a somewhat higher capacity for the diffusor case, and this maximum efficiency is not sustained over as wide a region. This is reflected in the head-capacity curve. It will be noticed on both sides of the design point that the diffusor-case head curve lies

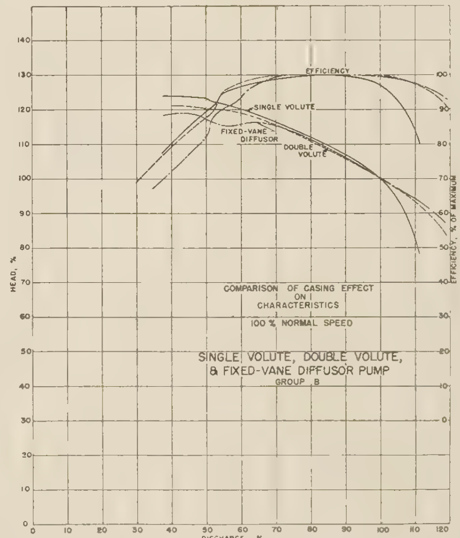

Fig. 2 Comparison of Casing Effect on Pump Characteristics; Group B, 100 Per Cent Normal Speed

under that of the double-volute curve. In the low-capacity region, i.e., from zero up to 75 per cent, the efficiency of the diffusor is markedly lower than that of the other two cases. This is probably the result of the discrepancy between the angle of the fixed guide vanes and that of the flow leaving the impeller under these conditions.

Fig. 2 shows the same comparison for an entirely different set of cases, working with another impeller. This series of units, group B, was designed for a prototype speed of 180 rpm in comparison with the 150-rpm speed of group A. Since the head and capacity are fixed, this 20 per cent increase in operating speed results in a 20 per cent increase in the specific speed as well, which corresponds approximately to a 16 per cent decrease in the diameters of the impeller and the base circle of the case.

The relative performances of the single- and double-volute cases are practically the same as those observed for group A, except that the single-volute pump shows its peak efficiency between 80 and 90 per cent of design discharge. This indicates that the case is too small for the specified conditions. The result is that, at the normal operating point, the efficiency is only about 97 per cent of the maximum. This accounts for the fact that its head-capacity curve is apparently above those for the double-volute

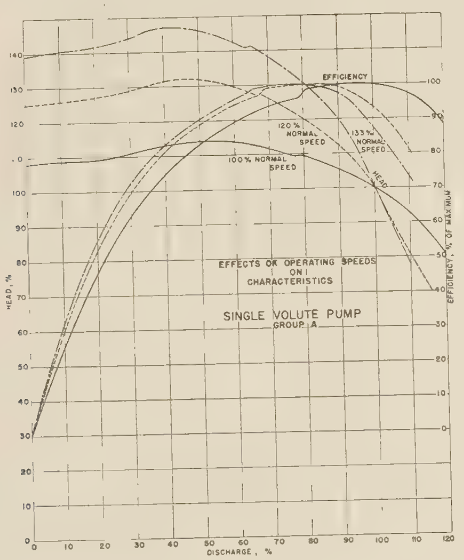

Fig. 3 Effects of Operating Speeds on Characteristics; Single-Volute Pump

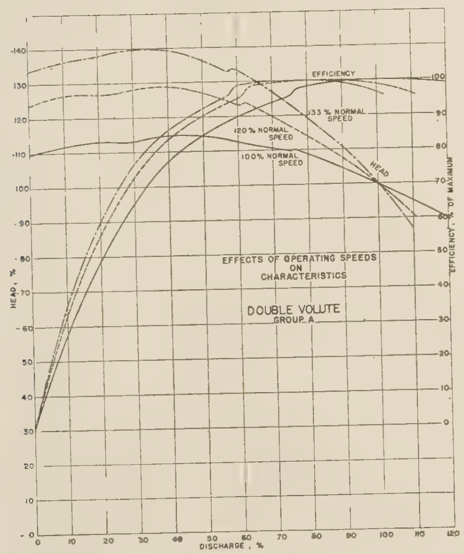

Fig. 4 Effects of Operating Speeds on Characteristics; Double-Volute Pump

type and fixed-vane-diffusor case, i.e., the steepness is obtained by sacrificing efficiency. The double-volute case again shows a surprisingly wide range of high-efficiency operation but, in this series, the diffusor case nearly duplicates its performance. However, the sharp drop in efficiency for the low-capacity region is again observed to be a diffusor-case characteristic.

Choice of Operating Speed. A given pump is designed to operate at a definite specific speed. In general, test results show that the unit has its maximum efficiency at this condition. However, if the performance characteristics show a reasonably broad zone of high efficiency, it may be possible to secure a better agreement between the pump characteristics and the field requirements if a different operating speed is chosen. The effect of the choice of operating speed may be observed in Figs. 3, 4, and 5. Fig. 3 shows the performance of the single-volute unit of group A operating at speeds of 100, 120, and 133 per cent of the design value. Fig. 4 presents the corresponding performance of the double-volute case, and Fig. 5 that of the fixed-vane pump.

All three units show the same trend, i.e., a marked steepening of the head-capacity characteristics with increase in operating speed. A closer examination of the three sets of curves shows that there are apparently two causes for this increase in steepness, (*a*) an increase due to the normal increase in the steepness

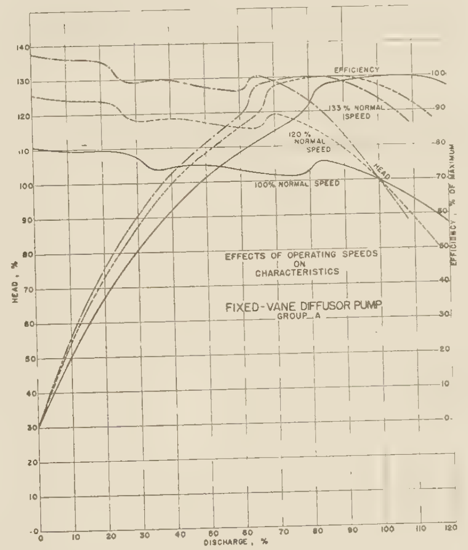

Fig. 5 Effects of Operating Speeds on Characteristics; Fixed-Vane-Diffusor Pump

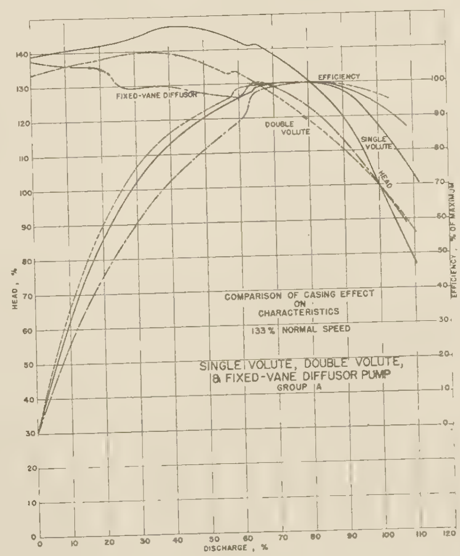

Fig. 6 Comparison of Casing Effect on Pump Characteristics; Group A, 133 Per Cent Normal Speed

of the impeller characteristic, as the capacity is increased, and (*b*) an increase in steepness due to a decrease in the efficiency of the case. Fig. 4 illustrates the effect of the former. It will be noted that, in the region of from 80 to 100 per cent discharge, the efficiency is high for all speeds, in fact, the lowest value is 96.5 per cent of the maximum. Thus, the change in steepness for this machine must be due largely to the shape of the impeller characteristics. The fixed-vane pump, Fig. 5, shows a larger variation in steepness, but the efficiency drops to 92 per cent in the same capacity range. Likewise, the single-volute pump, Fig. 3, shows an even greater variation in steepness, but the efficiency goes down to about 87 per cent of the maximum.

The difference between these three cases, when operated at the higher speed, is shown very clearly in Fig. 6. Here, the performance characteristics for the same three units, presented in Fig. 1, are plotted for a speed of 133 per cent of normal. At the design speed of Fig. 1, the head-capacity characteristic of each of the three cases shows about the same slope in the vicinity of the operating point. At the 33 per cent overspeed, however, the difference in steepness is quite marked.

From these comparisons, it would seem that, in view of the factors so far considered, the steepness of the head-capacity characteristics can be varied appreciably by choosing the speed at which the pump is to operate. If the choice is limited to speeds within the high-efficiency range, slight loss accompanies the variation. The double-volute case offers the widest possibilities within these limits because of its broad zone of high-efficiency performance. If steeper characteristics than those corresponding to the basic impeller performance are desired, they can be obtained only through sacrifice of efficiency. It should be remembered, however, that in this investigation no attempt has been made to explore fully the possibility of varying the impeller characteristics themselves.

Minor Operating Features

Hydraulic Balance and Radial Thrust. In the section, "Comparison of Normal Operating Characteristics," it was stated that the case can affect the impeller performance only by varying the pressure distribution around the periphery of the impeller and thus producing nonuniform discharge. Since this is an important feature for pump operation, it was thought desirable to make some experimental determinations of the pressure variation in the volute for the different types of cases. Consequently, piezometer connections were installed in the various cases—they were at constant radius. The piezometers for each case were spaced around a circle the diameter of which was slightly greater than the impeller and they covered the full 360 deg. Thus, the readings from them give a good picture of the pressure distribution around the impeller discharge.

Figs. 7, 8, and 9 show these measurements for the three cases of group B. The ordinates of all three curves are the static pressure at the piezometer connections, expressed in a percentage of the normal head produced by the pump. If the measurements for the single-volute pump, Fig. 7, are studied, it will be seen that the pressure distribution is reasonably uniform in the vicinity of the normal capacity; in fact, the most uniform distributions of those shown seem to be for the 93 per cent capacity. A glance at Fig. 1, shows that this is about the point of maximum efficiency. For higher and lower capacities, the pressure distribution is far from uniform and must affect the impeller discharge appreciably.

Fig. 8 shows that, for the double-volute pump, conditions are quite similar except that, of course, there are two pressure cycles in the 360 deg of the case. It will be noted here, however, that the range of pressure variation is considerably lower than in the corresponding single-volute case, although the basic design factors are similar.

Fig. 9 shows that the fixed-vane-diffusor pump has an even lower range of pressure variation. It should be remembered that these pressures are taken at a diameter corresponding to that of the impeller, i.e., at the inner side of the guide vanes. It will be seen that the pressure distribution is still nonsymmetrical. This is presumably due to the effect of the single volute on the outside of the guide vanes proper.

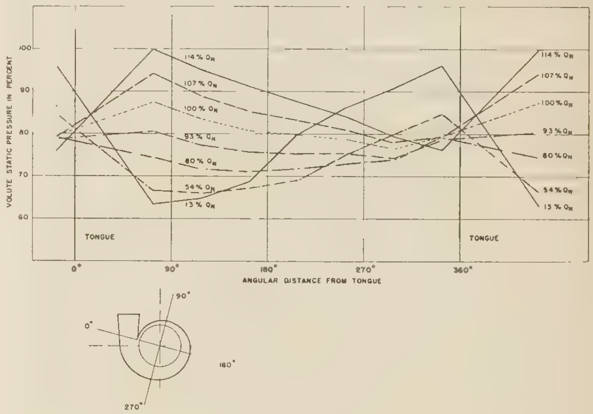

Fig. 7 Static-Pressure Distribution; Single-Volute Pump, Group B

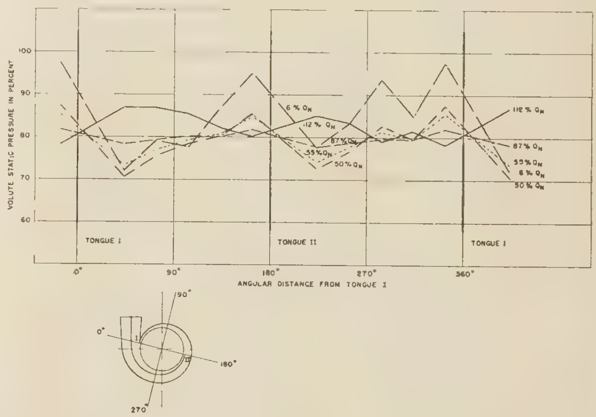

Fig. 8 Static-Pressure Distribution; Double-Volute Pump, Group B

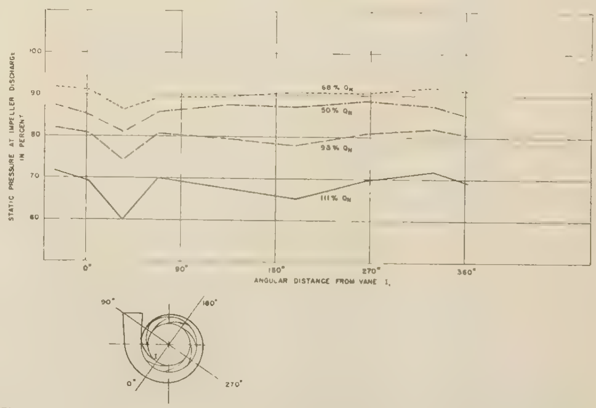

Fig. 9 Static-Pressure Distribution; Fixed-Vane-Diffusor Pump, Group B

Although this pressure variation must have a very marked effect upon the hydraulic performance of the unit, from an operating point of view, there is an even more direct result. A nonuniform pressure distribution such as, for example, the one shown in Fig. 7, for 114 per cent Q_n, indicates that there is a resultant radial thrust upon the impeller. This force must be taken care of in the mechanical design of bearings, case, and shaft, and may well be the controlling factor in the choice of shaft diameter and other important details. Failure to recognize this factor may result in mechanical contact of the wearing-ring surfaces and rapid deterioration of the equipment.

If the pressure-distribution diagrams for the double-volute pump, shown in Fig. 8, are integrated over the 360 deg, it will be found that the resultant radial force is small, since the effect of each of the two volutes nearly cancels the other. This is apparently true for all capacities and represents a distinct advantage of this type of construction. The resultant radial thrust upon the impeller of the diffusor pump, Fig. 9, is much lower than for the single-volute, but is somewhat higher than that of the double-volute. However, it should present no serious design problem, since it is not large.

It should be noted again that the radial unbalance of the fixed-vane-diffusor pump is due to the same cause that produced it in the single-volute pumps, i.e., the presence of the single volute itself. The main reason that the variation in pressure distribution and the resultant thrust are so much lower with the diffusor pump is that the flow is discharged into the volute at a much lower velocity than it is in the case of a single volute. If a fixed-vane case were designed, in which the vanes were used only as stay bolts and not as diffusors, high resultant radial forces should be expected. The importance of the investigation of these radial forces is illustrated by the fact that, for a good single-volute prototype, the unbalanced thrust is of the order of 50 tons. This would make illusory the feature of bearing-load elimination, commonly attributed to the vertical design.

Instability. Figs. 1 and 2 show that there are discontinuities in the head-capacity curves for all six cases. Such discontinuities appear to be characteristic of centrifugal-pump performance and are practically always found whenever tests of sufficient accuracy and detail are made. These discontinuities apparently are the result of a change in the flow from one regimen to another. For different design conditions, it seems that this change in flow can be localized either in the impeller or in the case. In addition, if the change is large enough in the impeller, it may also produce a significant change in the flow in the case. These flow discontinuities produce unstable ranges in the pump performance and, therefore, good practice indicates that the operating zone should not approach them too closely. For example, in the present study, one criterion tentatively proposed is that the maximum operating head should be at least 10 ft (3.5 per cent) below the break in the curve, as it is approached from the high-capacity side. This appears to be a quite satisfactory margin of safety for units having a reasonably small change in head at the discontinuity point, but may be somewhat inadequate for pumps having discontinuities as large as that shown by the fixed-vane diffusor of group A. For such pumps, it would seem advisable to restrict the maximum operating head to 1 or 2 per cent lower than the lowest value at the discontinuity region.

It is interesting to consider that significant information can be obtained by comparing the discontinuity regions, as shown by the capacity-head curves, with the torque or horsepower curves for the same conditions. If the flow regimen changes within the impeller passages, there will be a corresponding difference in the amount of angular momentum imparted to the fluid and this, in turn, will be apparent on the torque and horsepower curves. Thus, it may be concluded that, if a discontinuity in the head-capacity curve is reflected in the torque curve, the change in flow at least originates in the impeller. Conversely, if a discontinuity in the head-capacity curve is not accompanied by a similar break in the torque or horsepower curves, the change in the flow probably is localized in the casing. Unfortunately, space does not permit the plotting of the torque curves in Figs. 1 to 6, inclusive.

Cavitation Limits

Basic Limit of Eye Design. For each given design of an impeller eye, there is a relationship between capacity and inlet head which defines the beginning of cavitation. This basic limit, of course, assumes that, for all capacities, the flow has a normal velocity profile at the pump inlet; that the flow into the eye is circumferentially uniform; and that there are no tangential-velocity components present before the eye is entered. The difference between the basic characteristics of various eye designs for the same specific speeds will depend upon the abilities of the designers to keep their static pressures up and to eliminate local high-velocity regions in the vicinity of the passage entrances.

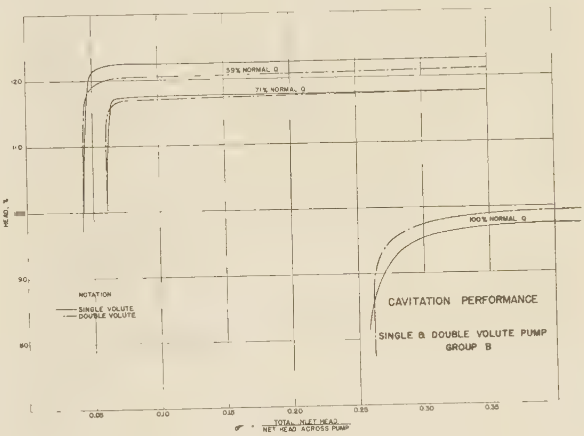

Fig. 10 Cavitation Performance; Single- and Double-Volute Pump, Group B

For a given impeller, however, this basic eye characteristic can be considered as the ideal limit for good performance. In actual operation, it can be modified either by the entrance conditions in the inlet piping approaching the pump, or by the reaction of the case on the inlet flow. The effect of the inlet piping is, of course, an installation problem, and will not be considered here, but the effect of the case is a question of basic pump design.

Effect of Case on Basic Limits. The effect of the case on the impeller characteristics has been discussed previously in the sections, "Comparison of Normal Operating Characteristics" and "Hydraulic Balance." It was seen that, in both high- and low-capacity regions, the case could produce a nonuniform pressure distribution around the impeller discharge. This must result in a pulsating flow in the impeller passages. Cavitation performance under these conditions must differ from that of steady flow. Previous studies at the laboratory (2) have shown that, under some conditions such as quite low capacity, the pressure unbalance on the impeller may be great enough to cause backflow from the case to the eye. Recent investigations also indicate that, in the same low-capacity region, the inlet tips of the impeller vanes may induce a radial-pressure difference sufficient to distort the flow further. It is difficult to separate these two phenomena, but together they seem to explain the "prerotation" which has been observed at times in pump inlets.

In an attempt to ascertain the effect of the various cases on

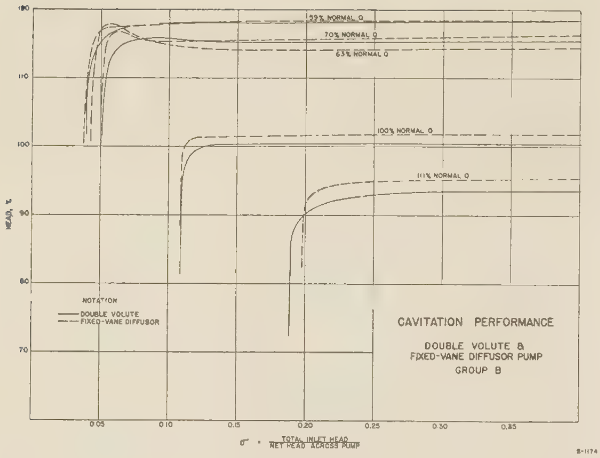

Fig. 11 Cavitation Performance; Double-Volute Case and Fixed-Vane-Diffusor Pump, Group B

the cavitation characteristics of the unit, Figs. 10 and 11 have been prepared. In both figures, the cavitation parameter σ has been plotted against the pump head for a series of constant capacities during which the inlet head was continuously lowered until cavitation was fully developed. Fig. 10 shows the comparative performance of the single- and double-volute cases of group B. It will be noted that the differences are slight, so slight in fact that little significance can be placed upon them. It is unfortunate that no runs are available at very low capacities, since this is the region in which the pressure distribution around the impeller differs widely for the two cases. Fig. 11 compares the double-volute case and the fixed-vane diffusor. These units are also from group B, but the results are not directly comparable to those of Fig. 10, because slightly different impellers were used in the two series of tests. Here, it will be noted that for one capacity the fixed-vane diffusor has a cavitation performance quite different from that shown by all other curves. The head rises rapidly, as σ decreases from 0.12 to 0.06. Since no such behavior is observed for either the single- or double-volute cases, it must be assumed that the fixed-vane-diffusor case is responsible for the difference.

The following logical explanation has been suggested by D. P. Barnes of the Bureau of Reclamation. The capacity at which this deviate behavior occurs is in the region for which the diffusor-vane angles must differ from the calculated discharge angle of the impeller. If cavitation starts in the impeller, it may quite possibly produce a change in the angle at which the flow leaves the impeller. If this angle more nearly coincides with that of the diffusor vanes, then the diffusion should be more effective and, therefore, the pump head should rise. Thus, it is possible that this rising head line on the σ diagram may be an indication of the beginning of cavitation, and hence marks a poorer rather than a better pump performance.

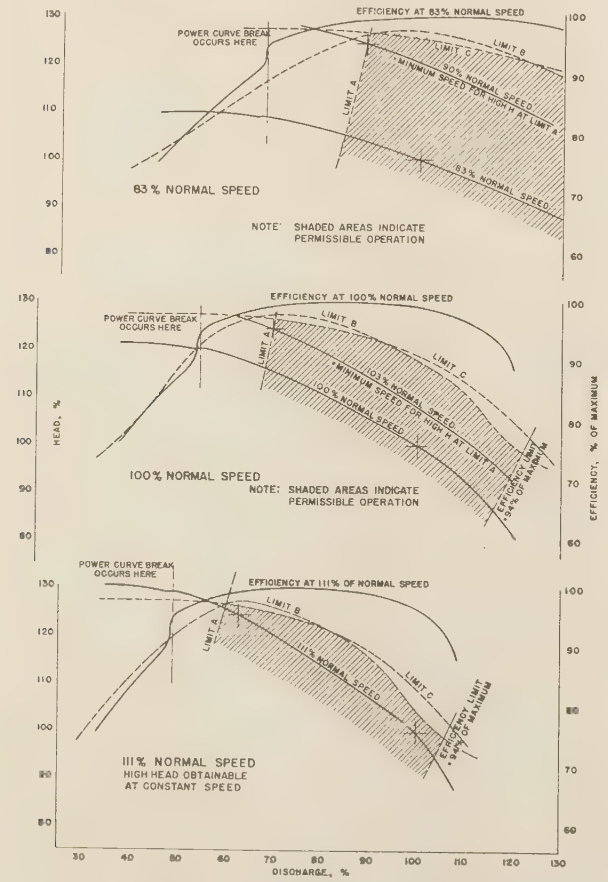

Fig. 12 Diagram Showing Limitations Upon Allowable Operating Regions of Characteristics of Double-Volute Pump, Group B

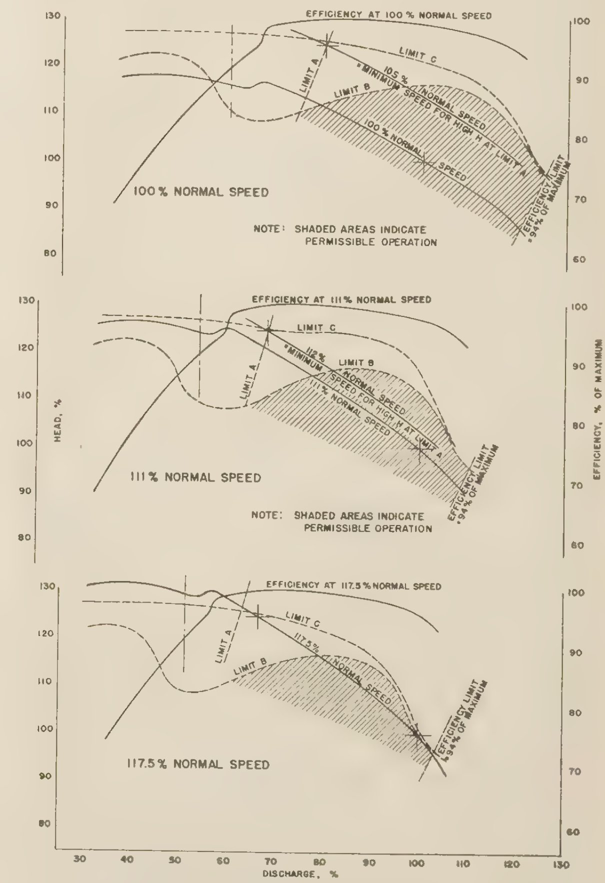

Fig. 13 Diagram Showing Limitations Upon Allowable Operating Regions of Characteristics of Fixed-Vane-Diffusor Pump, Group B

Selection of Operating Region. In an actual pump installation, the physical requirements impose many limitations upon the allowable operating regions of the pump characteristics. For example, the avoidance of discontinuity points has already been discussed. Freedom from cavitation is likewise necessary and this, in turn, is affected by the variation of inlet pressure due to change in reservoir level, etc. Fig. 12 presents a graphical diagram of these limitations as applied to the double-volute case of group B for three possible operating speeds. The limitations imposed are those obtained from a preliminary study of the Grand Coulee conditions. Limit *A* locates the permissible approach to the discontinuity or instability region. Limit *B* bounds the region for freedom from cavitation, as determined by the point at which there is a 0.5 per cent head drop on the σ curve (Fig. 11). Limit *C* bounds another cavitation parameter which is somewhat more complicated but which may be more satisfactory for certain units. The maximum and minimum operating heads are shown by the large crosses. The high-capacity boundary of the zone of permissible operation is arbitrarily defined by the condition that the efficiency has dropped to 94 per cent of the maximum value. The zone of permissible operation is indicated by the cross-hatched area and, within this zone, all of the criteria are met. This very useful type of presentation has been developed by D. P. Barnes.

It will be noted that, when the unit is operated at a speed of 83 per cent of the design value, only the low-head high-capacity portion of the required operating region can be covered. To obtain the high head required, an increase in speed to 90 per cent of the design value is necessary but, if this speed variation is permissible, the entire operating region can be covered satisfactorily. Conditions at the design speed are somewhat similar except that, to meet the high head condition, a speed increase to only something over 103 per cent is required. Operation at a speed of 111 per cent, however, permits the entire operating range to be obtained within the zone of permissible operation at constant speed.

Fig. 13 shows a similar diagram for the fixed-vane-diffusor pump of group B. Here, however, it is seen that, over a range of from 100 to 118 per cent of design speed, it is impossible to find any combination of constant- or variable-speed operation which will cover the desired range and yet meet the limitations imposed. It will be noted that in this unit the most serious deviation from limitations is from the "limit *B*" cavitation parameter.

Turbine Operation for Steady and Transient Conditions

One of the characteristic features of a pump installation is that transient conditions are quite commonly encountered under which the pump is called upon to operate as a turbine. Thus, for example, if the pump is operating normally and power should fail, unless there is a check valve in the line, the unit will slow down, reverse, and come up to runaway speed as a turbine, thus passing through the region of pump operation, a region of complete energy dissipation, and through the entire zone of turbine operation. In the design of large pump installations it is, therefore, very important for the plant designer to know the characteristics of the machines over the entire range of operating possibilities, in order that adequate provision may be made for maximum shaft torques, pressure surges, centrifugal forces, etc.

Complete Characteristic Diagrams. One of the first investigations of this complete range of pump operation was made by Kittredge and Thoma (3). It is convenient to present this information on a single diagram (4). Figs. 14 and 15 are two such diagrams for the single- and double-volute pumps, respectively, of group B. It will be noted that families of constant-head, constant-torque, and constant-efficiency lines are plotted against coordinates of capacity and speed. The performance of the unit at any constant speed is given by the intersection of these families of contours with a vertical line passing through the speed chosen.

Turbine Runaway Speed. The runaway speed of the unit, when operating as a turbine, is given by the intersection of the zero-torque line in the turbine region with the head curve corresponding to the pressure across the pump for that particular condition. For short pipe lines of ample proportions, this head is nearly the same as the pumping head since, under these conditions, the friction losses would be quite small. If the runaway speed exceeds the operating speed by a sufficient margin, it may be the controlling factor in the structural design of the impeller. Since the absolute value of this runaway speed is constant for a given unit operating under a given head, its value relative to the operating speed is determined by the choice of the latter. This can easily be seen by referring to Fig. 14. Consider that the normal operating head is represented by the 100 per cent head line. For the Grand Coulee installation, the maximum possible head which can cause turbine operation is about 120 per cent. The 120 per cent head line intersects the zero-torque line in the turbine zone at a negative speed of about 135 per cent. With a runaway speed of 35 per cent above that of normal operation, the impeller stresses may become quite serious. However, if it were decided that more suitable characteristics could be obtained by operating as a pump at 120 per cent of the design speed, then the runaway speed would exceed that of normal operation by about 12 per cent.

Turbine Operation for Possible Peak-Load Power Development. The Grand Coulee pumping plant of course is only a part of the total Grand Coulee project. A major function of the latter is power development. One of the problems always confronting a power project is the provision of sufficient capacity to meet peak-load demands. Therefore, the possibility has been suggested of using the pumping plant as a peak-load power supply by allowing the water to flow back from the upper reservoir, thus operating the pumps as turbines and the synchronous motors as generators. It will be noted in both Figs. 14 and 15 that these units have zones of very high efficiency in the turbine region, practically identical with the maximum efficiency obtained as pumps. Since the power must be supplied at constant frequency, it is necessary that the speed of turbine operation be the same as that of the pump. It is, of course, desirable to get as much power as possible from the turbines. However, the zone of turbine operation is determined by the selection of the pump operating speed.

For example, if in Fig. 15, the pump is considered to operate at 100 per cent speed, the torque and therefore the horsepower available in the turbine region will be 75 per cent of the corresponding values for the pump. For the high-head condition, i.e., for 120 per cent head, the turbine output will go up to about 110 per cent of the normal pump input at 100 per cent head. If, however, a normal operating speed of 111 per cent is selected for the pump, as was shown to be desirable in Fig. 12, conditions are quite different. Now, it will be observed that the normal torque input to the pump is 130 per cent for the low-head condition and about 120 per cent for the high-head condition, whereas, the corresponding turbine operation shows a torque of only about 35 per cent for the low-head condition and about 85 per cent for maximum-head. These values must be corrected to the new reference of 130 per cent, which was the input torque to the pump under normal head conditions. On this basis, the turbine output varies from 27 to 60 per cent of the power input to the pump at normal operating head. This output would appear to be so small as to be of doubtful value for a peak-load power supply. The trend, indicated by these examples, appears to be general, i.e., for a given design, if the operating point as a pump is located at a relatively low capacity, the operating speed will be low, the

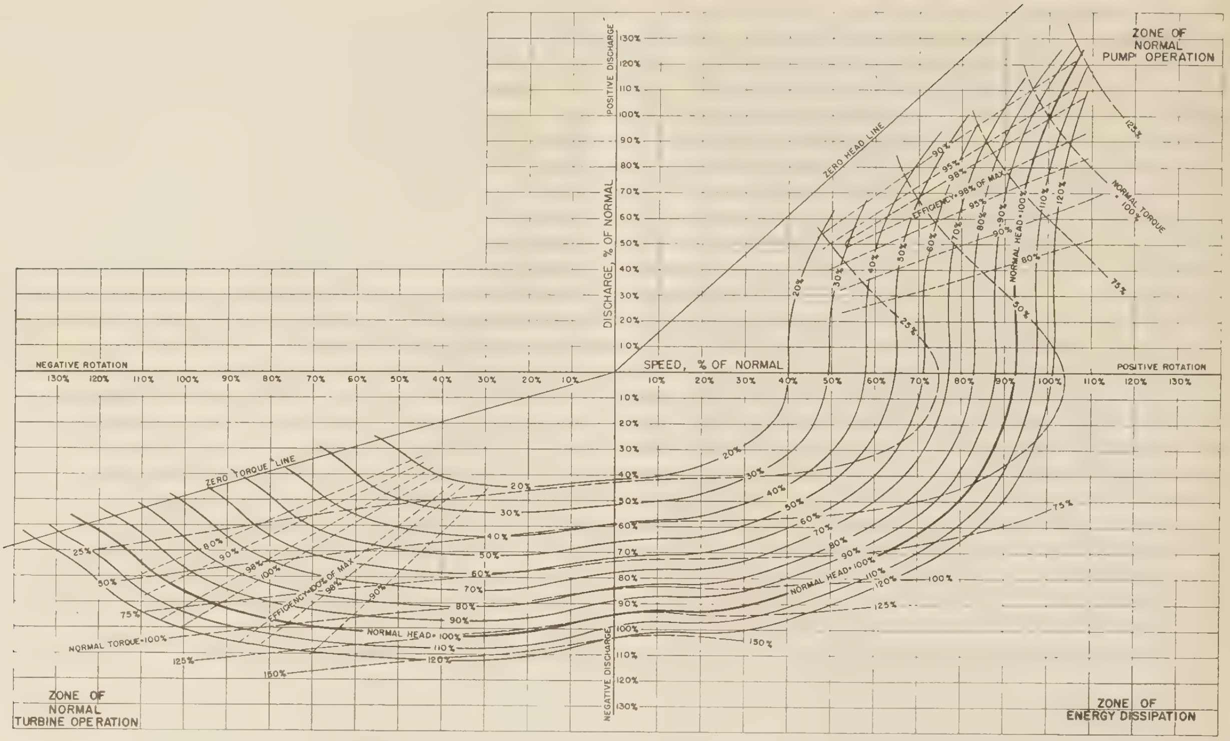

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Fig. 14 Complete Characteristic Diagram, Single-Volute Pump, Group B

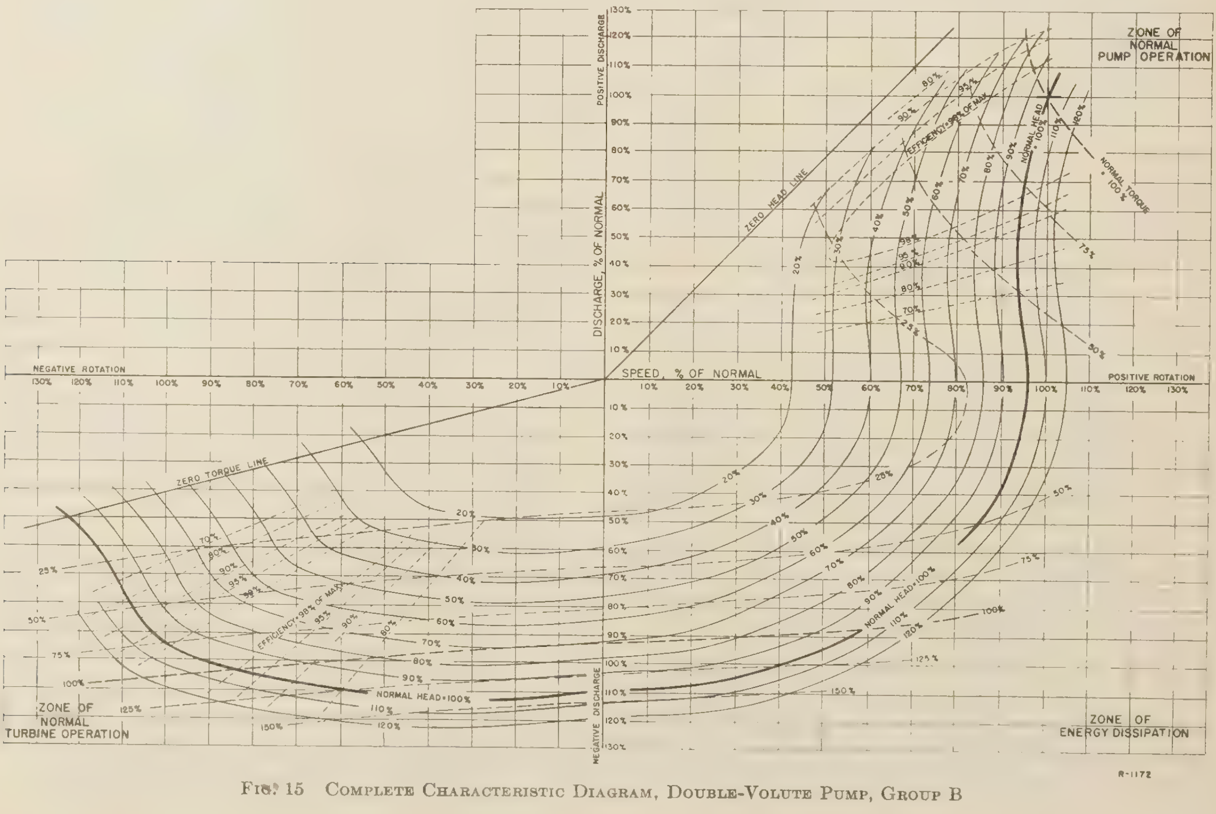

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Fig. 15 Complete Characteristic Diagram, Double-Volute Pump, Group B

turbine capacity will be high and the runaway speed will be high; whereas, if the operating point is chosen at a relatively high capacity and speed, the turbine capacity and the runaway speed will both be comparatively low. Thus, one more factor is added to the complicated set of requirements involved in the choice of the proper unit for the given installation.

Transient Behavior. The transient behavior of a pump is a function not only of the pump characteristics, but also of the pipe-line characteristics and other hydraulic and inertia features of the entire installation. The prediction of transient behavior has been briefly discussed in one of the previous references (4). Figs. 16 and 17 show typical transient characteristics for the double-volute pump of group B. These were computed by the

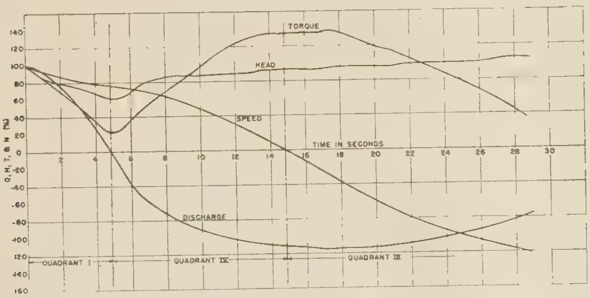

Fig. 16 Transient Characteristics of Double-Volute Pump, Group B

(Calculation for power failure when operating at 100 per cent speed and normal-head conditions.)

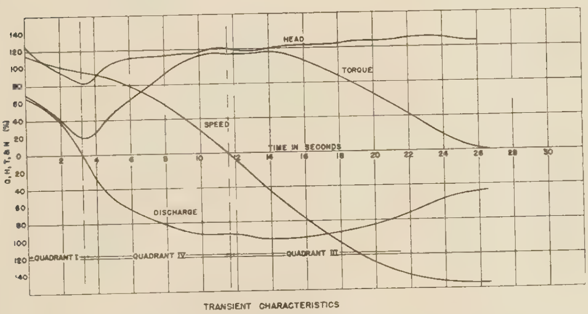

Fig. 17 Transient Characteristics of Double-Volute Pump, Group B

(Calculation for power failure when operating at 117.5 per cent speed and extreme high-head conditions.)

use of the Bergeron graphical method of water-hammer calculation (5) with the data from the laboratory for the complete pump characteristics.

Fig. 16 shows the performance following power failure when the pump has been operating at normal head and speed. Fig. 17 gives the corresponding characteristics for the extreme high-head condition, with the pump operating at 118 per cent of design speed at the time of power failure. For both conditions, it will be noted that the head fluctuations are quite moderate and present no problem. On the other hand, it is somewhat startling to imagine a 65,000-hp unit changing from a normal pump, operating at full speed in one direction, to a turbine operating at runaway speed in the other direction in an elapsed time of only 26 to 28 sec. The torque curves show that the maximum shaft stresses increase to 40 per cent above the normal operating value. The runaway speeds correspond closely to those already discussed.

Fig. 16 indicates that the unit operation remains in quadrant IV for as much as 10 sec. This is a region of complete energy dissipation, since energy is being poured into the machine through the deceleration of the rotating mass while, at the same time, energy is being given up in the machine by the fluid flowing through it. Little is known about cavitation conditions in this region, aside from the fact that they are apparently quite serious. It is felt that quadrant IV operation offers a fruitful field for further investigation.

Summary of Results

Limitation of Program. Before summarizing the results, it should be re-emphasized that, although this investigation has shed some light on a few of the factors involved in the selection of the type and design of pump to meet particular needs of a given installation, the amount of information is still very meager. Many possibilities of casing design remain to be explored. Cavitation limits are yet too empirical in character, and the possibilities of obtaining more desirable performance for a given installation through changes in the impeller design are barely touched.

Operating Characteristics and Speed. The over-all performance of a pump, using a given impeller, is greatly affected by the case design. For a given type of case, the characteristics may be varied considerably by the choice of the point at which the case "fits" the impeller. Of the three types of cases studied, the double-volute type appears to give the widest high-efficiency range.

A well-designed impeller has a fairly wide range of speeds over which it will operate satisfactorily when delivering against a given head. A proper choice of case "fit" therefore will result in a unit having the desired operating speed. For a given combination of impeller and case, the head-capacity characteristics can be "steepened" by choosing the operating point at a relatively high capacity and speed. If a head-capacity steepness greater than that of the basic impeller performance is desired, it can be obtained only by the sacrifice of efficiency, i.e., by pushing the operating point to a capacity out beyond the zone of maximum efficiency. This is equivalent to using a casing too small for the desired capacity.

Hydraulic Balance and Radial Thrust. Within the zone of maximum efficiency, the fit of the case to the impeller is usually satisfactory enough to produce a relatively uniform pressure distribution around the impeller discharger. Therefore, operation in this zone is accompanied by little or no radial thrust. Operation at higher or lower capacities distorts this uniformity and results in radial thrust. The resultant force on the impeller and shaft is highest for the single-volute case. The fixed-vane-diffusor construction greatly reduces the magnitude of the force and it is eliminated by a well-designed double-volute casing.

Instability. Discontinuities in the head-capacity characteristic seem to be an inherent feature of centrifugal pumps, or at least of high-efficiency ones. These discontinuities probably are due to changes in the flow regimen, either in the impeller or case. They often limit the extent of the satisfactory operating range. The closeness with which they may be approached is presumably a function of the magnitude of the discontinuity.

Cavitation. Cavitation is an impeller phenomenon and is relatively insensitive to casing design. However, severe unbalance of the pressure distribution around the impeller discharge may change the cavitation conditions. Cavitation usually produces a change in the head-capacity characteristic. In general, the head is lowered, but under some circumstances it seems that it may be first increased. Cavitation forms one of the major limitations in determining the zone of satisfactory operation. If, in order to obtain other desirable characteristics, the operation point for a given impeller is chosen some distance away from the design point, it may be necessary to modify the eye design to

secure satisfactory cavitation elimination. As yet, no satisfactory quantitative determination of the inception or degree of cavitation has been developed.

Turbine Operation. In general, a centrifugal pump can be operated very satisfactorily as a turbine and, over a limited range, with an efficiency equal to the best performance as a pump. In special cases, it may be feasible to utilize this possibility to supply a peak-load power demand by reversing the flow and operating the pump as a turbine and the motor as a generator. If this is to be done, careful consideration must be given to the design of the unit, since the selection of the pump operating point determines the turbine performance as well. The conditions for securing the optimum pump characteristics, turbine operation, and low runaway speed are usually not compatible, and therefore the relative value or the different elements of the performance must be evaluated carefully.

Conclusion

Although this study was designed to answer specific questions covering the selection of operating features for the pumping units to be installed at the Grand Coulee project, the results obtained are somewhat more generally applicable than might be expected. It is anticipated that, in the future, there will be more and more demand for hydraulic units, both pump and turbine, the characteristics of which are particularly adapted to the installation requirements, and it is felt that studies of the kind herein reported offer the most reliable means of securing the desired result.

Acknowledgments

During the period of the investigations, the Bureau of Reclamation, through the chief engineer and members of his technical staff, has kept in intimate contact with the work and has contributed much to its progress. Especial acknowledgment is due to Mr. D. P. Barnes, resident representative, of the Bureau, who has taken an active part in the experimental investigations and their analyses.

The program has been carried out under the immediate direction of Prof. Th. von Kármán, Prof. R. L. Daugherty, and the author. The technical staff has been in charge of Mr. J. W. Daily. The results reported are the joint product of the entire staff and should be so considered.

BIBLIOGRAPHY

1 "The Hydraulic Machinery Laboratory at the California Institute of Technology," by R. T. Knapp, Trans. A.S.M.E., vol. 58, 1936, pp. 663–676.

2 "Experimental Determinations of the Flow Characteristics in the Volutes of Centrifugal Pumps," by R. C. Binder and R. T. Knapp, Trans. A.S.M.E., vol. 58, 1936, pp. 649–661.

3 "Centrifugal Pumps Operated Under Abnormal Conditions," by C. P. Kittredge and D. Thoma, *Power*, vol. 73, 1931, pp. 881–884.

4 "Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behavior," by R. T. Knapp, Trans. A.S.M.E., vol. 59, 1937, pp. 683–689.

5 "Methode Graphique Générale de Calcul des Propagations d'Ondes Planes," by L. Bergeron, Mémoires de la Société de Ingénieurs Civils de France, Bulletin de Juillet, vol. 90, 1937, pp. 407–497.

Development of a Major Principle in Pulverized-Coal Firing

By FRED L. DORNBROOK,[1] MILWAUKEE, WIS.

Positive avoidance of the plastic-ash phase at all surfaces bounding the furnace has required emphasis upon heat-absorption performance of furnace surfaces. Evolution of the present-day furnace, which is practically self-cleaning, reliable, of high efficiency, and can operate continuously month after month under variable conditions of coal, load, and attendance, is treated in the paper from its inception in Milwaukee during 1918 to the present new installations of the author's company.

THE following 20-year-old conclusions, concerning the "slagging" problem and furnace heat absorption, constitute a basic principle in pulverized-coal firing, and also furnish the theme for this paper:

"If the refuse is to be removed easily, the bottom of the furnace must be kept below the temperature at which ash becomes sticky."

"A better way of keeping the furnace temperature . . . . slightly below that of the running slag . . . . is to expose a large amount of boiler heating surface to radiation from the furnace."

These statements were made in a bulletin[2] which covered an investigation of powdered coal as fuel at the Oneida Street Station in Milwaukee (now called East Wells Street Station), where experimental work had been carried on for some months.

A major problem in burning pulverized coal concerns the incombustible portion of the coal, namely, the ash. If the ash could be removed from the coal before firing, an enormous simplification of the combustion process would occur. Almost every element in the boiler plant would be affected favorably. Thus a minor constituent of coal requires the major attention of the designer and operator.

The Furnace Temperature Scale

Fig. 2 was prepared to show graphically the general conclusion that the ash problem dictates that over 50 per cent of the total heat absorption must occur in many furnaces of modern design if ash-slagging problems are to be avoided.

In an all-brick furnace, having no heat-absorbing surface, a theoretical temperature of 3700 F would be reached. This assumes 16 per cent CO_2 (15 per cent excess air) and 600 F preheated air. This temperature is 1400 F above the 2300 F ash-softening temperature of average Pennsylvania coal.

The flue gas or products of combustion must be cooled from 3700 F to 800 F by the water and steam surfaces before entering the air heater. This is a total of 2900 F of cooling and, since the specific heat of the gases varies only slightly over the temperature range, the furnace must absorb $\frac{1400 \text{ F}}{2900 \text{ F}}$ or 48 per cent of the total net absorption to lower the furnace exit temperature below the ash-softening temperature. Actually over 50 per cent should be

Fig. 1 Historic Pulverized-Fuel Boilers

(Tests of boiler No. 5, in foreground, were subject of Bureau of Mines Bulletin.[2])

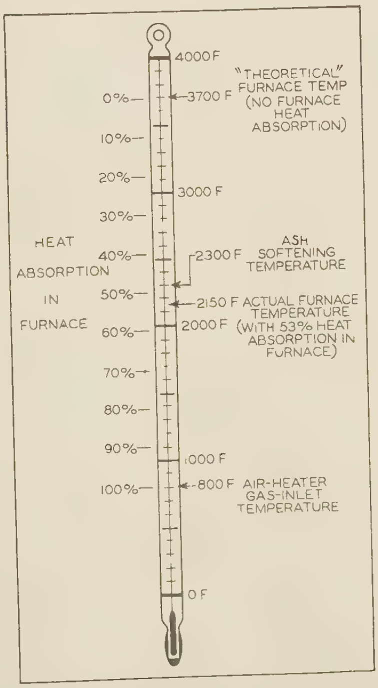

Fig. 2 Furnace-Temperature Scale

(Showing that over one half of heat absorption of modern boiler units must occur in furnace if ash-softening temperatures are not to be exceeded at boiler entrance.)

[1] Chief Engineer of Power Plants, Wisconsin Electric Power Company. Mem. A.S.M.E.

[2] "An Investigation of Powdered Coal as Fuel for Power-Plant Boilers," by H. Kreisinger, John Blizard, C. E. Augustine, and B. J. Cross, U. S. Bureau of Mines, Bulletin No. 223, 1923.

Contributed by the Power Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

TABLE 1 TRENDS OF FURNACE HEAT ABSORPTION IN MILWAUKEE BOILER UNITS

(Showing reduction of furnace exit temperatures, in spite of larger sizes, by greater furnace heat absorption)

	Oneida St.		Lakeside					
Item	Without water screen	With water screen	Boiler room No. 1	Boiler room No. 2	Boiler room No. 3, boiler No. 19	Port Washington, boiler No. 1	East Wells St., boiler No. 16	Commerce St., boiler No. 25
1 First operated, year	1918	1920	1920	1923	1928	1935	1938	1941 (scheduled)
2 Steam output (design max) lb per hr	18000	23000	80000	131200	300000	690000	225000	375000
3 Heat input, million Btu per hr	24.1	31.2	110.0	183.5	430	890	326	470
4 CO_2 at boiler outlet, per cent	14	15	13	15.5	14.5	14.25	14.8	15
5 Efficiency, over-all, per cent	79.7	78.8	87.5	88.8	85.7	86.0	85.3	86.1
Heat-absorbing surface, sq ft:								
6 Boiler, convection surface	4680	4680	13057	17650	23640	44087	13500	24000
7 Furnace, radiant surface[a]	101	171.0	618	1324	5338	9748	4609	6391
8 Economizer, convection surface	0	0	7603	14256	0	0	0	0
9 Air heater, convection surface	0	0	0	0	23640	121000	52300	66500
10 Furnace volume, cu ft	1608	1463	6450	11650	29780	61300	19951	30500
11 Btu released per hr per sq ft of furnace surface[a]	238000	182000	178000	138000	80500	91500	70500	73500
12 Btu released per hr per cu ft of furnace volume	15000	21300	17000	15700	17800	14500	16300	15400
13 Average ash-fushion temperature, F	2300	2300	2150	2150	2150	2250	2250	...
14 Average furnace temperature (calculated maximum output), F	2350	2350	2100	2100	2000	2100	1950	2000

[a] Includes projected surface of boiler tubes and superheater surfaces corrected to equivalent water surfaces.

provided to have some margin to cope with operating conditions and contingencies.

This example indicates that low excess air and high temperature of air preheating require greater heat absorption in the furnace, if ash problems are to be avoided. Air preheaters are being used in many plants in place of economizers since higher steam pressures are causing more extensive use of extracted steam to heat feedwater. The heat absorbed in the air heater will raise the furnace temperature unless the designer has provided for increasing heat absorption in the furnace.

Larger steaming capacities of boilers naturally have placed emphasis on the ash problem because the furnace heat-absorption area does not increase as rapidly as does the furnace volume when boiler outputs are increased. Doubling of furnace dimensions normally increases furnace volume 8 times but available water- or steam-wall area is increased only 4 times. Thus larger boiler units tend to have hotter furnaces unless the volumetric rate of combustion is decreased proportionately.

Milwaukee Experiences

Table 1 summarizes Milwaukee experience in this regard, starting with the small Oneida Street units, including the large Port Washington boiler unit, and ending with the Commerce Street installation now under construction. It shows:

1 The large increase in size of boiler units. Port Washington output is 30 times that of Oneida Street units. (See item 2 of Table 1.)

2 Retention of low furnace temperatures to a point below ash-softening temperature. (See item 14.)

3 Decrease of excess air, increase of air preheat, more furnace cooling, and uniformity of Btu per cu ft per hr heat release.

4 Decrease of Btu release per hr per sq ft of furnace heat-absorbing surface to about $^1/_3$ of early practice.

Comparison of items 13 and 14, Table 1, will show that, at maximum rated output, the furnace exit temperatures of the Lakeside units are about 50 F below ash-fusion temperatures. Since these units serve topping turbines and normally operate at 83 per cent of maximum rated output, it can be said that extensive experience with these four large units urges that furnace exit temperatures be limited to 200 F below ash-softening temperature, for normal operation.

Ever since Lakeside fuel changed in 1934 from Pennsylvania and West Virginia coal to southern Illinois coal, experience with this matter of maintaining a margin against slagging of boiler inlet tubes has been accumulated. Excess air required slight increases, cleaning methods were studied and, in general, operation had to be more carefully conducted in order to sustain load-carrying reliability. The more recent installations in subsequent stations have been provided with a greater margin before troublesome slagging can occur.

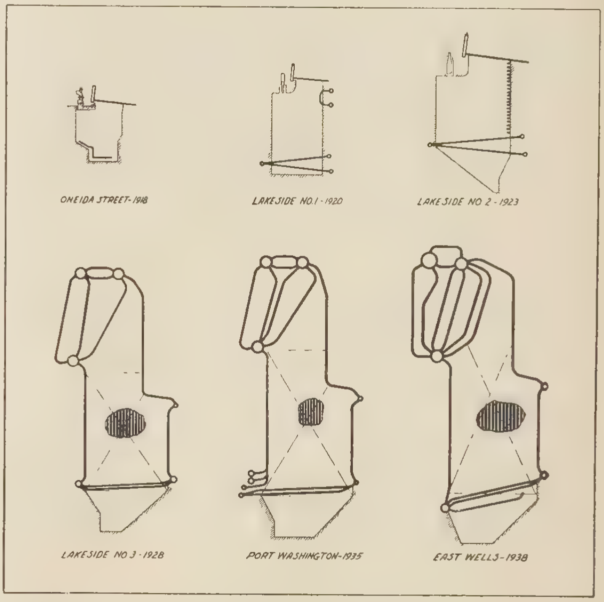

Fig. 3 Evolution of Furnace Cooling of Milwaukee Boiler Units

(Showing relative size, extent of brickwork, and "cold" surfaces [in black], and illustrating definite trend toward greater furnace heat absorption.)

The "furnace exit temperature" at Lakeside and in subsequent stations is actually about 200 F above the boiler inlet temperatures and is measured by high-velocity thermocouples. Calculations are made for this furnace exit temperature at the location where the gases enter the triangular portion above the arch level Fig. 3, in order to correlate the calculated heat-transfer rates with an extensive experience of radiant superheater transfer rates. When it is borne in mind that gases are reduced about 200 F in this triangular section, one realizes from Lakeside experience that boiler-inlet-gas temperatures 200 to 300 F below ash-softening temperatures afford scant margin against first-pass clogging.

History of Milwaukee Installations

The original experiments at Oneida Street were conducted on a 4680-sq ft water-tube boiler in 1918. The management of The

Milwaukee Electric Railway and Light Company (now Wisconsin Electric Power Company) encouraged the application of pulverized coal to this boiler. They considered it a good business risk to place the boiler at the disposal of the operating crew for experimental purposes feeling that, even under these conditions, the boiler would be available when needed. Almost endless experiments and tests were made before a furnace of the proper design was developed; in fact, one furnace was rebuilt five times before obtaining a satisfactory shape and volume.

The water screen played an important role in arriving at an acceptable furnace performance in this original pulverized-coal unit. To quote the Bureau of Mines Bulletin:[2] "The first 16 tests showed the impossibility of running the furnace with low excess air without the ash fusing at the bottom of the furnace. Consequently experiments were made with a water coil or screen to cool the bottom of the furnace. This coil, by partly screening the bottom from the radiation of the flame and by absorbing radiation from the bottom, kept the temperature of the bottom below that of the fusion point of the ash. Screen No. 5 in Fig. 13 was most effective in preventing troublesome slag at high rates of combustion and with low excess air."

Lakeside Station

At Lakeside, where the second application of dry-ash furnaces was made, the slag problem was largely overcome by the installation of so-called water screens in all furnaces. These screens were connected into the circulating system of the boiler by means of vertical risers. It was in this installation, too, that a radical departure from the convection type of superheater was attempted through the introduction of the then new radiant-heat-absorbing superheater. In 1923, John Anderson, in a technical report[3] on the subject stated: "In view of the keen interest manifested toward the furnace problem, the advent of this type of superheater on a commercial scale happens at an opportune time. It seems to be the consensus of opinion of many of the foremost power-plant engineers that the ultimate solution of the present furnace problems will be in steam- or water-cooled furnace walls. The installation of the water screen and the radiant-heat superheater seems to confirm the existence of a tendency toward steam- and water-cooled furnace walls."

It is natural that the developments in the first two of Lakeside's boiler rooms should become part of the design for its third boiler room. It is here that Lakeside's four 1300-lb boilers are located. Dry-ash furnaces with radiant superheating surfaces were again the order. Each of the four high-pressure boilers serves a topping turbine of 7700-kw capacity, all of which exhaust at constant pressure to the plant's 300-lb header after the steam passes through reheaters, located within the individual boilers.

The temperature of steam exhausting from the topping turbine increases with decreasing load, thereby requiring a reheater of convection characteristics to obtain a uniform outlet temperature over the operating load range. As a result, radiant reheating is not required. Radiant superheating surfaces, however, are used in the side walls of these furnaces with waterwall surfaces in front and rear.

Port Washington Station

In the Port Washington boiler, which is the largest on the Milwaukee system, the opportunity existed for using a radiant

Fig. 4 Cleanliness in Spite of Large Output
(With an output of 690,000 lb per hr, the Port Washington unit does not cause slagging troubles because of adequate heat-absorbing area. The condition of the furnace without cleaning, after a 6-months' continuous run at normal output of 465,000 lb per hr, is shown herewith.)

reheater surface advantageously. This resulted from a combination of uniform throttle steam temperature over a wide load range and compound-turbine operation. The upward steam-temperature trend of the radiant surface at lower loads fits well with the lowered outlet steam temperature from the high-pressure section of the turbine, thereby permitting the maintenance of a uniform reheater outlet temperature over the same wide load range. Incidentally, this inherent control of reheated-steam temperature has aided materially in maintaining high thermal efficiency in the plant month after month. The placing of the steam-cooled surface in the furnace wall has not only aided in retaining the desired low furnace temperature but has also resulted in a gain to the plant in thermal efficiency.

As steam pressures increase, reheating becomes more necessary to the economical operation of the steam cycle. This influence, together with the trend toward higher steam temperatures directs attention toward radiant steam surfaces, because they become increasingly more important as superheating and reheating mediums. The low-temperature furnace with its greater wall areas permits installation of radiant superheating and reheating surfaces and thus points the way toward further progress in the art of steam-power generation.

Fuel Versatility and Reliability

The ability to change from one coal to another to permit economies in the purchase of coal when price differentials change has long been a goal sought by boiler-plant owners. A versatile furnace which is not affected by a change in the constituents of coal, such as occurs when changing from a so-called high-grade coal to a low-grade coal, is of inestimable value. At Lakeside, for instance, the coal situation changed decidedly and rather suddenly during the depression. Eastern coal had been the most economical, but when midwestern operators began producing coal in large quantities and offering it at attractive prices, the economic picture changed. Tests in Lakeside's furnaces indicated

[3] "The Use of Pulverized Coal Under Central Station Boilers," by John Anderson. Paper presented before the Technical League of the Employes Mutual Benefit Association, The Milwaukee Electric Railway & Light Company, Milwaukee, Wis., February, 1920. Because of its historic value in the art of combustion, this paper has been reprinted by the Combustion Engineering Company, Inc., New York, N. Y.

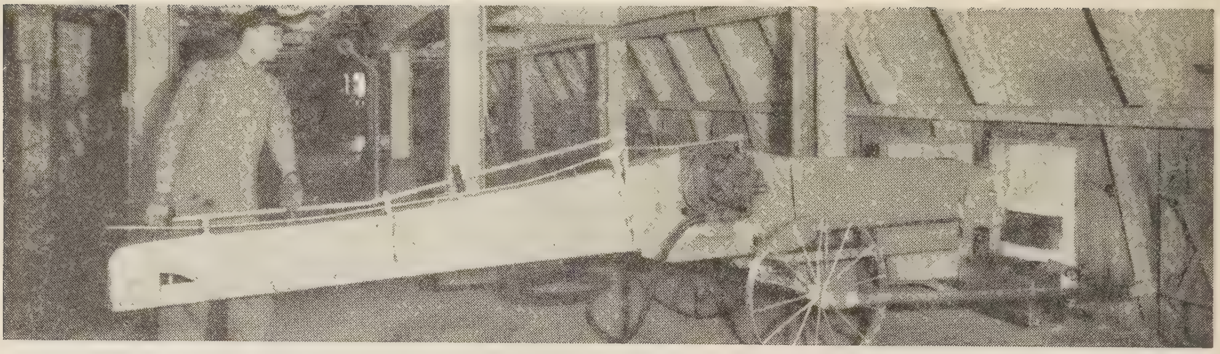

Fig. 5 Reliable Ash Removal

(A method that does not require any particular boiler-loading schedule nor reduce boiler-unit availability is shown. See Fig. 6 for outline of this mechanical hoe and furnace hopper bottom.)

that the new coal could be burned without difficulty and without undue added costs. On the basis of the tests, the change was made in 1934. Since that time, Lakeside has made appreciable annual savings, even after allowing for increased costs incurred by the poorer quality of the coal (ash disposal, etc.). This is tangible evidence of the value of a versatile furnace.

Reliability is an important factor in producing low over-all costs and its value cannot be overemphasized. The four high-pressure boilers at Lakeside have an availability record of 94.2 per cent over a 9-year period in spite of an average of 15 stops and starts per month. The Port Washington boiler has operated at an availability of 93.9 per cent over a 4.1-year period, Table 2.

TABLE 2 AVAILABILITIES OF FIVE 1300-LB BOILER UNITS FOR SEVERAL YEARS

(A total of 40 boiler years' experience at an average availability of 94 per cent attests to the point that moderate furnace temperatures are conducive to high reliability)

Year	Lakeside, average of 4 boilers, per cent	Port Washington, 1 boiler, per cent
1931	92.0	...
1932	93.8	...
1933	97.6	...
1934	94.0	...
1935	95.3	100[a]
1936	93.0	91.1
1937	95.0	93.0
1938	93.0	97.5
1939	93.9	93.4
Average (weighted)	94.2	93.9

[a] Started November 22, 1935.

These reliability records have been obtained on boilers having moderate furnace temperatures.

Efficiency Aspects

Consistently holding to optimum efficiency conditions for the many different operating situations experienced in typical plant routine is possible with the type of furnace under discussion. Change of coal, unusually high or low load, disability of a portion of auxiliaries which affect the furnace, or trouble with regulating superheat temperature, and several other situations that can cause expenditure of extra fuel with less versatile furnaces, affect the dry-ash furnace comparatively little. With favorable burners and proper air admission, combustion at low ratings is stable and efficient.

While it may be possible to operate a hotter furnace during many of the conditions mentioned without sacrifice of economy, provided special attention is given, it is usually necessary to compromise for the sake of reliability. When a boiler unit is needed definitely for daily peak loads, its operators will not take a chance of rendering it unavailable for the peak. A positive margin against clogging the first pass of boiler tubes, for instance,

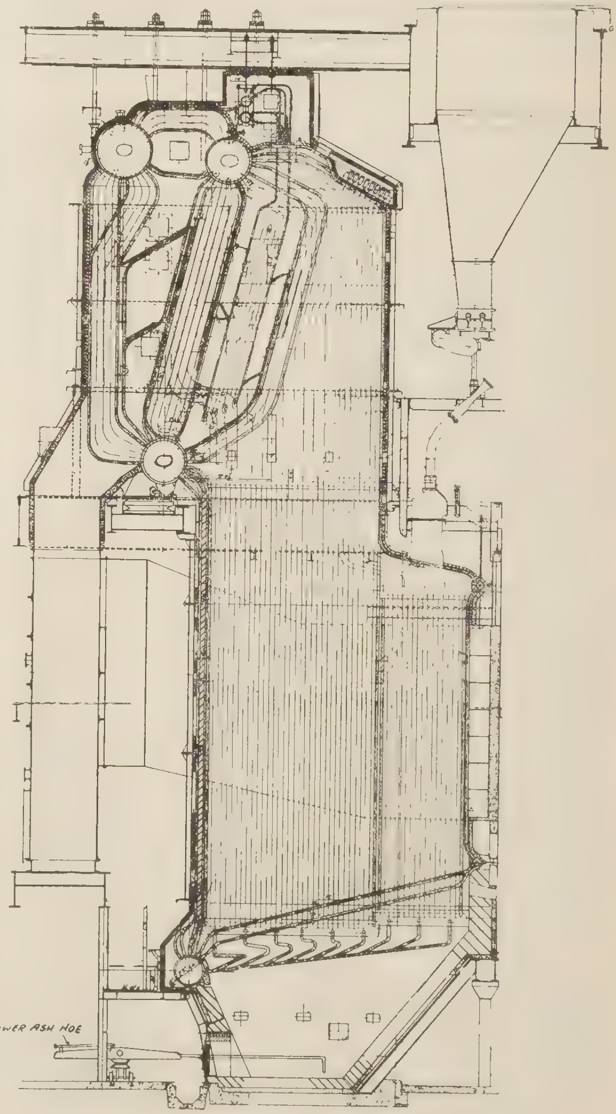

Fig. 6 A Recent Design

(This East Wells Street, 650-psi, 225,000-lb per hr boiler unit employs more liberal furnace cooling than any of its predecessors.)

must be provided continuously. Many boiler troubles are cumulative and, once they start, are difficult to arrest unless load can be reduced.

It is not what a boiler unit can do under special test, but what its operators actually get from it over a period of several years that is the criterion of boiler-unit efficiency. From available data, there appears substantiation of the premise that dry-ash furnaces are conducive to high sustained efficiency.

Long-Time Maintenance Aspects

It appears probable that furnace-maintenance costs increase in proportion to furnace temperatures, even when brickwork deterioration is disregarded. It has been found by experience that long-time service in furnace duty will cause maintenance which would not be anticipated even upon careful inspection after a year of operation.

When steel absorbs heat at the rate of 50,000 Btu per sq ft per hr, there is a 200-F temperature difference per in. of thickness perpendicular to the plane of absorption. This causes stresses of considerable magnitude. When this occurs, at least in the case of integral extended surfaces, minute cracks may start in a year or two, and then tenaciously travel without limit, even into the inside of the tube itself. Steel at high rates of heat transfer behaves like masonry at low temperatures; short-time service is no guarantee against long-time freedom from destructive cracking.

If several furnace tubes fail on different occasions from the same trouble, then all similarly affected tubes, even those deteriorated to a much lesser degree, must be replaced. Less drastic maintenance was possible in the days of many small boiler units serving one turbine, but present-day practice requires that boiler units be kept at peak reliability at all times. Cases are known where forced outage of one boiler caused prompt outage of other boilers operating in parallel because the assumption of the high load caused similar trouble. Operators, who have experienced the distress of troubles that become cumulative, thereafter try hard to avoid their cause.

Dry-Bottom Versus Dry-Ash Furnaces

There is a difference between a dry-bottom furnace and a dry-ash furnace. Dry-bottom furnaces have experienced serious falling of enormous accumulations of dense slag from upper parts of the furnace onto screen tubes or floor tubes. The dry-ash furnace permits of no destructive nor outage-provoking accumulations at any point, not even on brickwork. Attachment of ash is so fragile that the light honeycomb deposits fall when reaching a few pounds weight.

Superheat-Temperature Control

Variations in ash deposits will present a problem of steam-temperature control in boiler units of the future, where heat absorbed by superheating surfaces may exceed 50 per cent of the total heat absorbed by the entire unit. The dry-ash furnace is not entirely free from ash deposit but the variations in ash deposit will not cause a severe problem in control of superheat temperature.

Economics of Furnaces

Wet-ash furnaces are chosen frequently because of minimum installation costs. Space limitations often require the use of small furnaces; in fact, many have been installed between the same building columns used for older units of less output.

Boiler-unit reliability has a decided bearing upon total investment costs and operating costs. Savings due to furnace size are lost if the design lacks reliability and requires more spare boiler units. It is felt that too much emphasis cannot be placed upon reliability.

Conclusions

This paper shows how the present-day furnace in Milwaukee has been evolved from the original pulverized-fuel furnace in 1918, and traces the development of a major principle in pulverized-fuel firing.

In order to prevent ash from sticking to the inside surfaces of any furnace, these surfaces must be designed to cool properly the ash below its plastic state.

Experiences and data are presented indicating that a practical solution of the so-called "slagging problem" consists of designing boiler furnaces for such appreciable heat absorption that gases are approximately 200 F below ash-softening temperatures when entering the first rows of boiler tubes.

Discussion

J. M. Drabelle.[4] The Milwaukee Electric Railway and Light Company and the late John Anderson are two names which will always be inseparably linked together in the commercial development and use of pulverized-coal firing in large power boilers in this country.

The fundamental principles established by that company and by Mr. Anderson are as applicable today as when they were established. Making use of such principles, the topping unit at the Cedar Rapids Power Station of the Iowa Electric Light and Power Company is of interest.

The boiler has a nominal maximum rating of 300,000 lb of steam per hr at a temperature of 750 F and a pressure of 750 psi gage. The coal fired is from the Illinois strip-mine fields and has a heating value of 10,100 Btu, moisture content 18.3 per cent, an ash content of 10 per cent, and a sulphur content of 2.9 per cent, the ash having a fusion point of approximately 1900 to 2000 F.

The boiler unit and waterwalls were furnished by the Springfield Boiler Company, Springfield, Ill. The furnace is unique in that it is divided into two sections separated by a vertical waterwall in the center. This wall is of the open type and is made up of two banks of tubes, one bank each per furnace section. The total water surface of the sidewalls of the furnace based on projected area is 4315 sq ft. The total water surface facing the fire, including the first row of boiler tubes and the V-bottom ashpit tubes, is 4810 sq ft. The total cubic volume of the two furnaces is 15,385 cu ft. Including the V-bottom ashpit section, the total is 16,125 cu ft.

The principal dimensions of each section of the furnace are width 12 ft 8 in.; depth 19 ft 6 in.; height, from upper section of V-bottom ash section to first row of steam-generating tubes of the boiler, 31 ft 9 in.

The performance of this furnace with pulverized coal has thoroughly proved the accuracy of the author's conclusions, i.e., heat-absorbing surface is so arranged as to avoid hot-gas-flow sections entering the tube bank of the boiler with consequent fouling, bridging, and other difficulties typical of some furnaces.

Each section of the furnace at the nominal rating of the boiler is fired by two pulverized-coal burners handling 9.2 tons of coal per hr per furnace section. There have been no troublesome deposits of any kind; such deposits as have appeared on the sidewall tube bank are of the light, fragile, honeycomb type as described by the author. There has been absolutely no objectionable smoke or other troubles due to this comparatively cold furnace.

J. B. Johnson.[5] From the viewpoint of the inspecting engi-

[4] Consulting Engineer, Iowa Electric Light and Power Company, Cedar Rapids, Ia.

[5] Engineer, The Travelers, Milwaukee, Wis.

neer, it can be mentioned that there is another factor affecting the economy of operation of boiler units of the type described by the author. Observation over a period of years has developed the conclusion that a distinct reduction in time of outage, either scheduled or forced, is obtained by dry-ash-furnace operation. The limiting factor for cooling speed under fan control has been found to consist almost entirely of an hourly temperature drop that is not so great as to be unfavorable to the tube joints in the drum.

Access to the interior of the furnace and boiler passes being appreciably hastened by this factor, it is evident that the availability percentage is less affected by the outages for inspection and maintenance, whether scheduled or otherwise.

There is still another item on the favorable side from the standpoint of the inspecting engineer. At the most, there is a light "whisker ash" which is easily removed, and the bare surfaces are exposed for easy and rapid examination. This again means a saving in outage time and, not only can the inspections be made more rapidly, but their quality is enhanced.

In these days when the color of the metal surfaces has a distinctive message for the inspection engineer, a rapid method of securing a clean surface is indeed appreciated. In the case of the dry-ash furnace, all that is required to secure a truly bare surface is a light water wash, atomized with compressed air.

Summing up, it can be stated that this type of furnace has inherent advantages favoring the availability percentage, due to a decrease in time for cooling the furnace, resulting in quicker attainment of temperatures at which inspection and maintenance operations may be conducted with less time and labor being required for making ready for these operations.

Author's Closure

Use of a dividing wall in the furnace reported by Mr. Drabelle, in order to cool it sufficiently for trouble-free operation with unusually poor coal, is certainly of interest and of significance.

Mr. Johnson's appropriate remarks suggest that better inspection possible with the dry-ash furnace assists in realization of higher reliability.

Steam Locomotives—Notes on Ages and Proportions, With Suggestions for Improvements

By J. L. Ryan,[1] Springfield, Mo.

Locomotives are often built and maintained in kind for their service life, renewal parts being made according to their original design, whereas at but little if any additional cost, they might be renewed to modern design and proportions. In this paper the author discusses the high percentage of the total number of steam locomotives having road assignment which do not have modern proportions and which will be continued in service for many years. This situation leads to the suggestion that increased capacity and economy may be built into them at slight additional expense by following the practice of making required maintenance renewals according to modern proportions, with particular emphasis on adequate steam space, increased gas area through the boiler, a high degree of superheat, and valve events for speed and capacity.

The demand for faster service, longer runs, and high mileage on the railroads has left almost all of them with many locomotives on their hands which are not adapted to meet such requirements. In other words, the horsepower demand cannot be met. Many of these locomotives may be improved for faster and more sustained service by making changes which will not incur a great deal of expense.

From month to month one may see, in publications concerned with railway transportation, articles giving the proportions and design features of locomotives that are being delivered to some railway. If the reader is not something of a student of the motive-power field, he may come to the conclusion that the railways are being well stocked with new locomotives. This, however, is far from being true. In fact, time passes so rapidly, making obsolete locomotives which we are inclined to consider as modern, that those of us concerned with motive-power problems may well be startled by the actual conditions when making compilations of the locomotives handling our transportation services, their ages, proportions, and construction.

Degree of Obsolescence of Nation's Motive Power

In attempting to approximate the extent to which our road-service steam motive power may be considered modern, the author uses as an example the locomotives of his employer, which is considered an average-size railway. Out of an ownership of 610 locomotives, 425 or 70 per cent are assigned to road service. In view of the speeding up of freight and passenger schedules, the horsepower rating of locomotives is a better yardstick to apply than the rated tractive effort which is so frequently used. Thus, using Cole's values for cylinder-horsepower rating for locomotives built prior to 1920, and the railway company's test results for those built in 1920 and later, the 425 locomotives having road assignment have a rating of 1,096,100 hp, an average of 2579 hp each. Table 1 indicates the periods in which certain of these locomotives have been built.

TABLE 1 PERIOD OF BUILDING LOCOMOTIVES OF THE ST. LOUIS-SAN FRANCISCO RAILWAY

	Horsepower rating	Percentage of total
1935 and later	132700	12.0
1930 and later	220700	20.0
1923 and later	543200	49.5
1919 (U.S.R.A.) and later	623500	56.7

Of the locomotives producing 220,700 hp built or rebuilt in 1930 and later, only 132,700 hp, or 12 per cent of the total considered, fully meet the transportation department's operating requirements and have the desired proportions for economy of operation and maintenance. There are 31 locomotives included in this 12 per cent, having an average rating of 4300 hp. Numerically, these locomotives are 7.3 per cent of the total having road-service assignment.

Now considering the railroads as a whole, we find that, in 1939, reports[2] were filed for 45,965 steam locomotives. Should the road-service ratio of 70 per cent be applied to the 45,965 steam locomotives reported in order to arrive at the approximate number having road assignment, we would have a total of 32,175.

The record[3] of purchases of steam locomotives for service in the United States, 1934 to 1939, inclusive, is given in Table 2.

TABLE 2 LOCOMOTIVE PURCHASES; 1934–1939

Construction orders placed	Service: Road	Service: Yard
1934	63	9
1935	17	11
1936	349	84
1937	149	27
1938	33	2
1939	88	2
Total	699	135

Numerically the 699 steam locomotives listed in Table 2, purchased for road service, constitute only 2.2 per cent of the 32,175 steam locomotives considered as having assignment to this service. These new locomotives have approximately double the rated horsepower capacity of the average of the total and accumulate mileage at rates 2 to 3 times that of the average. On this basis, they should account for 10 to 15 per cent of the transportation movement. This leaves 85 to 90 per cent of the movement being handled by locomotives built prior to 1934. A number of the freight locomotives, built in the period 1928 to 1931, were proportioned to meet present operating requirements; the majority, however, while having good boiler proportions and good steam

[1] Mechanical Engineer, St. Louis-San Francisco Railway Company. Contributed by the Railroad Division and presented at the Semi-Annual Meeting, Milwaukee, Wis., June 17–20, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

[2] Twenty-Eighth Annual Report of the Chief Inspector, Bureau of Locomotive Inspection, Interstate Commerce Commission, U. S. Department of Commerce, Washington, D. C., 1939.

[3] "Locomotive Purchases by American Railroads," Annual Statistical issues of *Railway Age*, vol. 98, 1935, p. 155; vol. 100, 1936, p. 71; vol. 102, 1937, p. 70; vol. 104, 1938, p. 76; vol. 106, 1939, p. 79; January 6, 1940, p. 78.

distribution, continued with wheel diameters which are a handicap today.

Returning to the figures on the locomotive ownership of the author's employer, it will be observed that, of the 1,096,100 rated horsepower, representing the total capacity of the 425 locomotives assigned to road service, locomotives having 12 per cent of the rated total are considered modern, while locomotives accounting for 44.7 per cent of the rated total were built commencing with the U.S.R.A. period and from then on to the time when those having modern operating proportions were constructed. With the groups of locomotives in mind which will fall within the period of construction of the 44.7 per cent mentioned, it is suggested that studies similar to the following be undertaken with the object of making maintenance replacements as nearly according to modern proportions as possible in preference to the "as-built" proportions. Regardless of our opinions with respect to the economical retirement age of equipment, these locomotives will, in all probability, be continued in service for many years.

In some instances at no additional cost, and in many instances at a nominal additional cost, distinct improvements may be effected in the capacity and economy of locomotives by the reproportioning of parts which are subject to renewal from time to time through the routine of maintenance.

Boiler Proportions Which Should Be Examined

The boiler is an excellent starting point when reviewing the design and proportions of a locomotive for possible improvement.

Many of the boilers designed in the days of drag service have inadequate steam space for the high steam-release rate obtained under present operating conditions. When a new firebox is applied, this condition can be readily corrected. The lowering of the crown sheet 3 in. will increase the volume of the steam space 20 to 25 per cent. This is frequently sufficient to transform a poor water-carrying boiler into a good performer. When this is effected, the results are:

(*a*) Better performance on line of road;

(*b*) Higher superheat temperature;

(*c*) Reduction in maintenance of valves, pistons, and superheater units.

Locomotives, designed for operation on heavy-grade lines and having permanent reassignment where only light-grade lines are encountered, should be checked for the lowest reading of the water glass relative to the highest point of the crown sheet and for the visible length of the water glass used. A gain of 15 to 20 per cent in steam space is at times possible by a slight lowering of the water glass and reduction in its visible length, maintaining the same degree of safety in operation on the light-grade line as prevailed on the heavy-grade line for which the locomotives were built.

The gas area through the barrel of the boiler is one of the all-important details which should be checked in order to provide the maximum attainable. In the design of locomotives constructed in the period 1919 to 1930, some railroads incorporated practices in the spacing of tubes which today are recognized as not being consistent with spacing that may be followed with good results, water treatment and welded flues effecting this permissible change.

A case in point was the building some years ago of 50 type-2-8-2 locomotives by a certain railroad, following in detail the boiler dimensions of the U.S.R.A. 2-8-2 B, except for the layout of the tube sheets. The latter type had 45 flues $5^1/_2$ in. in diam, and 247 tubes $2^1/_4$ in. in diam. The 50 locomotives of the 2-8-2 type mentioned have 45 flues $5^1/_2$ in. in diam and 219 tubes $2^1/_4$ in. in diam.

A kindred condition can also be found in the proportioning of some boilers having combustion chambers with the water space around the chambers greater than is now required for good practice. The area of the back tube sheet is generally the limiting factor in the tube application to these locomotives. A reduction of the water space around the combustion chamber when applying a new firebox could be capitalized upon through the application of additional boiler tubes.

Boiler-Tube-Sheet Layout and Superheat

With the results at hand on the improved cylinder performance of modern and semimodern locomotives, a high percentage of which is attributable to steam-chest temperatures of 700 to 750 F, when we apply new tube sheets in the course of maintenance, the reproportioning of the tube layout to provide high steam-chest temperatures offers an excellent opportunity for increased capacity and economy.

The tube-sheet layout of the U.S.R.A. locomotives is proportioned so that the $5^1/_2$-inch flues will have a flue-gas area of 45 to 46 per cent of the total gas area through the boiler. Such a proportion with the type-A superheater gives a steam temperature approximately 100 F below that desired in today's operation.

Table 3 shows the tube-and-flue application with resulting proportions for the U.S.R.A. 2-8-2 B locomotive as built, as well as a number of possible applications without requiring any change in the crown height or water space around the combustion chamber. The order of application to attain increased capacity, as well as for fuel economy, would be as follows:

1 Type-E superheater.

2 A 6 × 9 layout of $5^1/_2$-in. flues with the application of HA superheater units or their equivalent.

3 A 6 × 10 layout of $5^1/_2$-in. flues, with the top corner flues omitted; application of 58 type-A superheater units.

Increasing the capacity of the superheater effects a material gain in addition to that of reducing the steam rate per unit of work, since the increased number of units reduces the pressure drop, which at a high work rate is equivalent to a substantial increase in the boiler pressure.

TABLE 3 POSSIBLE TUBE AND FLUE APPLICATIONS ON U.S.R.A. 2-8-2 B TYPE LOCOMOTIVE WITHOUT CHANGE IN DIMENSIONS OF BACK TUBE SHEET

Superheater type	A	HA or equivalent	A	E
Superheater flue layout	5 × 9 (As built)	6 × 9	6 × 10	......
Distance over tube sheets, ft	19	19	19	18
Number of $5^1/_2$-in. flues	45	54	58	201–$3^1/_2$
Number of $2^1/_4$-in. tubes	247	217	196	62–$2^1/_4$
Vertical pitch of $5^1/_2$-in. flues, in.	$6^1/_2$	$6^1/_2$	$6^1/_2$	4.22 (Mean pitch)
Pitch of $2^1/_4$-in. tubes, in.	3	$2^{15}/_{16}$	$2^{15}/_{16}$	3
Heating surface of flues, sq ft	1226	1471	1580	3298
Heating surface of tubes, sq ft	2752	2418	2184	697
Total heating surface, tubes and flues, sq ft	3978	3889	3764	3995
Superheater heating surface, sq ft	993	1742	1280	1920
Steam area through superheater, sq in.	51.3	61.6	66.1	71.92
Net gas area through boiler, sq in.	1414	1447	1438	1427
Net gas area through $5^1/_2$-in. flues, per cent	45.1	52.9	57.2	......
Approximate temperature range of steam in branch pipe, deg F at high work rate	630–640	HA 710–730	700–720	710–730
Maximum evaporation, tubes and flues (Cole's values)	37984	37125	36175	39320
Maximum evaporation, including firebox heating surface (Cole's values)	54809	54010	53060	56205

TABLE 4 EXAMPLES OF REPROPORTIONED TUBE-SHEET LAYOUT BY ST. LOUIS-SAN FRANCISCO RAILWAY

	Example No. 1			Example No. 2	
	As built	Reproportioned	Reproportioned	As built	Reproportioned
Superheater type	A	A	HA	A	A
Superheater flue layout	5 × 9	7 × 9	6 × 9	5 × 8	6 × 8
Combustion chamber—with or none	With	With	With	None	None
Back tube sheet altered	..	No	No	..	Crown sheet lowered 3 in.
Distance over tube sheets, ft-in.	22-0	22-0	20-0	21-0	20-11
Number of $5\frac{1}{2}$-in. flues	45	63	54	38	48
Number of $2\frac{1}{4}$-in. tubes	251	211	242	225	176
Vertical pitch of $5\frac{1}{2}$-in. flues, in.	$6\frac{3}{4}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$	$6\frac{1}{2}$
Pitch of $2\frac{1}{4}$-in. tubes, in.	$3\frac{1}{8}$	$3\frac{1}{16}$	$3\frac{1}{16}$	$3\frac{1}{8}$	$3\frac{1}{16}$
Heating surface of flues, sq ft	1420	1988	1548	1144	1440
Heating surface of tubes, sq ft	3240	2724	2839	2772	2160
Total heating surface, tubes and flues, sq ft	4660	4712	4387	3916	3600
Superheater heating surface, sq ft	1233	1726	1834	978	1235
Steam area through superheater, sq in.	51.3	71.8	61.6	43.3	54.7
Net gas area through boiler, sq in.	1426	1556	1500	1245	1233
Net gas area through $5\frac{1}{2}$-in. flues, per cent	44.7	57.4	49.3	43.3	55.2
Temperature range of steam in branch pipe, deg F, at high work rate	590–620	690–710	[a]690–720	...	680–700
Maximum evaporation, tubes and flues (Cole's values)	41610	42220	41370	36010	33170

[a] Combustion chamber lengthened and siphon applied. Firebox heating surface increased 18.3 per cent.

TABLE 5 EXAMPLES OF CHANGE IN VALVE EVENTS AS MADE BY THE ST. LOUIS-SAN FRANCISCO RAILWAY TO MEET ALTERED OPERATING REQUIREMENTS; EXAMPLES OF RECENT CONSTRUCTION

	Example No. 1		Example No. 2		Example No. 3 recent construction		
	Original	As altered	Original	As altered			
Locomotive type	4-6-2	4-6-2	4-6-2	4-6-4	2-8-2	4-8-2	4-8-2
Class of service	Passenger	Light fast passenger	Heavy passenger	Conversion, heavy fast passenger	Freight	Freight	Freight
Boiler pressure, psi	200	200	210	225	235	250	210
Cylinders, diameter and stroke, in.	24 × 28	24 × 28	26 × 28	26 × 28	27 × 32	27 × 30	29 × 32
Drivers, diameter, in.	69	73	74	74	64	70	70
Valves, diameter, in.	13	13	13	13	14	14	15
Maximum travel, in.	$7\frac{1}{2}$	$7\frac{1}{2}$	$6\frac{1}{2}$	$7\frac{1}{2}$	$8\frac{1}{2}$	$7\frac{3}{4}$	$8\frac{1}{4}$
Steam lap, in.	$1\frac{1}{4}$	$1\frac{1}{2}$	1	$1\frac{5}{8}$	$1\frac{15}{16}$	$1\frac{3}{4}$	$1\frac{15}{16}$
Lead, in.	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{2}{16}$	$\frac{3}{16}$
Exhaust clearance, in.	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{3}{16}$	0	0	0
Maximum cutoff, per cent	...	82	...	79	77	77	76

Table 4 contains examples of reproportioning the superheater application on two classes of locomotives by the author's employer.

Effect of Valve Events on Locomotive Operation

As important as the proportioning of the boiler and the superheater, are the valve events upon the operation of a locomotive. Classes should be checked having in mind today's assignment. A 4-6-2 type built to handle the heavy trains of another period, with valves having 1 to $1\frac{1}{8}$-in. steam lap, should not be assigned to light, high-speed trains without altering the valves and valve gear to provide events to suit. Locomotives of the 2-8-2 type, designed in the days of drag service, may be found operating on near passenger schedules and with practically the original restricted steam ports and valve events. This necessarily results in loss of power and fuel.

Classes, which are receiving the application of new cylinders, should have the diameter of the valve, area of the exhaust channels, and the steam ports carefully examined. They should be proportioned to meet today's requirements. Only a few locomotives need be involved to justify the cost of a new cylinder pattern, should it be required, in order to obtain the desired proportions.

Considering the fact that locomotives in freight service are rated today on their power output at piston speeds of 1200 to 1400 fpm, instead of on their initial tractive effort, the responsibility devolves upon the mechanical engineers at least to point out the potential power increases which may be effected through moderate changes. At the time of heavy shopping, a valve gear, providing drag-service events, can be replaced with a gear providing modern events, often at slight cost over that which would be involved in maintaining the original in kind.

Table 5, examples Nos. 1 and 2 are instances of altering the valve gears to meet changed assignments and operating conditions. In both cases the originals were for passenger service with running speeds of 55 to 60 mph. The alterations were made to provide valve events to accommodate an economical cruising speed of 70 to 75 mph, with occasional top speeds of 80 to 85 mph.

Should one review a table showing the steam lap, lead, exhaust clearance, valve diameter, etc., for the various locomotives recently built, the question could well be asked what proportions and valve events should be provided to meet today's operating requirements most satisfactorily? The locomotives which we are considering are those built from 1919 to 1930, the majority having working pressures within the range of 200 to 250 psi.

The problem is to provide the highest possible mean effective pressure at piston speeds of 1200 fpm and higher. L. H. Fry's recent review[4] of the reproportioning of locomotives by the Paris-Orleans Railway may be read to advantage by those having to do with steam-locomotive proportions; also by those having to do with the maintenance. In the latter case, the review should be studied in order that a better understanding will exist when a slight increase in maintenance is assumed in order to effect a substantial increase in the work-rate capacity.

Indicator cards, Figs. 1 to 4, inclusive, are shown as an example of the increase in mean effective pressure which may be effected through the adoption of a long steam lap. They were taken from a 2-8-2-type locomotive having 45 type-A superheater units, 27 × 32-in. cylinder, 14-in-diam valve, $8\frac{3}{4}$-in. maximum travel. The valve setting for cards, Figs. 1 and 3, was as follows: $1\frac{1}{4}$-in. steam lap, $\frac{3}{16}$-in. lead, 0-in. exhaust clearance, $1\frac{5}{8}$-in-width steam port. The valve setting for cards, Figs. 2 and 4, was as follows: $2\frac{1}{2}$-in. steam lap, $\frac{3}{16}$-in. lead, $\frac{1}{16}$-in. exhaust lap, $2\frac{3}{16}$-in-width steam ports.

Example No. 3 Table 5 shows the steam lap, lead, exhaust clearance, valve diameter, maximum cutoff, etc., which the author's company uses as the most practical for fast heavy freight service. With a $1\frac{15}{16}$-in. steam lap and valve travel to provide 75 to 77 per cent maximum cutoff, auxiliary starting ports are not required. Cards shown in Figs. 5 and 6 were

[4] "The Locomotive in France," by L. H. Fry, *Railway Mechanical Engineer*, vol. 112, 1938, p. 473; vol. 113, 1939, p. 1; vol. 113, 1939, p. 345.

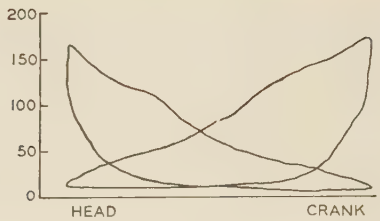

Fig. 1

Piston speed	953 fpm
Cutoff	36.5 per cent
Boiler pressure	199 psi
Mep, head end	53.8 psi
Mep, crank end	66.7 psi
Horsepower, r.s.	1004
Total engine horsepower	2008

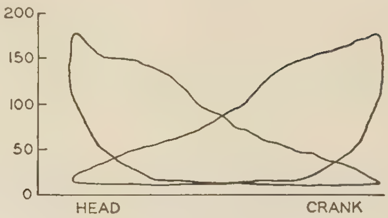

Fig. 2

Piston speed	1041 fpm
Cutoff	35.8 per cent
Boiler pressure	195 psi
Mep, head end	69.2 psi
Mep, crank end	76.1 psi
Horsepower, r.s.	1330
Total engine horsepower	2660

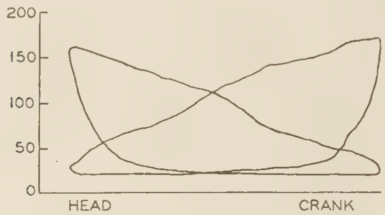

Fig. 3

Piston speed	991 fpm
Cutoff	53 per cent
Boiler pressure	198 psi
Mep, head end	67.7 psi
Mep, crank end	76.8 psi
Horsepower, r.s.	1257
Total engine horsepower	2514

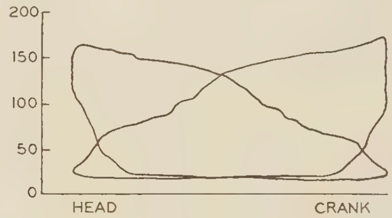

Fig. 4

Piston speed	964 fpm
Cutoff	51 per cent
Boiler pressure	193 psi
Mep, head end	86.6 psi
Mep, crank end	93.8 psi
Horsepower, r.s.	1531
Total engine horsepower	3062

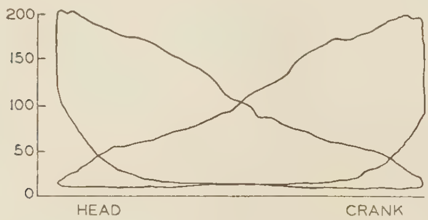

Fig. 5

Piston speed	1041 fpm
Cutoff	38 per cent
Boiler pressure	237 psi
Mep, head end	91.3 psi
Mep, crank end	92.3 psi
Horsepower, r.s.	1629
Total engine horsepower	3258

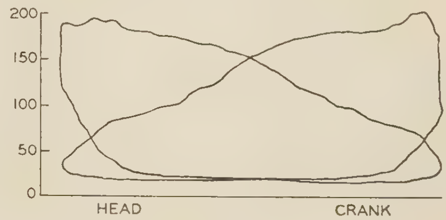

Fig. 6

Piston speed	1016 fpm
Cutoff	52 per cent
Boiler pressure	237 psi
Mep, head end	111.5 psi
Mep, crank end	115.0 psi
Horsepower, r.s.	1960
Total engine horsepower	3920

taken from the 2-8-2-type locomotive listed in Table 5, having 14-in-diam valves with $1^{15}/_{16}$ in. steam lap. These cards were taken when the locomotive was operating with a piston speed of approximately 1000 fpm. It is not difficult to visualize the shrinkage in the mean effective pressure which would result from either a reduction in the diameter of the valve or in the steam lap. With today's piston speeds of 1200 to 1600 fpm, it is doubly important that the inflow and outflow of steam be as unrestricted as is practicable. The use of valves of a diameter which may be considered large need not incur excessive weight. Lightweight built-up valves using gas pipe with a wall thickness of $^3/_{16}$ in. and steel castings having $^1/_4$ in. section have been standard practice with the author's company for 7 years.

Boiler Pressure

In the light of the excellent on-line-of-road operation which has been obtained from locomotives of recent construction, with working pressures within the higher pressure range, one might be inclined to the thought that the 200 to 225 psi working pressure, to which many of the locomotives built in the 1919 to 1929 period are limited, presents an extreme handicap to one attempting to provide increased economy and power with which to meet today's operating requirements.

From the economy viewpoint, some encouragement may be obtained from a review of the design of the recently built high-pressure locomotives. The adoption of the higher working pressures without modification of design to provide for increased ratio of expansion does not admit of the increase in thermal efficiency of the engine which is ordinarily considered a result of the use of the higher pressure. The decrease in the differential between the two is particularly true where a heavy work rate with reduced ratio of expansion is involved, the locomotives in both pressure ranges working at approximately the same cutoff or with the same ratio of expansion.

High pressure is forced at times where high piston thrust is required, and the cylinder diameter must be limited to keep within clearance limits. There is no denying that high pressure gives an engine a "smartness" of response; however, from the

standpoint of capacity and for operation within the present operating requirements of high-speed freight service, much may be accomplished with working pressures of 200 to 225 psi. An example of this is on one of the divisions of the author's company, where the two groups of 4-8-2-type locomotives listed in example No. 3 of Table 5 are in a pool. The steaming capacity of the locomotives in the two groups is approximately the same. The 250-lb locomotives have 54 type HA superheater units with 61.6 sq in. of steam area through them. The size of the valves, events, etc., are given in Table 5.

When it was decided to condition the group of locomotives having 210 psi to work in a pool with the 250-lb locomotives, the $5^1/_2$-in. flues were increased from 45 to 63, with a resulting increase in steam area through the superheater units from 51.3 to 71.8 sq in.; the dry pipe and branch pipes were increased to suit; 29-in. cylinders having 13-in-diam valves were replaced with 29-in. cylinders having 15-in-diam valves; valve gears providing $6^1/_2$ in. maximum travel were replaced with gears providing $8^1/_4$ in. maximum travel; valves having 1-in. steam lap were replaced with valves having $1^{15}/_{16}$-in. steam lap. There is some difference in the response of the two groups, so far as the enginemen are concerned, but none, so far as the dispatchers are concerned, both handling the same tonnage on the same schedules.

Steaming Capacity—Value of Feedwater Heating

When treating the subject of providing the maximum possible capacity in existing steam locomotives, the steaming capacity which may be added to the boiler at high work rates by the application of feedwater-heating equipment, utilizing exhaust steam, should be analyzed and a distinction made between the percentage of return on the investment and the percentage increase in power; also that a net 10 per cent increase in boiler capacity is 12 to 13 per cent at the drawbar.

Conclusion

The groups of locomotives that were built in the years 1919 to 1930 offer, in general, a fertile field for a substantial addition to the work-rate capacity of our locomotives through the adoption of a policy of providing proportions to give a high degree of superheat, low pressure drop from boiler to steam chest, and valve events to conform with present-day operating requirements, these changes building up the mean effective pressure at the higher work rates and without increasing the maximum stress in frames, driving axles, crankpins, rods, etc.

These suggestions are not advanced with the intention of detracting from the thought and effort which ordinarily are put into the development of new locomotives but merely to take advantage of the opportunity that is offered daily in our shops to build greater capacity into locomotives which are having renewals made as maintenance routine and involving only a nominal amount of re-engineering.

Discussion

C. T. Ripley.[5] This paper suggests the need for improvement in valves and valve gears, in order to get better performance while a locomotive is working at high capacity. The writer is in agreement with the author's conclusions as to the possibilities of the poppet-type valve. There appears to be no question but that better results can be secured with this type of valve, provided the operating mechanism is satisfactory. Some years ago experiments were made with the Caprotti design on American locomotives. These showed a number of desirable features, but difficulties developed in making the parts rugged enough to withstand service in large locomotives. This design did, however, show very satisfactory drifting characteristics.

This matter of drifting has not been referred to, although it is a most important one. Practically all American locomotives have been built without any suitable device to make them drift satisfactorily, unless considerable steam is used in the cylinders. There have been numerous so-called drifting valves, but these have been inadequate to meet the requirements. On downgrades it is necessary to have the throttle open considerably beyond the pilot-valve stage, in order to get proper lubrication and avoid pounding. This results in a wastage of fuel and the necessity for increased braking, which also causes a loss in brake shoes and wheels. In some of the western mountain territory, locomotives drift as much as 28 per cent of the total time; for example, from Albuquerque to Gallup, a distance of 127 miles, they drift about 27 per cent of the time. From Gallup to Albuquerque, 127 miles, about 28 per cent of the time. From Seligman to Needles, 150 miles, they drift 67 per cent of the time. It is appreciated that these are extreme conditions and apply only to mountainous territory. However, there is a very considerable amount of drifting even on the more level eastern railroads. There is a distinct need for improvement in devices to permit of more economical and satisfactory drifting.

One western railroad made a study of this matter and designed large by-pass valves, connecting the ends of the valve chambers, similar to the practice on German locomotives. These connections must be about 9 or 10 in. in diam, in order to meet the requirements. They are automatic in operation, as the pressure in the steam chest closes the by-pass valve. When the pressure is not present, coiled springs act upon the valve to open it. There is also a steam connection to the cab, which can be used in emergency in case the valves are stuck open, but it is seldom necessary to use it. Operation of these valves has been very satisfactory and there is no difficulty from pounding or in maintaining lubrication in the cylinders. They have reduced the amount of braking necessary on the trains and have resulted in marked fuel savings. Recent tests have shown a saving of 7 per cent in fuel, while operating over the territory between Barstow, Calif., and Albuquerque, New Mexico, a distance of 748 miles. It may be noted that the saving in braking, through use of proper types of drifting valves, increases in the case of locomotives and trains equipped with roller bearings with their lessened friction.

The use of the poppet valve will probably make it unnecessary to have any special devices applied for improving drifting characteristics. In the case of old engines which are to be rebuilt, as suggested by the author, and in designing new engines with piston valves, some consideration should be given to this matter.

Arthur Williams.[6] Those familiar with the locomotives of the St. Louis-San Francisco Railway referred to in this paper will agree that the author and the mechanical department of that railway have produced excellent results. It is the writer's impression that this is due to the careful consideration given to all details of locomotive and cylinder design. In discussing the various items, the paper starts with the possibility of lowering the crown sheet to increase the volume of the steam space, and finishes with the valve gear and valve settings. Between the two points, the gas area through the boiler, steam area through the boiler, steam area through the superheater units and steam pipes, degree of superheat, and pressure drop of the steam are all considered.

In general, it is possible to improve a locomotive with a small superheater by increasing the number of superheater units and flues. This will increase the superheater heating surface and

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[6] Manager, Production Engineering Division, The Superheater Company, East Chicago, Ind. Mem. A.S.M.E.

superheat, but care must be taken that the evaporative heating surface is not decreased too much with a corresponding decrease in boiler efficiency. In Table 4 of the paper an increase in the heating surface of the tubes and flues was obtained with the larger superheater, but this was done with a decrease in the flue spacing. It is not always possible to do this. The type-E superheater is designed to give the maximum superheater and evaporative heating surface so that a high superheat is obtained at the same time as a low smokebox temperature and high boiler efficiency.

Author's Closure

The author believes that Mr. Ripley's discussion of the poppet-type valve refers either to the paper[7] presented by Charles F. Krauss, "Notes on the Trends in Reciprocating Valve Mechanisms Employing Piston Valves" or to the paper[7] presented by A. G. Hoppe, "Notes on Valve and Valve Motion Designed for Modern High-Speed Passenger Steam Locomotives." The paper presented by the author did not touch on the subject of the poppet-type valve.

Mr. Williams' discussion points out that in general it is possible to improve a locomotive with a small superheater by increasing number of superheater units and flues, but that when increasing the superheating surface care must be taken that the evaporative heating surface is not decreased too much with a corresponding decrease in boiler efficiency. The author agrees with Mr. Williams, but adds that the increased degree of superheat obtained through an increase in the number of superheater units effects an improvement in cylinder performance throughout the entire range of work rates from low to high. The decreased evaporating capacity is not vital except at the higher work rates and then is generally more than offset by the decreased steam rate resulting from the higher degree of superheat. This is particularly true of oil-burning locomotives.

[7] Abstracts of both of these papers were published in *Mechanical Engineering*, vol. 63, February, 1941, pp. 140–144.

Mill Drying of Coal

By M. E. FITZE,[1] MILWAUKEE, WIS.

This paper develops the fact that the drying of coal in the mill while grinding, instead of in separate driers previous to mill operations, makes possible large savings in equipment, building, and operating costs, besides making possible a net gain in boiler-plant efficiency in the order of 0.5 per cent. This gain is due to improvement in air-heater performance, resulting from lesser gas flow through it as well as the reduction of approximately 6 per cent of the total gas to 150 F instead of the usual 350 F. Use of an auxiliary cyclone in the mill-vent circuit, with return of the coal separated therefrom to the mill, results in a coal loss from the system of only about 0.25 per cent.

IN THE burning of coal under power boilers one of the unavoidable losses involved is due to the moisture content of the coal as fed to the furnace. The loss is measured at the point where the flue gases leave the unit and consists of the heat of the liquid between the temperature of the incoming coal and the boiling point at atmospheric pressure, plus the latent heat of vaporization and the heat of superheat between the boiling temperature and the temperature of the outgoing gases. This is one of the minor losses in the burning of coal; nevertheless, it does attain appreciable proportions for the higher moisture coals, as shown in Fig. 1, which gives the percentage of loss for 12,000-Btu coal for various moistures and final flue-gas temperatures.

Besides the efficiency loss caused by the moisture content of the coal, moisture in the flue gas is a contributing factor to the accumulation of deposits in the lower temperature regions of air heaters and economizers, especially with the higher sulphur coals. These deposits seriously impair the heat-absorbing capacity of the surface and, in extreme cases, result in reduction of load-carrying capacity and sometimes in forced shutdown. Their removal is often difficult and expensive.

The extraction of moisture from coal for stoker firing has not been given much attention; in fact, some conditions of operation require "tempering" of the coal with water to improve combustion.

In order that over-all plant-efficiency gains, due to removal of moisture from coal may be fully realized, the moisture must be removed prior to its entrance to the furnace and be discarded to some point entirely beyond the heat-absorbing areas. Its removal must be accomplished with heat of the lowest temperature head available, preferably with otherwise waste gases.

It is evident that in the unit-fired system of pulverized-coal burning, hot-air injection into the mill is of value only in so far as it benefits mill operation and capacity. The moisture removed from the coal is injected directly into the furnace and constitutes the same loss as would be encountered in stoker firing.

The bin-and-feeder system requires removal of moisture (especially surface moisture) down to the lowest practicable value, in order that the powdered coal in handling, storing, and feeding will flow freely and uniformly at all times.

Early Drying Methods

Early developments in the art of pulverized-fuel firing by the bin-and-feeder system concentrated the drying efforts on the coal previous to pulverization. The first generally successful driers were of the cement-kiln type; inclined drums some 5 ft in diam × 40 ft long, rotating about 3 rpm. The coal was fed into the upper end, gradually working down and out at the lower end by the process of slow rotation. The coal was lifted by suitable "pick-up" plates on the inside of the drum and dropped through the hot gases which were admitted at the lower end and removed from the upper. Fines carried out by the gases were removed by a cyclone separator before the gases were discharged. Unless the drier was close enough to the boiler flue to make possible a supply of heat from that source it was necessary to equip the drier with a separately fired furnace of its own. To prevent overheating of the drier, large amounts of excess air were used which, of course, meant inefficient utilization of this portion of the coal (approximately 0.5 per cent) charged to the plant.

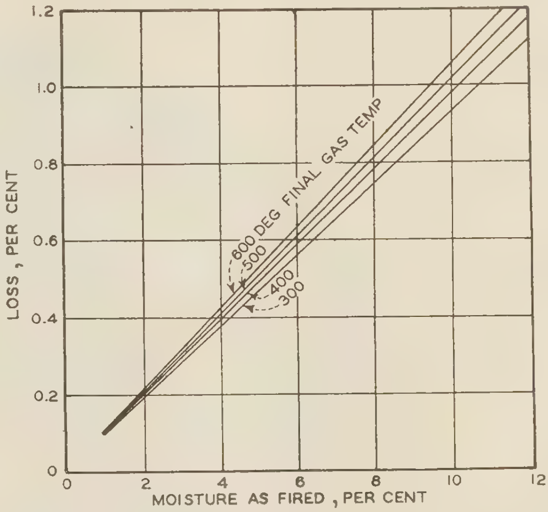

FIG. 1 MOISTURE LOSS FOR VARIOUS MOISTURE CONTENTS AND FINAL FLUE-GAS TEMPERATURES
(12,000 Btu per lb of coal as fired.)

Later, driers were developed in which the coal was passed over steam-heated grids or plates with a stream of air passing counterflow to the coal. This made possible the utilization of bled steam from the turbines, but capacities were not very high for the space occupied and, where long steam lines were required, considerable installation expense was involved.

All these separate drying schemes were predicated on the use of a pulverizing plant separate from the rest of the boiler plant. This involved:

1. High building and equipment cost.
2. High maintenance cost.
3. High operating labor cost.

Mill-Drying Development

In order to overcome these three high-cost factors of separate

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coal drying and milling and, at the same time, to retain the inherent high efficiency and all-around operating flexibility of the bin-and-feeder system, the mill drying of coal with flue gas taken from the latter part of the working cycle was developed.

Fig. 2 shows diagrammatically the arrangement of mill-drying equipment installed in three plants of the Wisconsin Electric Power Company and now being installed in a fourth.

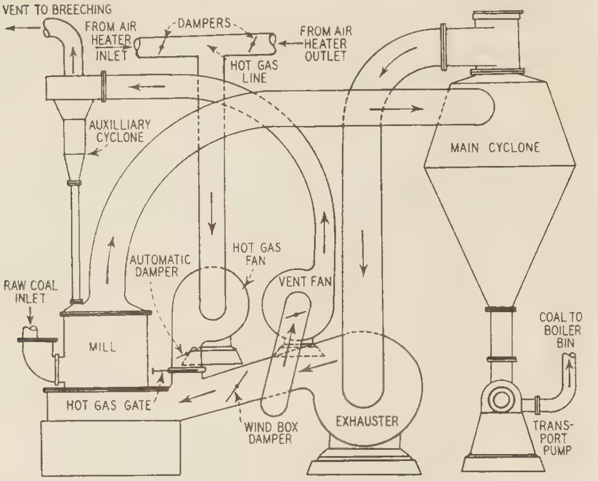

FIG. 2 DIAGRAMMATIC ARRANGEMENT OF MILL-DRYING EQUIPMENT USING HOT FLUE GAS FROM BOILER UNIT AS SOURCE OF HEAT

Flue gas is taken from the last pass of the boiler and the air-heater outlet, properly proportioned by suitable damper control to give the desired temperature, and injected by the hot-gas fan into the mill circuit at the entrance to the mill wind box. Gas temperatures about 550 F are used. In order to maintain pressure in the mill circuit at some constant value and to rid the circulating system of moisture removed from the coal, a vent fan withdraws from the circuit, at the discharge of the main mill fan and just prior to the injection of the hot gas, such volume of moisture-laden gases as is necessary. The vent fan discharges through a cyclone separator and thence to the flue for discharge with the boiler flue gas.

It will be noted that the mill exhauster is located on the clean side of the main cyclone, obviating the necessity of accelerating the entire mill output through the fan wheel. This results in a saving in fan power and blade erosion which is considerable. It also makes possible the use of a more efficient type of fan wheel. The power saving may amount to 2 or 3 kwhr per ton of coal.

The auxiliary collectors discharge their separated coal into the top of the mill; the principle of allowing considerable downward gas flow through their bases aids their efficiency materially. When separating coal of such fineness that 98 per cent passes a 325-mesh sieve, their efficiency averages upward of 80 per cent when 500 cfm is induced downward through the apex.

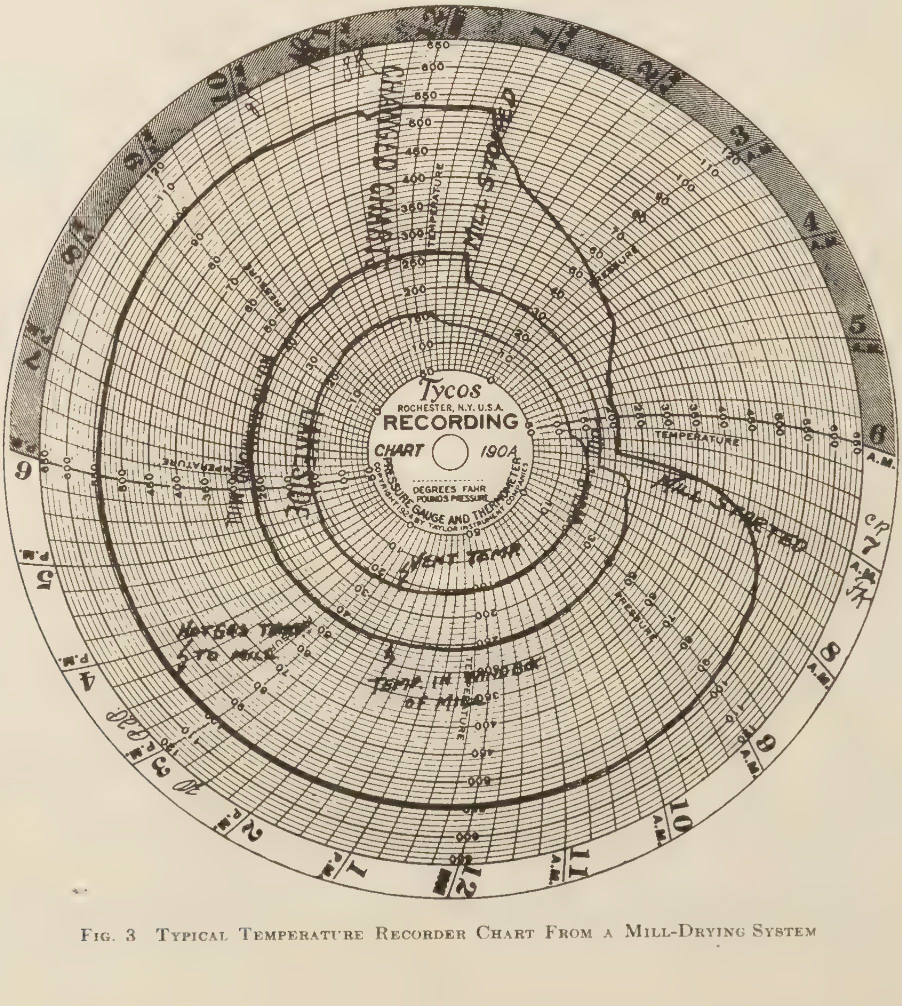

FIG. 3 TYPICAL TEMPERATURE RECORDER CHART FROM A MILL-DRYING SYSTEM

TABLE 1 TEST RESULTS OF A MILL-DRYING SYSTEM

General	
Date	1/8/30
Mill	20
Kind of coal	High fusion (Eastern)
Coal milled, tons	43
Hours run	3.58
Mill-motor current, amp	230
Pump-motor current, amp	23
Air Data	
Top of mill pressure, in. water	−6.9
Fan-inlet pressure, in. water	−17.4
Fan-discharge pressure, in. water	1.8
Pressure rise through fan, in. water	19.2
Wind-box pressure, in. water	−0.34
Wind-box temperature, F	210
CO_2 in hot gas, per cent	12.0
CO_2 in mill system, per cent	10.5
Air leakage, per cent	12.5
Air leakage (mostly through feeders), lb per hr	9440
Humidity of leakage air (approx), per cent	90
Moisture in leakage air, lb per hr	74
Return-air temperature (dry bulb), F	147
Return-air temperature (wet bulb), F	123
Saturation, per cent	50
Velocity head in return pipe ($25\frac{3}{4}$ in. diam), in. water	1.62
Volume through mill, cfm	20100
Vent-fan-inlet pressure, in. water	−2
Vent-fan-discharge pressure, in. water	2
Pressure rise, in. water	4
Vent-fan speed, rpm	640
Velocity head in vent-fan-discharge pipe ($23\frac{3}{8}$ in. diam), in. water	0.139
Vent, cfm	4700
Vent, lb per hr	17700
Moisture per lb of dry vent air, grains	590
Moisture removed in vent, lb per hr	1490
Hot-gas fan-inlet pressure, in. water	−3
Hot-gas fan-discharge pressure, in. water	0.28
Pressure rise through fan, in. water	3.28
Hot-gas fan speed, rpm	700
Velocity head in hot-gas line ($29\frac{3}{4}$ in. diam), in. water	0.048
Hot-gas, temperature, F	470
Hot gas, cfm	5700
Hot gas, lb per hr	14700
Dew point of hot gas, F	100
Moisture per lb dry gas, grains	300
Moisture in hot gas, lb per hr	630
Dust Concentrations	
Vent-cyclone inlet (main cyclone outlet) lb per cu ft	0.0019
Vent-cyclone outlet, lb per cu ft	0.00023
Power Consumption	
Mill motor, kwhr	436
Total kwhr (mill, fan, pump, vent fan, and hot-gas fan)	816
Coal Data	
Proximate analysis as milled	
Moisture, per cent	5.2
Volatile matter, per cent	34.47
Fixed carbon, per cent	55.32
Ash, per cent	10.21
Total, per cent	100
Sulphur, per cent	1.80
Dry, Btu	13279
As received, Btu	12587
Temperature after milling, F	147
Temperature before milling, F	42
Temperature rise, F	105
Moisture before milling, per cent	5.2
Moisture after milling, per cent	2.1
Moisture removed, per cent	3.1
Fineness before milling	
Per cent through $\frac{1}{4}$-in. mesh	61.6
Per cent through $\frac{1}{2}$-in. mesh	88.5
Per cent through $\frac{3}{4}$-in. mesh	93.9
Per cent through 1-in. mesh	100
Fineness after milling (Tyler Standard Screen Scale)	
Per cent through 200 mesh	60.9
Per cent through 100 mesh	81.8
Per cent through 48 mesh	97.16
Per cent through 28 mesh	99.75
Per cent through 20 mesh	99.95
Per cent through 10 mesh	100

Moisture Balance

	Lb per hr	Per cent
Moisture input with coal	1250	64
Moisture input with hot gas	630	32
Moisture input with room air leakage	74	4
Total moisture input	1954	100
Moisture remaining in coal	505	26
Moisture removed in vent gas	1490	76
Total moisture accounted for	1995	102
Moisture unaccounted for	−41	−2

When extracting flue gases for mill drying from adjacent to the air-heater gas inlet, flue-gas losses at the exit of the air heater are reduced, since less heat requires transfer and the ratio of air flow to gas flow is appreciably increased. In cases where no economizers are employed and the boiler-outlet-gas temperature is high, improvement in this ratio nets rapid gains in air-heater performance.

Further reduction in flue-gas losses results with mill-drying systems in the lowering of some 6 per cent of the flue gas to the vent temperature of 150 F instead of the 350 F at the outlet of the air heater. Thus, the cold moist coal is used as an "economizer," reducing flue-gas losses far below any present practice. Returning the vent to the furnace merely to reclaim the dust loss would not make these gains possible, for full gas flow through the air heater would result in the following:

1 No different air-heater performance from usual.

2 Vent gas which had already been cooled to 150 F would be reheated again to air-heater-outlet temperature.

3 Lowered moisture loss would not be realized, since the full moisture content of the coal would be injected into the furnace, just as with unit mill or stoker firing.

4 Air-heater deposits would be aggravated by the higher moisture gases passing through it.

Operation of Drying System

In operation, the system has proved highly successful. During the last 11 years 268,000 mill-hr have been amassed by eight mills aggregating some 55 mill-years of operation. Approximately 4,400,000 tons of both Eastern and Midwestern coal have been ground.

Maintaining an atmosphere of CO_2 in the mill circuit is a safety feature of major importance, for the operating record cited has been accomplished without any fires or explosions having occurred. With this inert gas in the system, safety codes allow the installation of mill-drying equipment within the boiler room without the use of fire walls for separation from other equipment.

In starting up the equipment, a CO_2 content in the system of upward of 12 per cent can be established in 1 min time by closing the wind-box damper shown in Fig. 2, opening the vent damper, hot-gas gate, and hot-gas automatic damper and allowing the high draft in the boiler outlet to pull gases backward through the system from the breeching.

In operation, the automatic hot-gas damper is held open pneumatically by suction in the top of the mill. Should the

TABLE 1 (*Continued*)

Dust Balance and Cyclone Efficiency

Lb per hr of coal in main cyclone outlet (vent-cyclone inlet)	2290
Lb per hr of coal in vent-cyclone inlet	540
Lb per hr of coal in vent-cyclone outlet	65
Main-cyclone efficiency, per cent	90.5
Vent-cyclone efficiency, per cent	88
Coal lost in vent, per cent	0.27

Heat Balance

(Datum is return-air temperature)

	Mill Btu per hr	Per cent
Heat input		
Hot gas, above return air	1.19	68.4
Electrical (mill and exhauster)	0.55	31.6
Total	1.74	100
Heat output		
Sensible heat in coal	0.63	36.2
Heat-up and evaporate moisture	0.88	50.6
Heat-leakage air	0.23	13.2
Total accounted for	1.74	100
Unaccounted for	0	0

Mill Performance

Tons milled per hr	12
Mill power, kwhr per ton	10.1
Total power, kwhr per ton	19

TABLE 2 PRINCIPAL INSTALLATION AND OPERATING DATA FOR THREE MILL-DRYING INSTALLATIONS OF WISCONSIN ELECTRIC POWER COMPANY

Plant	Lakeside	Port Washington	East Wells Street
Year installed	1929-1930	1935	1938
No. of units installed	4	2	2
Type of mill	Roller	Roller	Bowl
Rated capacity, tons per hr	15	15	12
Total hours run to date	212500	48000	7700
OPERATING DATA (1939)			
Kind of coal	Midwestern	Eastern	Operating data not available
Tons milled	361900	181000	
Mill-hours	21620	10920	
Power consumed, kwhr	6433000	2730000	
Tons, milled per hr	16.7	16.6	
Kwhr consumed per ton	17.8	15.1	
Moisture			
Inlet, per cent	9.03	4.0	
Outlet, per cent	5.06	1.8	
Removed, per cent	3.97	2.2	
Fineness, per cent, through			
200 mesh	65.87	66.42	
100 mesh	85.56	89.10	
48 mesh	98.32	98.20	
28 mesh	99.81	99.83	
20 mesh	99.98	99.98	
10 mesh	100	100	

main exhauster fail, the loss of suction in the mill releases a holding latch and the damper is closed by a falling weight. This prevents injection of heat into the system at a time when it cannot be circulated.

Corrosion of the mill system was anticipated until experience with test specimens, placed in locations most favorable to corrosion, indicated that the flue-gas system would deteriorate equipment practically no sooner than an air-drying system. Thorough insulation of all piping and collectors apparently limits deterioration principally to erosion. No other maintenance problems of consequence have developed which are attributable to the use of flue gas.

Mill capacities have not been found different from those obtained using coal dried in separately fired driers. Practically the only troubles encountered have been in connection with feeding wet coal to the mills through supply hoppers and feeders.

Table 1 shows results of a test run on one of the mill-drying mills at Lakeside soon after the first installation was made, including data on moisture and heat balances and dust loss to the vent. The mill capacity found on this test is low and power consumption high, due to reduced mill speed for the trial and also coal that was unusually hard to grind. Outputs in the order of 15 to 18 tons per hr are more usual.

Later installations make use of a more efficient type of main cyclone, resulting in removal efficiencies of around 97 per cent instead of 90.5 per cent, so coal losses to the vent are correspondingly less.

Table 2 shows the principal installation and operating data for the mill-drying equipment installed to date by the Wisconsin Electric Power Company.

Fig. 3 shows a typical temperature recorder chart from a mill-drying installation.

ECONOMY OF SYSTEM

An accounting of heat gains and losses due to the mill-drying system of coal preparation shows the following:

	Per cent
Improvement in air-heater performance; 6 per cent gas extracted (10 per cent less gas through heater = 30 F lower out temperature) (37.5 F lower flue-gas temperature = 1 per cent boiler efficiency)	0.48
Reduction of 6 per cent of flue gas from 350 F to 150 F	0.32
Total credit to mill drying	0.80
Coal lost in vent	0.25
Net credit due to mill drying	0.55

This tabulation shows that the drying is effected more cheaply than it can be done in the furnace or with air drying, for the system actually improves boiler-room efficiencies by 0.5 per cent. Operating separately fired or steam driers requires about an equivalent heat expenditure, which indicates that mill drying is one per cent more efficient than the older drying processes.

The economies mentioned are all thermal economies. In addition, mill drying must be credited with large savings in building, equipment, maintenance, and operation costs. The process has practically eliminated the fire and explosion hazards in the milling of coal.

Steam Generation in Steel Mills

By H. J. KERR,[1] NEW YORK, N. Y.

The author compares operating results of blast-furnace power plants for the years 1922, 1931, and 1940, at the same time indicating the changes which have occurred since 1931, when F. G. Cutler (1)[2] ably presented the developments for the preceding decade. In the course of the paper, an analysis is given of the factors delaying the use of high steam pressures in steel-mill practice, and the problems to be solved in raising steam temperatures. Numerous blast-furnace boiler installations, typical of the best modern practice, are briefly described and illustrated.

IN APRIL, 1931, the late F. G. Cutler, an outstanding steam engineer of the steel industry, presented, from this platform, a paper entitled, "Design Features and Operating Results of Fairfield Blast Furnace Power Plant" (1).[3] In Table 2 of that paper he gave a digest of the results that had been obtained, and compared them with those of 1922. In this comparison, Mr. Cutler showed the improvement obtained by: The more efficient combustion of blast-furnace gas, the use of electrostatic precipitators for the cleaning of gas, the increase in steam pressure from 150 to 340 pounds, the combination of pulverized coal and blast-furnace gas in the same boiler furnace, and the increase in capacity of boiler units from 70,000 to 175,000 pounds of steam per hour.

Nine years have elapsed since the presentation of Mr. Cutler's paper and, as many changes have taken place in steam generation during this period, it may be of interest to compare present-day conditions with those of that time. Table 1 shows such a comparison for 1922, 1931, and 1940:

The first two columns of Table 1 constitute additions to, and abbreviations of, Table 2 (1) as given by Mr. Cutler. In making up the third column, some liberty has been exercised, by taking known facts from central-station practice and modifying these values by consideration of steel-mill problems.

Changes in Steam-Generating Practice 1931–1940

Considering in turn the various items in the list, the following changes taking place from 1931 to 1940 will be noted:

1 Steam pressure increased from 340 to 900 psi, with steam temperature increasing from 650 to 850 F.

2 Boiler efficiency has not been materially increased, due largely to economics, but has been maintained, notwithstanding the higher steam temperatures of today, by the use of air heaters and economizers. Stove efficiency about the same.

3 Capacity of boiler units has been greatly increased. This development, in the steel industry, has been held back by the dirtiness factor of blast-furnace gas, since large units do not lead to flexibility of operations, unless they can be maintained in service over reasonably long periods of time. It will be noted that this limitation does not exist at Fairfield, since clean blast-furnace gas has been in use for some time. Other steel plants are making real improvements in this direction.

4 Kilowatthours per ton of product (surplus) increased, due largely to increased efficiency of high-pressure steam units.

In regard to furnaces, two yardsticks are given in Table 1. The first is the capacity factor of "Btu per cubic foot of furnace volume." In the past this expression has been greatly misused. It is, however, a definite measure of the size of furnace in use for the combustion of a specific quantity of fuel, and should not be discarded any more than it should be misused.

The table shows the reduction in this value during the transition from stokers to pulverized fuel, between 1922 and 1931, and then the increase in value as water cooling in furnace practice developed, as shown for 1940. These figures indicate that furnace sizes have not been determined by the combustion rates possible with pulverized coal.

Perhaps the values of Btu per cubic foot, with blast-furnace gas, furnish further evidence of this general statement, since these values are often as high as with pulverized coal, and have been for many years, notwithstanding the lower heat value of this fuel.

The second measuring stick for furnaces, "Btu available per square foot of equivalent cold surface," is one which the author discussed in 1932, at Pittsburgh (2). This factor, in general, denotes the amount of cooling received by the gases of combustion before entering the convection bank, which, in turn, determines the amount of trouble to be expected from slag. With coal-firing, it is to be noted that high values were used in the past with stokers operating at capacities which permitted burning the coal on the stoker grate. As the practice shifted to pulverized coal, and not considering unsatisfactory installations, this value seemed to level off at approximately 200,000 Btu per sq ft of equivalent cold surface for coal having an ash-fusion temperature of the order of 2200 F.

In the case of the blast-furnace gas, very much higher values

TABLE 1 STEEL-MILL POWER PERFORMANCE IN 1922, 1931, AND 1940

	1922	1931	1940
Steam pressure, psi	150	340	900
Steam temperature, F	450	650	850
Boiler efficiency, per cent	66	83	83
Stove efficiency, per cent	60	70	70
Dirt in blast-furnace gas, grains per cu ft	3	0.5	0.05
Size of boiler units, lb steam per hr	60000	175000	400000
Electric current per ton of product (surplus), kw	25.3	41	53
Boiler-furnace factors			
Btu per cu ft of furnace volume			
Coal	35000	14000	40000
Blast-furnace gas	25000	20000	25000
Btu available per sq ft of equivalent cold surface			
Coal	300000	207000	200000
Blast-furnace gas	350000	200000	150000

TABLE 2 SUPERHEATER ABSORPTION

Throttle pressure, psi	200	400	800	1200
Steam temperature required, F	570	695	835	935
Gas temperature drop across superheater; no by-pass, F	288	435	636	795
Heat absorbed by superheater, per cent	9.6	14.5	21.2	26.5
Gas temperature drop across superheater; with regulation for constant superheat to 0.5 load, F	...	...	960	1200

[1] Executive Assistant, The Babcock & Wilcox Co. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

[3] Bibliography (1), Table 2, p. 17.

Contributed by the Fuels Division and presented at the Joint Meeting, Birmingham, Ala., November 7–9, 1940, of the Coal Division of the American Institute of Mining and Metallurgical Engineers and the Fuels Division of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

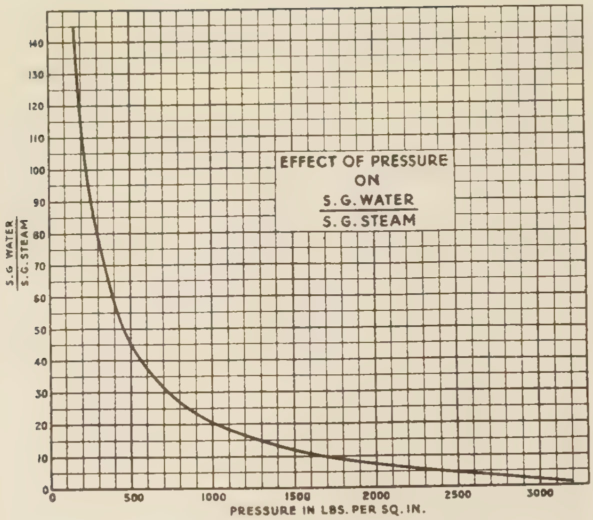

Fig. 1 Effect of Pressure on $\frac{\text{S. G. Water}}{\text{S. G. Steam}}$

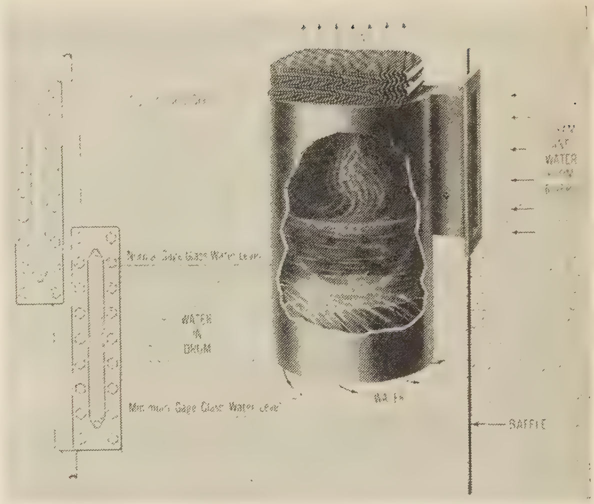

Fig. 3 Cutaway Section of Cyclone Separator in Boiler Drum

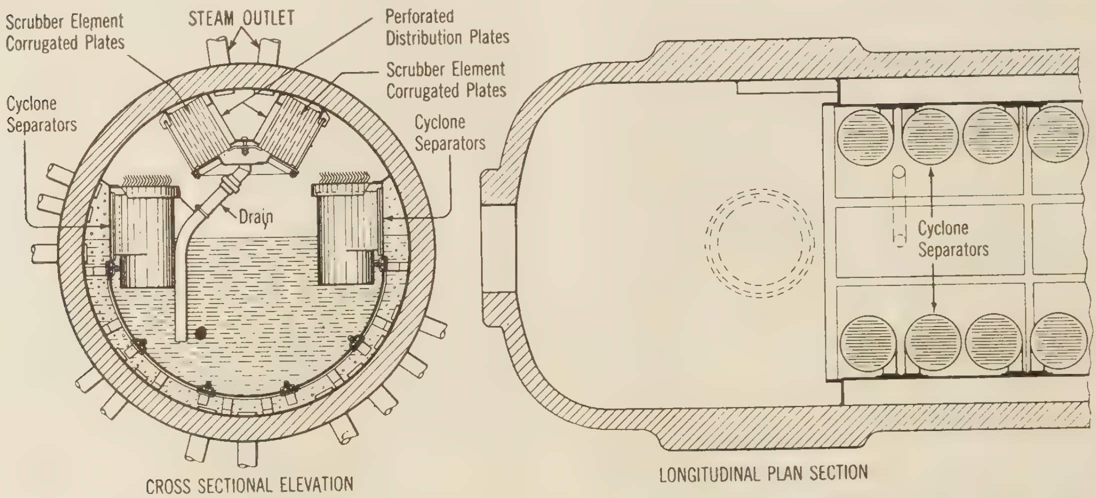

Fig. 2 Typical Installation of Cyclone Separators in Large Central-Station Boiler Drum

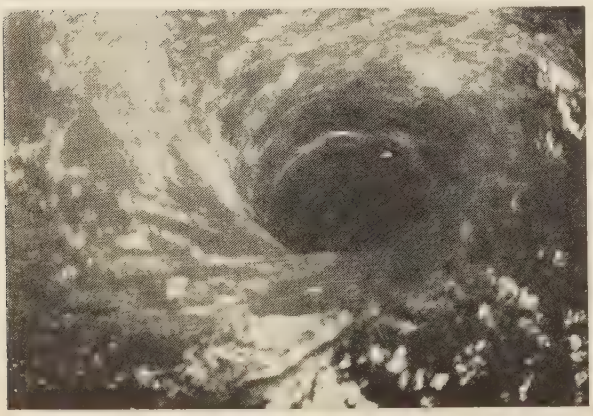

Fig. 4 Vortex in Downcomer of Experimental Boiler Drum

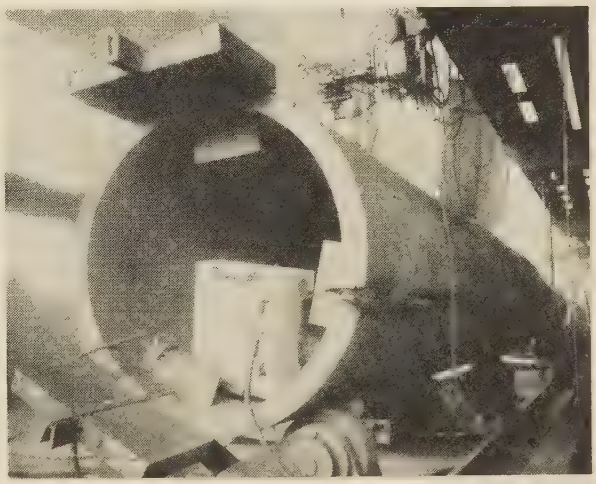

Fig. 5 Welded Drum for High-Pressure Boiler in Course of Construction

of Btu per square foot of cold surface have been used in the past, as will be noted from Table 1. This, of course, was possible because of the low combustion temperatures obtained with this fuel. With later installations, utilizing blast-furnace gas, this value has decreased, solely because pulverized coal is being burned in the same furnace and is the determining factor in furnace design.

Within the last few years, E. G. Bailey has been giving considerable attention to this factor, and has referred to it in a recent paper (3).

Some discussion of the principal factors presented in Table 1 may be of interest:

A steam pressure of 1500 psi has now been in use for a sufficient period in central stations as to leave no doubt of its commercial success. One manufacturer has been using units at 1500 psi in process work, for years; a second manufacturing plant has in service 2200 psi pressure in connection with power and process, and another central station is now installing a very large unit for 2600 psi with steam reheat. Up to the present, the operating pressure in steel plants in this country has not exceeded 900 psi.

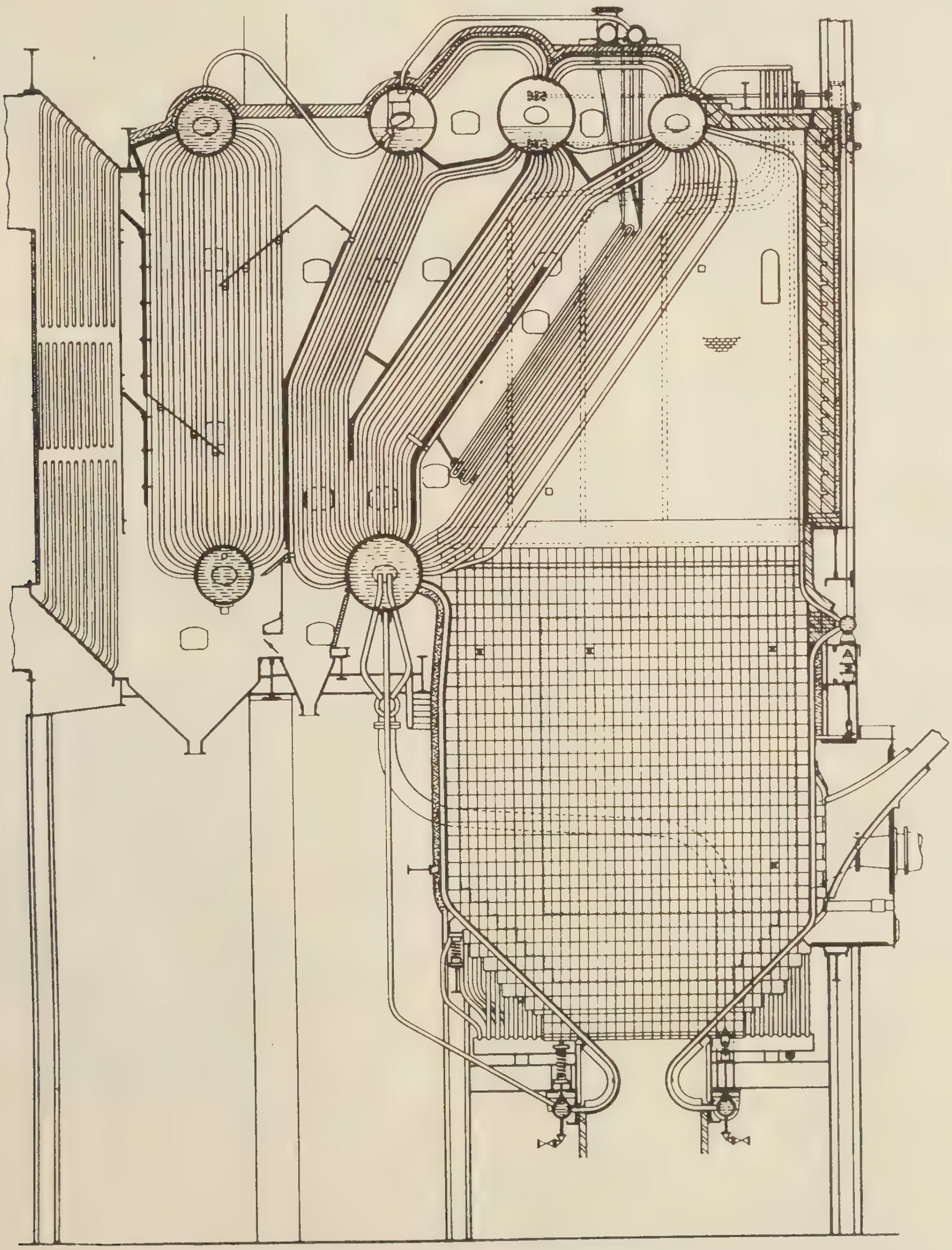

Fig. 6 Stirling Boiler Installed at Fairfield in 1932

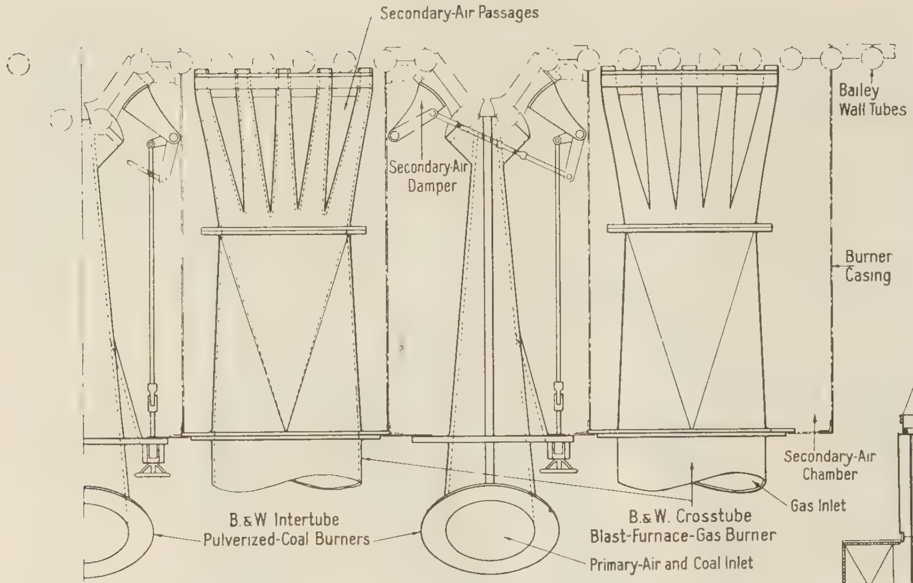

Fig. 7 (Left) Combination Burner for Pulverized Coal and Blast-Furnace Gas

Factors Delaying Use of High Pressures in Steel Mills

There are four factors which have caused some steel men to hesitate in using high-pressure steam:

The first is the matter of circulation. Many engineers believe there is a definite limit to the steam pressure permissible with natural circulation. Fortunately, the company, with which the author is connected, made a careful study of this problem about 10 years ago, which proved that natural circulation was entirely satisfactory with proper design for pressures up to at least 2600 psi.

This statement presupposes a definite separation of water and steam in the boiler drum, so that the water in the downtakes will be of normal density. As the steam pressure increases, Fig. 1, the steam and water densities approach each other, making separation of steam and water more difficult.

Not only is it necessary to remove the water from the steam but for proper circulation conditions, it is also necessary to remove practically all the steam from the water, so as to be able to maintain the desired circulation head in the downcomers. This has been satisfactorily accomplished by the present construction of cyclones followed by steam scrubbers, as shown in Figs. 2 and 3. In these cyclones, the force available for separating the steam and water may be 5 to 10 times the force of gravity. For more detailed information on this construction and results obtained, reference is made to a paper by M. D. Baker (4).

After the steam and water are separated in the drum, it yet remains for the designer to take care of such conditions as are shown by Fig. 4, which shows an experimental drum, looking down at the entrance to a downcomer. It will be noted that, if proper means are not taken to prevent its formation, a vortex of no mean proportions may result, in which case, the circulation head on a boiler unit may be tremendously decreased. It is perhaps not generally realized that, in a boiler unit furnishing 400,000 lb of steam per hr, there may be passing through the drum from 4,000,000 to 8,000,000 lb of water per hr.

The second factor which arises in connection with the use of high-pressure steam is that of feedwater. Central stations use condensate, whereas, steel plants generally use all make-up water. This is a very definite difference and one which requires every consideration. It may be stated that today feedwater can be quite satisfactorily treated for high-pressure steam operations, with the one question open, which chemists call by various names,

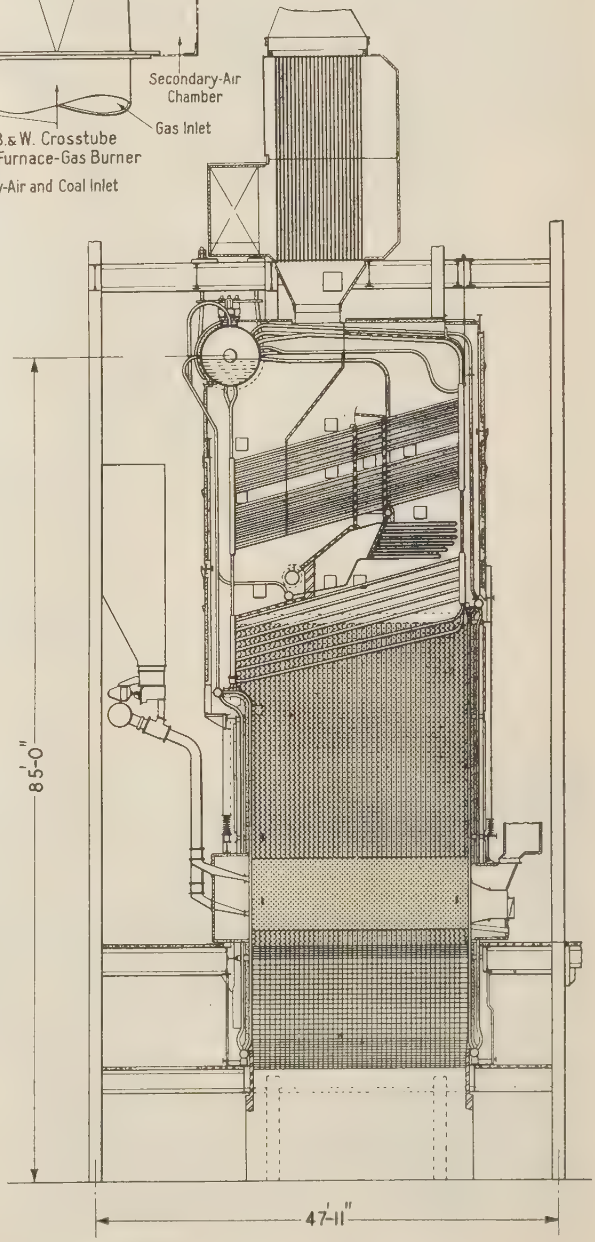

Fig. 8 Cross-Drum Boiler Installed at Eliza Furnace of Jones & Loughlin Steel Company

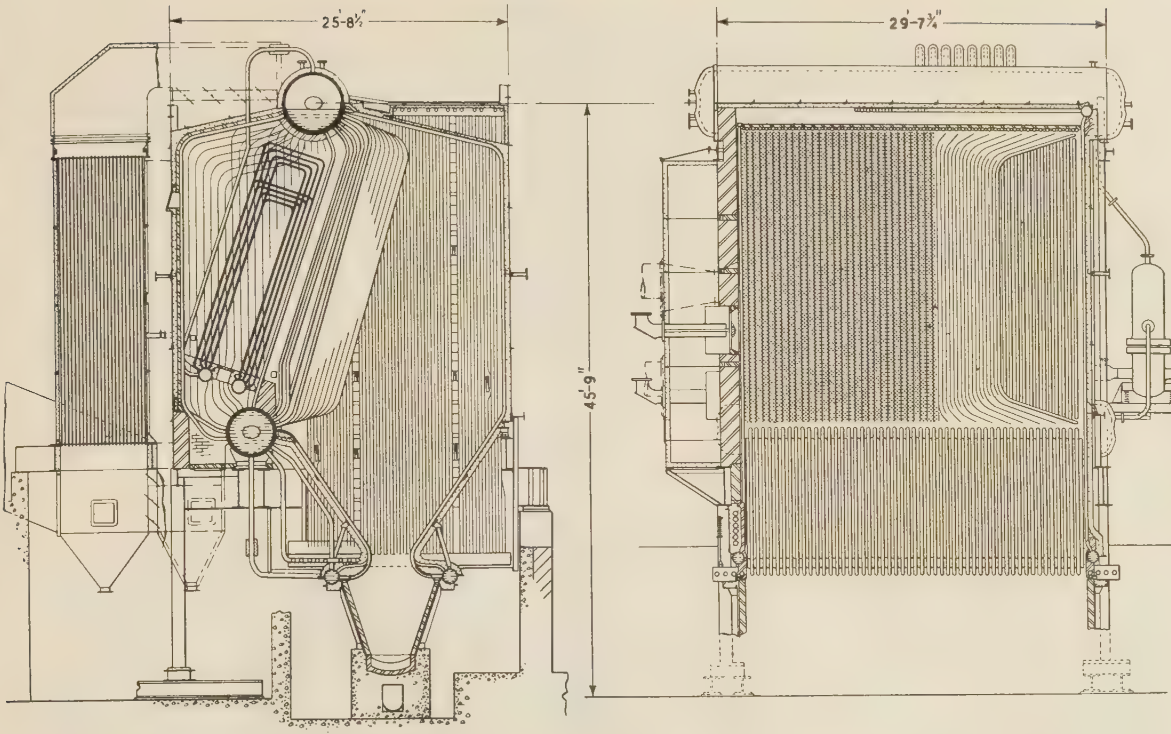

Fig. 10 Integral-Furnace Boiler for Firing With Pulverized Coal and Blast-Furnace Gas; Youngstown Sheet and Tube Company

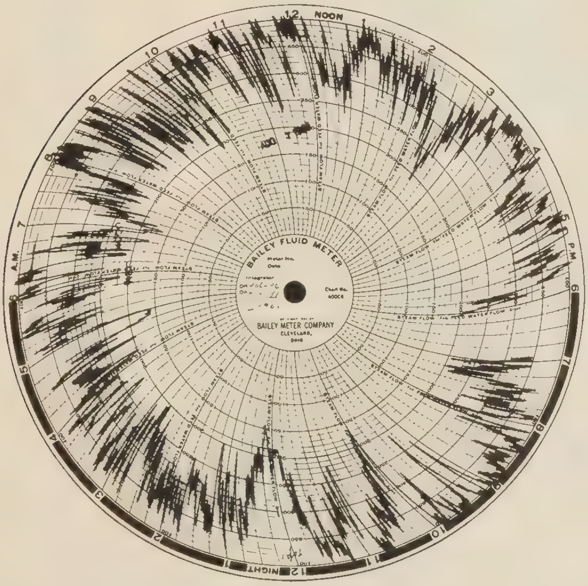

Fig. 9 Steam-Flow Chart From Boiler at Eliza Furnace

but which, fundamentally, is the problem of silica removal. Without going into details, it may be stated that the solubility of silica is reduced at higher temperatures, so that less concentration of silica is permissible at high pressures than at low. If, therefore, the water available has a natural silica content, scale may form at high pressures. Much work has been accomplished in silica removal, as evidenced by the results obtained at the Baton Rouge plant of Gulf States Utilities. These results were presented in a paper by M. C. Schwartz (5).

New and successful efforts are being made along other lines toward the elimination of silica. The Weirton Steel Company installation, operating at 850 psi pressure, using Ohio River water for feedwater, is evidence of successful operation at that pressure with practically 100 per cent make-up water. This installation is well described in a paper by H. G. Strassburger (6).

Priming, which might be accelerated by feedwater conditions, has been practically eliminated by the cyclones previously described. Where they are used, water at full density is maintained in the drum.

The third factor is one of mechanics. Fig. 5 shows a high-pressure welded drum in course of construction. There are no unsolved mechanical problems involved in meeting the requirements in equipment for using high-pressure steam. So long as steelmakers can make good steel of 70,000 psi tensile strength, boilermakers will make satisfactory boilers for any pressure which may be required. There is little difference, if any, in the reliability of the construction of a 1500-psi boiler and one built for 150 psi.

The fourth point which sometimes is cited in connection with high pressures is possible operating difficulties. In regard to this, the final paragraph of Mr. Cutler's paper (1) of 1931, is quoted:

"All of the operators were taken from other plants of the Tennessee Company, and they had very little if any experience with steam pressures over 150 psi, superheaters, turbines, condensers, pulverized coal, or automatic combustion control; however, the operating difficulties encountered have been but nominal."

Problems Encountered in Raising Steam Temperatures

The increase of steam temperature from 650 to 850 F in steel

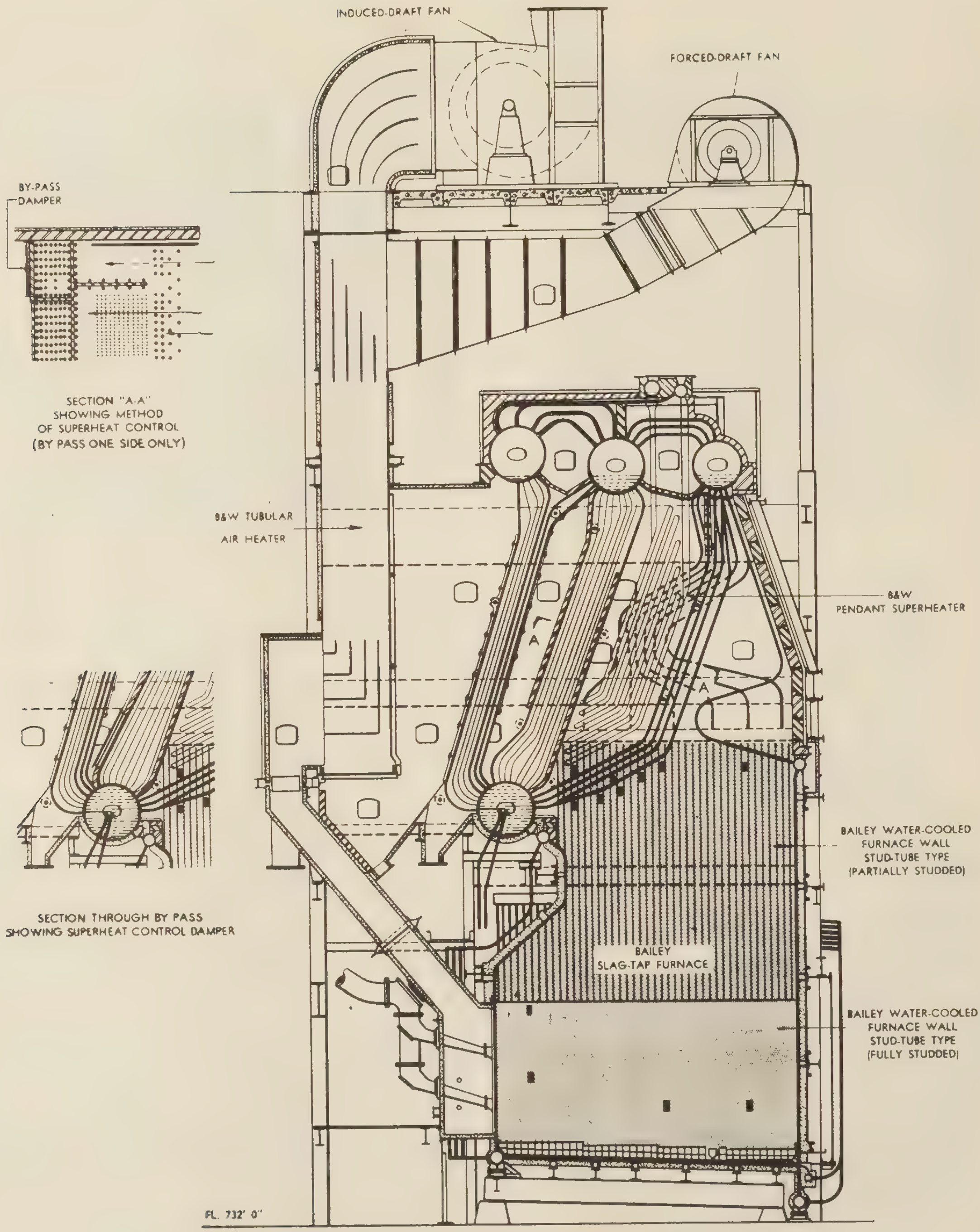

Fig. 11 Stirling Boiler With Slag-Tap Furnace; Weirton Steel Company

plants, and to 950 F in central stations, required the solution of three problems:

1 In selecting materials for superheated steam, carbon steels were unsatisfactory, due to oxidation and low creep strength. Partly as a result of the pioneer work done in the oil industry alloy metals have been developed, so that today eleven different chromium alloys are available for tubes, the chromium content ranging from 1 to 27 per cent. Therefore, materials are obtainable which are fully capable of meeting present-day requirements from the standpoint of both oxidation and strength.

2 As for tightness, the rolled joint has maintained its position with the increase in pressure, but has been found deficient with increase in temperature. Fortunately, welding was developed as required and, through its use, high-temperature construction gives less trouble from leakage than was common with constructions employed with lower temperatures. To permit the commercial application of welding to superheaters, material changes in their construction have been developed.

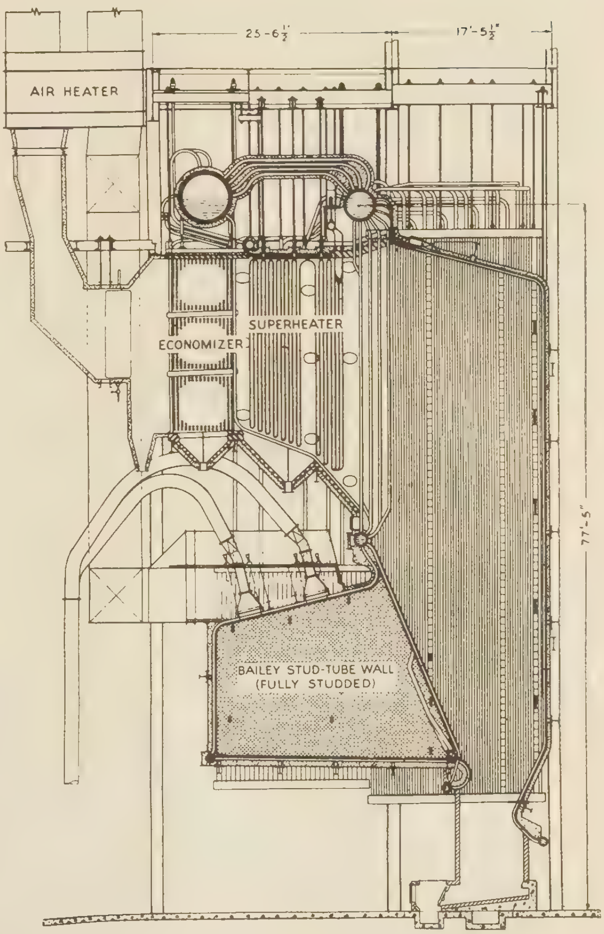

Fig. 13 Radiant Boiler in Service at Essex Generating Station, Public Service Electric & Gas Company, New Jersey

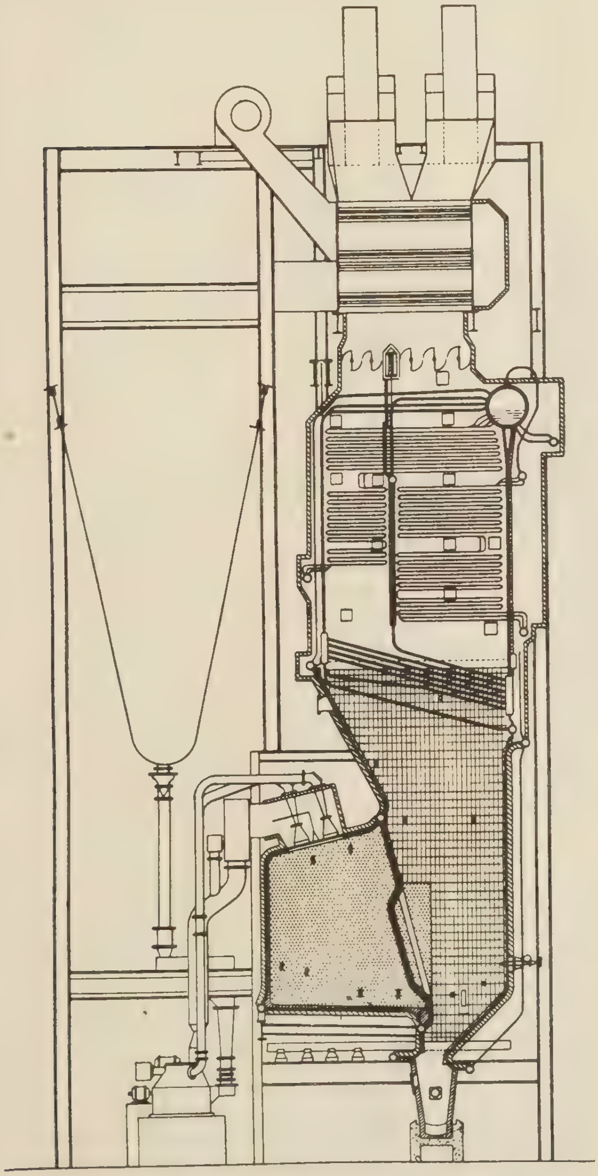

Fig. 12 High-Head Boilers in Service at Rivesville Station, Monongahela & West Penn Power Company

3 In conjunction with high pressure, Table 2, high steam temperature has changed the percentage of total heat required for superheat to such an extent as to have brought about new designs of boilers. The necessity of maintaining constant superheat over a considerable range in rating has accentuated this condition.

Present-Day Furnace Practice

With blast-furnace gas (particularly if clean and hot), few problems exist in present-day furnaces with properly designed burners, unless perhaps at low ratings with too cold a furnace. The reasons for this are that the flame temperature with this fuel is so low as to cause negligible slagging under any rates of combustion contemplated; and, since pulverized-fuel firing in the same furnace is required in steel mills, the limitations set by this latter fuel determine furnace design.

What, then, are the limitations of pulverized-fuel furnaces? As stated earlier, Btu per cubic foot is not the limitation, and for the reason that this expression means only the amount of fuel burned in a given space in a given time. The industry is still in the elementary class on this particular problem. This term has been used by many as an expression of limitation of furnace capacity; and, in some cases, this has been approximately correct, but only because other things happened to occur in proportion.

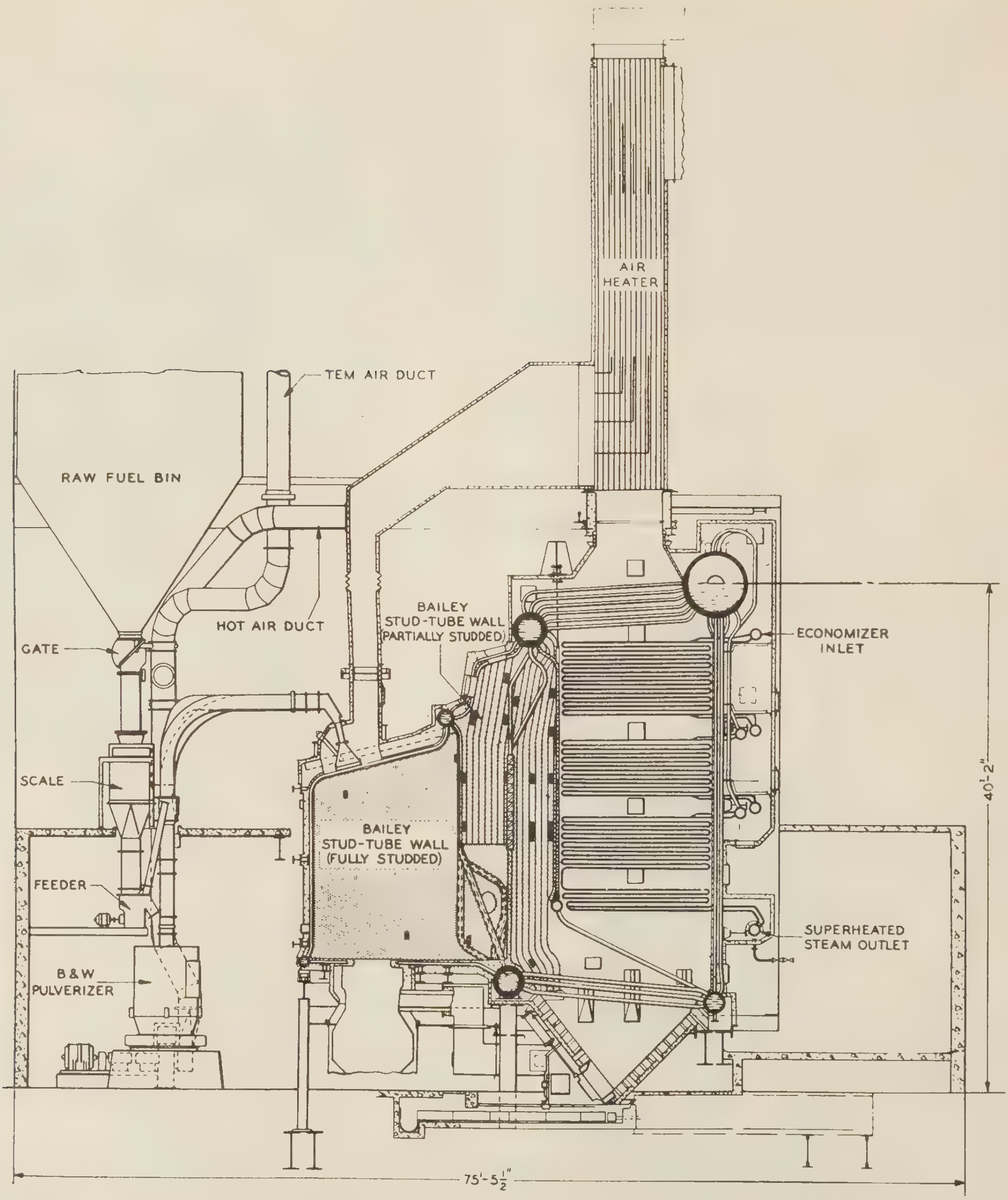

FIG. 14 OPEN-PASS BOILER IN SERVICE AT WEST END STATION, UNION GAS & ELECTRIC COMPANY, CINCINNATI

The real limitation of pulverized-fuel furnaces is slag—nothing else. Two methods are available to overcome this difficulty; (*a*) The elimination of the slag in the furnace gases or, (*b*) the reduction of the temperature of gases leaving the furnace.

While not accomplishing the first result completely, real progress has been made with slag-tap furnaces, from which 50 per cent of the ash in the coal may be removed in molten form.

The second method is to cool the gases to a temperature below the ash-fusion temperature before they leave the furnace. To do this requires a certain amount of cooling surface around the furnace and, hence, this result is measured by the expression, "Btu available per square foot of cooling surface." This sounds like a simple remedy, but it is perhaps not so simple. The boundary surface of a cube increases as the square of its dimension, while the volume increases as the cube; hence, with large-size units it becomes difficult to obtain sufficient surface without excessive volume. Again, if furnaces are made too cold, combustion rates decrease, and certain lighting and low-rating problems are encountered. Finally, with high superheat and high pressure, particularly with constant superheat maintained at low ratings, a definitely low limit of gas temperature exists, below which it is impractical commercially to obtain the superheat required.

In furnaces which have been installed by the author's company,

progress has been made by combining the two principles discussed, as will be noted from the illustrations of various units, representing boilers of comparatively recent construction.

Blast-Furnace Boiler Installations

Fig. 6 shows a four-drum Stirling boiler, installed at Fairfield in 1932. This unit was designed for 300,000 lb of steam per hr with coal, and 200,000 lb per hr with blast-furnace gas. It is similar to the units described in Mr. Cutler's paper (1), but has two additional features; i.e., complete water cooling of the furnace, and a combination of coal and blast-furnace-gas burners, as shown in Fig. 7. This system has proved quite successful in combination firing. The capacities mentioned for both fuels have been exceeded.

Fig. 8 shows a unit installed at the same time as the new strip mill at the Eliza furnace of the Jones & Loughlin Steel Company, for a maximum capacity of 400,000 lb of steam per hr at 500 psi pressure. Fig. 8 is shown particularly for two reasons; i.e., it illustrates a gas by-pass around the superheater for the purpose of obtaining constant superheat at various ratings; and because this unit exemplifies what can be done to handle successfully the extremely variable load so often met with in steel-plant operations.

Fig. 9 is a steam chart from the unit, Fig. 8; variations in steam flow from 200,000 to 400,000 lb per hr will be noted.

This unit was installed in the same powerhouse as four existing boilers which were carrying as much of the swing in load as they could successfully handle. The new unit was required to accommodate the type of swings shown by the chart, Fig. 9. This was accomplished by installing, in the steam line of this boiler, a butterfly valve which opens wide as the load demand increases, thus obtaining the required steam flow which, in many cases, increases at too fast a rate to be obtained by change in fuel rate.

Fig. 10 is a sectional view of an integral furnace boiler unit in course of construction for the Youngstown Sheet & Tube Company, Chicago. Two such units have been in service for some time using pulverized fuel alone. This unit is to be fired with both pulverized coal and blast-furnace gas, at a maximum capacity of 170,000 lb of steam per hr and a steam pressure of 825 psi.

Fig. 11 shows a side view of the units at the Weirton Steel Company, which were described by J. H. Strassburger (6). This is a typical application of a slag-tap furnace to a four-drum Stirling boiler. These units, operating at 850 psi, 850 F steam temperature, and at a maximum capacity of 400,000 lb per hr, are showing highly reliable operating results. Another unit of the same type now being installed will combine both blast-furnace-gas and pulverized-coal firing.

Fig. 12 illustrates two units in service at the Rivesville Station of the Monongahela & West Penn Power Company. These units were designed for 350,000 lb of steam per hr at 1250 psi, and have operated at materially greater capacities. As shown by the illustration, the furnace is of the two-stage slag-tap type, designed for burning coal containing ash of 2100 F fusion temperature. The heat release at full load in the entire furnace is 40,000 Btu per cu ft. The heat release in Btu per sq ft of equivalent cold surface is approximately 150,000.

Fig. 13 shows a side elevation of the radiant-heat boilers operating at the Essex Station of the Public Service Company of New Jersey. They are designed for a pressure of 1475 psi and a steam temperature of 950 F, with a capacity of 600,000 lb of steam per hr. These units also have two-stage slag-tap furnaces. This design permits the combination of a high-temperature primary furnace, wherein are obtained high rates of combustion per cubic foot and a high value of heat available per square foot

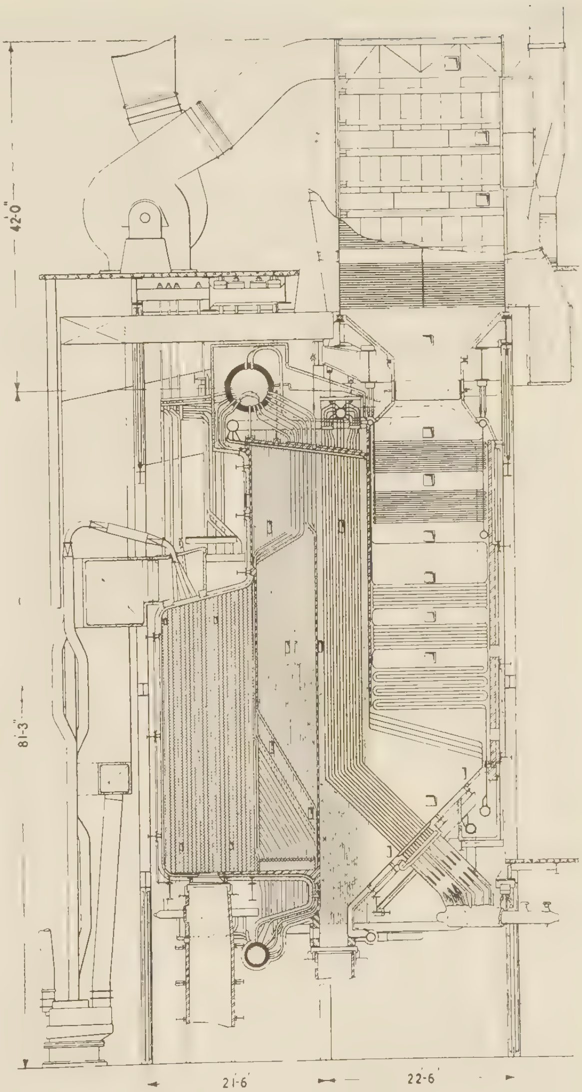

Fig. 15 Open-Pass Boiler to Operate at 2600 Psi Pressure; Twin Branch Station, American Gas & Electric Company

of surface, and a secondary furnace, furnishing a high cooling factor for the gases, which have already shed a considerable percentage of the ash in the fuel.

Fig. 14 is a side elevation of one of the units in operation at the West End Station of the Union Gas & Electric Company, Cincinnati. They were designed to fit into an existing building, and have a maximum steam production of 350,000 lb per hr at 1275 psi pressure and 910 F steam temperature. The primary furnace is of the same general type as that shown in previous illustrations and from which slag is tapped continuously through the center of the floor into a sluice tank. The release in the primary furnace is of the order of 82,000 Btu per cu ft.

To the rear of the primary furnace are two gas passages surrounded by partially studded tubes, providing for a gas travel of about 70 ft between the burners and the heating surface of the superheater and economizer. With this construction, a further step was made in the effort to obtain high furnace temperature for high combustion efficiency and the removal of a greater percentage of ash in the primary furnace. By the com-

plete shielding of radiation from the primary furnace to the heating surface of the boiler proper, and by increasing the convection-transfer rate of the boundary tubes in the latter stages of the furnace, a lower temperature leaving the furnace is obtained. With this arrangement, the combustion rate up to the convection surface is approximately 40,000 Btu per cu ft. The Btu per sq ft of cold surface is 110,000.

Fig. 15 is a side elevation of the boiler now being installed at the Twin Branch Station of the American Gas & Electric Company, to operate at 2600 psi, 940 F steam temperature, and at a capacity of 550,000 lb of steam per hr. It is to be noted that the general form of this unit, which is for operation at the highest pressure for which natural-circulation boilers have been built, follows the same general design as the units which are shown in Fig. 14.

BIBLIOGRAPHY

1 "Design Features and Operating Results of Fairfield Blast-Furnace Power Plant," by F. G. Cutler, Trans. A.S.M.E., vol. 54, 1932, paper FSP-54-2, pp. 13–21.

2 "Water-Cooled Furnaces, Steam Boilers," by H. J. Kerr and R. M. Hardgrove, *Iron and Steel Engineer*, vol. 9, 1932, pp. 375–394.

3 "Steam-Boiler Performance and a Method of Comparison," by E. G. Bailey, Trans. A.S.M.E., vol. 62, July, 1940, pp. 367–375.

4 "Mechanical Purification of Steam Within the Boiler Drum," by M. D. Baker, Trans. A.S.M.E., vol. 61, 1939, pp. 711–718.

5 "Removal of Silica From Water for Boiler Feed Purposes," by M. C. Schwartz, *Journal of the American Water Works Association*, vol. 30, 1938, pp. 659–678.

6 "800-Pound Pressure, 10,000-Kw Topping-Turbine-Generator Installation at Weirton Steel Company," by J. H. Strassburger, presented at the Pittsburgh, October, 1938, meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS and the Engineers' Society of Western Pennsylvania.

Discussion

F. E. NORTON.[4] In the following discussion it will be assumed that the steel mill includes blast furnaces and other sources of so-called waste fuels.

The paper illustrates eight boilers of which four are suitable for general steel-mill practice, utilizing coal or gas fuel, or a combination of fuels. The other four types are not, in general, suited to gas fuel or combined fuels.

In the paper, considerable emphasis is placed on the rate of heat release in boiler furnaces as it affects slag formation and consequent loss of efficiency. The author states: "The real limitation of pulverized-fuel furnaces is slag—nothing else." The writer agrees with this statement but would add that, for blast-furnace gas, the limitation is "mud" from the flue dust and ash in wet-washed gas.

The furnace gas may bring 0.03 to 0.05 grain of dirt per cu ft into the boiler furnace, but the effect on operation and efficiency depends upon whether or not the gas is wet or dry and upon the nature of the dust. The real criterion is not the Btu realized by the combustion of wet or dry gas, but lies in the effective use of the heat in making steam (i.e., dirty boilers versus clean boilers).

There is disagreement among operators of blast-furnace plants as to the relative merits of dry or wet gas in boilers. The disagreement may be due to the reaction of the blast-furnace department as to stove gas. Modern stoves require clean gas, and wet primary washing is the rule, while secondary wet electric precipitators are coming into general use for stove gas.

It is quite usual to wet-wash all the furnace gas and to divert a portion through precipitators for the stoves. This leaves the boilerhouse with wet gas, which quite often is very dirty and worse than the dry gas. The resulting effect on boiler steam production is very bad, especially if there are shots of dry dirty gas, as is often the case. The presence of any coal ash makes the condition yet worse. The mud deposits on boiler tubes and burners and soot blowers are incapable of removing it. Since the blast furnace, quite often, is given credit for the full coal equivalent of the gas, boiler troubles do not directly affect the furnace costs; indirectly, however, steam costs go up for blowing the furnace.

A further discussion of waste-gas fuel for boilers will bring out some essential features of boiler design and seems justified in view of the unsatisfactory performance of a number of plants using partially cleaned wet gas.

The use of dry dirty gas at stoves is not advisable but such gas may be used at the boilerhouse.

It is quite apparent that a satisfactory gas supply at stoves may not assure good results at boilers.

Of course, the obvious procedure would be to clean all the furnace gas with precipitators, at a very considerable investment cost. The dry gas would be cheaper. The alternative is to use dry dirty gas at boilers and wet-washed and precipitated gas at stoves.

An example of the use of dry dirty gas in two boilers, each rated at 165,000 lb of steam per hr, appears in a current article.[5] A third boiler is being installed at the Sydney (Nova Scotia) plant of the Dominion Steel and Coal Company. An efficiency of 80 per cent for the gas is reported. The performance of the first two boilers has been satisfactory. The writer's remarks are prompted by a comparison of these with similar boilers using wet-washed gas which are not at all satisfactory because of dirty gas.

The difficulties experienced with mud have several aspects. It is not likely that the operators will agree when experience at several plants is compared. For instance, the writer is familiar with a boiler plant in which dry fly ash from coke breeze is used to remove the deposit on tubes. The boiler output is satisfactory using wet gas.

Pulverized coal may or may not be effective in removing such deposits. If the coal is coarsely pulverized, and partly burned, the result may be a tar, flue-dust, coal-ash compound, which looks like slag and is difficult to remove from the boiler. Circumstances alter the effect.

The difficulty with wet gas is that the mud deposited from a rush of dirty gas will remain on the tubes for a long time. The gas may be perfectly clean for hours, but a few minutes of dirty gas will cut down boiler output for days. This aspect of the dirt problem leads to the statement that dry dirty gas is to be preferred to wet gas with more than say 0.05 grain of dirt per cu ft. This requires virtually perfect washing and precipitation; furnace slips or accidental gusts of dirty gas must be eliminated.

If dry dirty gas is to be used, it is essential that the boiler and furnace be arranged so that soot blowers can clean the tubes effectively and the dust can be removed from the furnace without loss of service or undue labor.

It is surprising that so few modern boiler plants are arranged to use dry gas. Boiler manufacturers should properly be interested in dust removal by mechanical means. The amount of money involved may be very large as illustrated by the following example which might hold for a plant with four blast furnaces.

Suppose a total of 1500 tons per day of coal equivalent is to be used in a boiler; say, 1000 tons by furnace gas and 500 tons by actual coal. The efficiency of the boilers with wet-washed dirty gas may fall to 60 per cent. With clean gas the efficiency may be 80 per cent. The amount of steam made by the clean gas would be $^1/_3$ greater than with the same amount of dirty gas.

[4] Special Engineer, Youngstown Sheet & Tube Company, Youngstown, Ohio

[5] "Hot-Blast Gas for Dominion Steel," by W. S. Wilson, *Power*, vol. 84, 1940 p. 733

The dirty boilers cannot give good results on coal and the efficiency of the entire plant would be lowered. The loss would all come on coal and would amount to 375 tons of coal per day for the same steam output.

With coal at $2.50 per ton, this amounts to $937.50 per day. Should the efficiency of dirty boilers be 65 per cent and the fuel consumption remain at 1500 tons per day, the loss would be 281 tons at $2.50 or say $702.50, as compared with clean boilers for the same amount of fuel. These figures are assumed from actual experience but are not taken from any actual records and do not represent actual operating data.

In such a plant, the loss of $700 per day amounts to $21,000 per month, as a result of using wet dirty gas. Per year, the loss may be $250,000. For this reason, the boiler design should contemplate the use of dry gas, unless capital is available for precipitators.

In view of such facts, it is difficult to understand why so few plants use dry dirty gas at boilers and even more so why plants use a mixture of dirty wet gas and dry gas. The boilers usually get the blame for poor efficiency and, in some cases, large investments have been made for economizers, air heaters, and other expensive devices to raise the efficiency, while the dirty gas wipes out all gain and adds the cost of primary washing.

In some cases the loss by poor efficiency may result in an increase of purchased power. If 1400 kw can be generated per ton of coal, which if purchased may cost 1cent per kwhr, the equivalent coal cost would be $14 per ton. The actual figure may be as low as $7 per ton of coal, depending upon circumstances and upon power schedules.

It will be noted that boilers for dry dirty gas must be arranged for easy removal of dust. The proper arrangement of gas and coal burners is also of great importance and may have a bearing on the flue-dust problem. The air for combustion of furnace gas can be kept in contact with fuel until combustion is complete and 20 to 22 per cent of CO_2 may be reached in the burned gas.

On the other hand, the coal does not readily carry the air for combustion along with the flame. The products of coal combustion may be 14 to 18 per cent CO_2 and, if coal is discharged into an atmosphere of burned furnace gas, the combustion may be delayed or even stopped by the 22 per cent CO_2 atmosphere.

In this connection, Fig. 6 of the paper (Fairfield boiler) is of interest, as showing the coal burner pointed down. In this case the coal flame should separate from the gas flame. The boilers at Sydney, Nova Scotia, have similar coal burners.

The writer recalls an early application of pulverized coal to waste-gas boilers at another plant in Birmingham which gave great difficulty in starting the coal burners. These were placed in the roof of the furnace so that the flame was compelled to mix with burned furnace gas. The result was a very dirty furnace.

Fig. 10 of the paper shows parallel coal-and-gas firing with intertube nonturbulent gas burners. This boiler is not designed for normal use of coal. It would be well suited to the use of unwashed dry gas. The use of a desuperheater in the boiler circulation system is also to be noted.

The operation of waste-furnace-gas boilers apparently is a simple problem and casual inspection of such a plant would seem to confirm this: With perfectly clean gas and a steady load, corresponding to the amount of steam the gas can make, it is simple so long as the boilers are clean.

When a widely varying load must be carried with a constant supply of gas, it is evident that the fluctuations must be carried on coal or other fuel. The problem of regulation becomes serious if combined coal-and-gas firing is used. The cases the writer has in mind involve a swing of say 30 to 60 per cent of maximum capacity of the coal-fired boiler; the period of swing may be $1^1/_2$ min or less. It will be realized that rapid fuel and draft control must be provided to meet the load conditions. An airtight furnace is demanded, since a leaky furnace means excessive loss at the stack, unless the fans and dampers give balanced furnace pressure. For this reason the coal should be burned on boilers which are adapted to coal and widely swinging loads. The base load could be carried by gas in order to obtain full output from the boiler.

A comparison of designs in use leads to the conclusion that practice has not been crystallized into standards which might be expected in view of the large number of installations and the amount of money involved in fuel costs.

The $250,000 differential mentioned would seem to warrant careful consideration of the conditions at the boilers when the gas-supply system is being laid out. The requirements of the stoves should not be the sole factor in final design.

J. H. Strassburger.[6] Since July, 1936, The Weirton Steel Company has been operating two boilers, each having a capacity of 400,000 lb of steam per hr at 850 psi working pressure and 820 F total temperature. These boilers have been supplying steam to a 10,000-kw topping-turbine generator as well as steam through a reducing plant for general power and process use. Operating experience with this boiler equipment has been highly satisfactory. The actual handling of boilers at this pressure has been no more difficult for our personnel than the handling of boilers at lower pressures.

During the period mentioned, these boilers have been supplied with feedwater conditioned by the hot-process system, utilizing 100 per cent raw water to the softening plant. During the first 2 years of operation, these boilers had an availability of 89 per cent, during the next $1^1/_2$ years the availability averaged 91 per cent and, during the year 1940, these boilers will have had an availability of 96 per cent. During the last 4 years every shutdown on each of these boilers has been a scheduled one for the purpose of periodic inspection and cleanup.

We are now completing the installation of a third boiler and a second generator. The third boiler is for the purpose of replacing low-pressure equipment which has completed its useful life. From our experience with this installation, we would not hesitate to operate one generator on one boiler.

We have experienced no difficulty in the operation of the topping generator. The 10,000-kw machine is loaded so that month in and month out it produces approximately 11,000 kw for every hour of the month, with peaks up to 13,000 kw depending upon back-pressure conditions. The availability of this generator, based on an inspection every 2 years and on washing out periodically every 3 months, amounts to 98 per cent.

We believe that in the next few years an extended use of high-pressure boilers with companion generating equipment will be widely adopted by the steel industry, and should create large savings with no serious operating difficulties.

E. J. Kohn.[7] Mr. Kerr has presented an interesting paper which has a particular appeal to the writer because he sets forth the advantages of high-pressure steam as applied to a local plant.

Referring to the unit "kw per ton of product from surplus blast-furnace gas" as shown in Mr. Kerr's article, Table 1, the writer refers to Mr. Cutler's article,[8] which presents an over-all comparison typical of 1922 and 1930, including blast furnaces, blast-furnace stoves, boilers, and steam users. The net result is an increase for the 1930 period in the steam available for electric generation.

[6] Combustion Engineer, The Weirton Steel Co., Weirton, W. Va.

[7] Chief, Bureau of Steam Engineering, Tennessee Coal, Iron and Railroad Company, Ensley, Ala. Mem. A.S.M.E.

[8] See reference 1 of Bibliography to paper.

Extending this comparison, Mr. Kerr justly emphasizes the advantages resulting from more efficient boiler design and operations, but fails to take into account the improvements incident to blast-furnace operation.

When one considers the improvements of blast-furnace practice, namely, ore conditioning, sintering, and possible air conditioning of blast, a lower coke rate per ton of iron would be in effect, which in turn would give a lower "heat in top gas." The net result would be less fuel per ton of iron available to the boilers which would materially lower the "kw per ton of product." Mr. Kerr's subject "Steam Generation in Steel Mills" would show up adversely if he considers improvements in blast-furnace practice and adheres to the unit "kw per ton of product from surplus gas."

For comparative results, conforming to the subject, Mr. Kerr should limit his comparison to the boilers and assume the heat to be the same in all cases. Mr. Kerr gives four reasons why steel men hesitate in the use of high-pressure steam: (1) circulation, (2) feedwater, (3) mechanics, (4) possible operating difficulties. As pointed out by Mr. Kerr, these problems have been overcome so we must look further for a possible reason for the hesitation.

Under feedwater, Mr. Kerr calls attention to the fact that central stations use condensate whereas steel plants generally use make-up water for boiler feed. From the latter one would infer that the steam was being used for mill operation. Relatively low-pressure steam, 200 lb and under, is required in the ordinary steel-plant operation, which of course, precludes the economical generation of steam at the relatively high-pressure range.

On the other hand if the steam is used for turboblower operation and electric-power generation, any refinement that could be economically applied to central stations could be applied in the same measure to steel plants. As evidence Mr. Kerr calls attention to the use of steam up to 900 lb in some steel plants. The entire question is one of economic balance between the cost of equipment installed as compared with the operating expense. Locally the relatively low cost of fuel would be largely a determining factor.

In brief the steel plants like the central stations will take the advantages to be derived from high steam pressure when it is economically possible to do so.

Author's Closure

The author appreciates the discussions, as they have added value to the paper.

Mr. Kohn has brought out the effect of improvements in blast-furnace operation and the cost of fuel on the general economic problem of steam generation in steel mills. It would be very difficult to cover the many variations in blast-furnace operation in a paper of this sort and, therefore, the author took as a base the blast-furnace data as furnished in Mr. Cutler's paper of 1932.

Undoubtedly the cost of fuel will appreciably affect the steam pressure and temperature conditions for most economical operation at any plant.

Mr. Strassburger has been good enough to give the actual experience which he has had over the last four years and which demonstrates the reliability of 850 lb pressure, 850 deg steam operation in steel-mill practice with 100 per cent make-up water.

Mr. Norton's discussion is a timely one as it brings out the problem which many of us have had to face when burning washed blast-furnace gas, namely, the tenacious scale formation on the boiler tubes. The steady increase in stack temperature when burning washed gas has made it necessary periodically to wash boilers on the outside as soot blowers will not remove this type of scale. Some experience indicates that periodic burning of pulverized coal in the same unit minimizes this trouble, as apparently the fine ash in the flue gas has sufficient abrasive effect to cut down this scale. With dirty gas the problem is the handling of large quantities of dirt in the unit and induced-draft fans. However, with dirty gas the stack temperature does not materially increase with reasonable use of soot blowers.

As indicated in the paper, the proper cleaning of blast-furnace gas results in long periods of operation of steam-generating units at maximum efficiency, and hence the desirability of such preparation of the gas where economically possible.

Coal Resources of Washington

By JOSEPH DANIELS,[1] SEATTLE, WASH.

Coal is the principal mineral fuel resource of Washington, overshadowed at the present time by the water-power developments at Bonneville and Grand Coulee. No petroleum is produced commercially, but there is a small production of natural gas. Wood, forest and mill, obtained largely as waste products of the lumber industry, is used extensively for heat, power, and industrial purposes. This paper discusses the location of coal regions, the characteristics of Washington coals, mining operations, transportation, cost of coal, and other factors, giving a comprehensive idea of the production and use of this fuel in the Pacific Northwest.

To MANY engineers the State of Washington is best known for its water power. They have heard of its scenery, its timber, excellent harbors, fishing, and possibly something of its metallic- and industrial-mineral resources. The fuel resources are not so well known or understood. Facts concerning these resources, with particular reference to Washington coals, will, therefore, be presented in this paper

Fuels Other Than Coal

In the State of Washington, no petroleum has been produced commercially, although much prospecting, mainly of the wildcat variety, has been conducted in various parts of the state. However, to date the results have been negative. A pipe line into Spokane from Montana and tanker transportation from California ports to tidewater points deliver the various fuel oils required to strategic distributing centers. California fuel-oil shipments into Washington averaged slightly over 9,000,000 bbl annually from 1930 to 1938.

A small production of natural gas, predominantly methane, of approximately 900-Btu value is obtained from a series of shallow wells in the Rattlesnake Hills field of Benton County, 18 miles northeast of Prosser. Distribution began in 1929; the gas is sold in seven cities in the Yakima Valley. The reported production in 1938 was 121,000,000 cu ft, of which 111,000,000 were sold. The natural gas has been mixed with butane-air gas before distribution. It does not appear that natural gas from outside sources can be economically piped and distributed through the state, although the matter has been discussed and some plans were once projected for lines from out-of-state fields. Brief mention should be made here that manufactured and liquefied petroleum gases are available to consumers in 29 cities other than those served by natural gas.

Forest wood and waste in the form of millwood, slabs, edgings, sawdust, and hogged fuel must be listed in any inventory of fuel resources. The exact volume or tonnage used is not known, but it plays an important part in supplying heat and power in domestic and industrial uses everywhere throughout the state. The availability and relatively low price make wood, particularly sawdust and hogged fuel, a very attractive fuel to many consumers, especially in western Washington.

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Contributed by the Fuels Division and presented at the Fall Meeting, Spokane, Wash., September 3–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

Location of Coal Regions

Topographically the state can be partitioned into five divisions, represented by the Olympic Mountains or Coast Range on the west, the Puget Sound Basin, the Cascade Range, the Okanogan Highlands of northeastern Washington, and the Columbia Plateau area, making up the greater portion of the eastern and southeastern part of the state. The economically significant coal region lies along the western flank of the Cascades from the Canadian boundary on the north to the Columbia River on the south in Whatcom, Skagit, Snohomish, King, Pierce, Thurston, Lewis, and Cowlitz Counties, and on the eastern slope in a limited, although important, outlier in Kittitas County. This major coal region is served by four railroad systems, is close to water transportation, and is tributary to an excellent system of roads and highways which permits truck delivery to practically every part of the state. Most of the deposits lie within a radius of 100 miles of Seattle.

The United States Geological Survey, using certain measuring sticks, estimated the content of coal in 1917, and again in 1925, and reported a total of 63,877,000,000 tons of coal. Similar estimates were made for other portions of the country. For purposes of comparison with other districts on the Pacific Coast, a summary of the resources of Alaska, British Columbia, Oregon, and California is presented in Table 1.

TABLE 1 SUMMARY OF ESTIMATED RESOURCES OF COAL ON PACIFIC COAST

(Millions of tons)

Geographical division and rank of coal	Actual reserve	Probable reserve
Alaska[a]	Metric tons	
Anthracite and semianthracite	1931	.....
Semibituminous and bituminous	1369	.....
Subbituminous	3681	.....
Lignite	12612	.....
Total	19593	.....
British Columbia[a]	Metric tons	Metric tons
Anthracite and semianthracite	7	1343
Bituminous	23764	43925
Subbituminous or lignite	60	5136 1800
Total	23831	52204
Washington[b]	Net tons	
Anthracite and semianthracite	23	...
Bituminous	11412	....
Subbituminous	52442	....
Total	63877	.....
Oregon[b]	Net tons	
Bituminous	3000	.....
Subbituminous	7000	.....
Total	10000	.....
California[b]	Net tons	
Bituminous	27	.....
Subbituminous	16	.....
Total	43	.....

[a] Refer to Bibliography (1).[2]
[b] Refer to Bibliography (2).

All of these estimates have been questioned because they were based on interpretations of broad geological structures, in some cases on inadequate data, and because they did not consider the limitations imposed by operating and economic factors which govern the workability of deposits. In a broad way, however,

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

since the same yardstick was used everywhere, the estimates give a relative or comparative picture of the magnitude of the resources. The figures appear to indicate that Washington originally contained the largest resource of tonnage in the Pacific Coast strip. Using actual reserve tonnages as a basis, Washington had 56.4 per cent of the total, British Columbia 21 per cent, Alaska 13.7 per cent, Oregon 8.8 per cent, and California 0.1 per cent. The significant position of Washington with respect to future supplies of solid fuel is indicated by this analysis. The figures quoted, however, cannot be used for estimating present commercial reserves or probable life of the various fields or districts within the major areas.

Coal Mining in Washington

Coal mining began in the state about 1850; shipments to adjacent territory were reported in 1860. No records of production are available before 1860, but since that date figures of production are well known. Production to the end of 1938 was 126,677,000 tons. Maximum output began shortly after 1900 and was maintained over a period of years until 1920. The production in 1910 was 3,911,899 tons; in 1915, it was 4,082,212 tons; the low point occurred in 1934, when only 1,382,991 tons were mined. Annual tonnages have been relatively small; the decline has been due mainly to slow growth of population and industry, increased efficiency of utilization, growing competition with liquid and gaseous fuel, and the development of hydroelectricity as a substitute for heat and power formerly supplied by coal.

Geologically, the coals of Washington are relatively young, that is, they are found in the Eocene formations of the Tertiary period. However, one important fact must be noted, i.e., dynamochemical forces were active during later parts of recent geologic time with the result that all ranks of coal from lignite to anthracite were produced in the state. A distinct relationship exists between the degree of metamorphism of the Eocene sedimentaries and the rank of coal; thus, for example, the higher-rank coals are found near the Cascade Range and, away from this area of deformation, the coals show complete gradations to lignites. This relationship of rank and structure has been an important factor in the development and operation of mines and in the preparation of coal for market.

The principal resource is subbituminous in rank; bituminous, both coking and free-burning, constitutes the next major proportion; lignitic and anthracitic coals a relatively small proportion of the total. The coking coals constitute an important asset in connection with their utilization for metallurgical uses. The greater part of the production, 77 per cent, has been bituminous coal; subbituminous is second with 23 per cent; lignitic coals have been insignificant; and anthracitic coals have not yet been produced in commercial quantities.

Mining conditions in the state are in general complex. Very few seams of gentle dip are mined, most operations are conducted on steep dips, mechanization is limited to a few operating properties, and hand methods predominate. Cost of mining consequently is high, and a large proportion of the output, 66 per cent in 1936, must be washed before shipment. Seventy mines were in operation in 1937, with a total output of 2,018,036 tons, valued at $6,402,968; the number of men employed was 2934. Days worked during the year were 202. Kittitas County was the largest producer followed by King, Whatcom, and Pierce.

It is not possible here to give many analyses or extended data on properties of coals. A few typical examples from each of the producing counties are submitted in Table 2. Accurate data giving analyses of face samples and delivered coals, softening temperatures of ash, agglutinating and slacking values, and other related data are contained in a Bureau of Mines report (3) and a supplement to that report. The engineer interested in the utilization of Washington coals will find these two publications timely and authentic.

Modes of Transportation

The coal produced in the north-central and in the western part of the state, reaches all parts of the state by four transcontinental railroad lines or their branches. Western Washington coal is trucked extensively in the Puget Sound and southwestern districts; a small tonnage of coal from the Roslyn field moves easterly by truck but the main movement is by rail; some coal is handled by tidewater transportation from Seattle, Tacoma, and Bellingham. The principal distribution area is within the state; there is a small interstate movement to Alaska, Idaho, and Oregon; and some export trade to British Columbia and Pacific Ocean points. The truck movement of coal amounted to 25.5 per cent of the tonnage produced in 1938. Storage of coal because of seasonal and transportation difficulties is unnecessary, and all the important population centers can easily be served directly from the mines.

Utilization of Washington Coals

Washington coals enter into every field of use and activity in raw, briquetted, pulverized, and carbonized forms. Household heating, steam generation, locomotive and steamship fuel, copper smelting, magnesite burning, and ceramic firing represent direct uses either in hand, mechanical-stoker, or pulverized-firing methods. Coal has been used to manufacture coke in bench, beehive, and by-product ovens, also in the various coal-gas, producer-gas, and water-gas processes. Briquettes are used in domestic and industrial firing and occasionally for orchard-heating purposes. The coal can be used as a reducing agent or as raw material in some chemical and metallurgical processes. Washington coke has been used in metallurgical plants, in foundries, in sugar refineries, in producer- and water-gas production, and in domestic heating. In short, some type of local product has been found suitable for coal utilization in its many fields and applications.

In 1938, the output was utilized as shown in Table 3. Although the demand for coal for some uses, namely, manufactured-gas plants, cement mills, smelters, miscellaneous industrial plants, and commercial and domestic heating, shows an increase over the last decade when expressed as a percentage of state

TABLE 2 TYPICAL ANALYSES OF TIPPLE SAMPLES OF WASHINGTON COALS

County and district	Designation and size,[a] in.	Preparation	Moisture, as received, per cent	Moisture-free basis: Volatile matter, per cent	Fixed carbon, per cent	Ash, per cent	Sulphur, per cent	Btu	Ash-softening temperature F
Whatcom	Stoker, 5/8 sh −5/16 sh	Washed	10.5	35.6	45.0	19.4	0.3	11010	2640
	Buckwheat, −5/16 sh	Washed	8.9	36.0	48.7	15.3	0.3	11600	2510
King									
McKay	Steam, −1 sh	Washed	12.8	43.0	52.3	4.7	0.6	13310	2160
Cumberland	Steam, −7/8 sh	Washed	5.4	36.9	45.0	18.1	0.6	12070	2910+
Renton	Buckwheat, −1/2 sh	Washed	14.7	39.9	46.6	13.5	0.6	11790	..
Pierce	Steam, −1 1/4 bar	Raw	3.2	35.9	52.0	12.1	1.0	13530	2280
Kittitas	Steam, −1 5/8 rh	Washed and dried	4.9	38.4	46.9	14.7	0.3	12710	2370
	Steam, −3/4 rh	Washed and dried	3.4	38.4	48.4	12.8	0.4	12890	2600
Thurston	Steam, −5 bar	Raw	19.8	40.4	47.0	12.6	0.7	11360	2420
	Stoker, 1 3/4 rh −1/4 sh	Washed	22 4	42.8	46.9	10.3	0.6	11600	2390

[a] sh = square-hole screen; rh = round-hole screen; bar = bar screen.

TABLE 3 USES OF COAL, 1927, 1936–1938[a]

Use	Per cent 1927	1936	1937	1938
Colliery fuel	2	1	1	1
Railroad fuel, including shop and station coal	43	31	29	29
Public-utility central-heating plants	5	3	4	4
Public-utility steam-electric plants			1	0
Fuel-briquette plants	3	0	0	0
Beehive-coke ovens	2	0	0	0
By-product-coke ovens	3	3	1	0
Other manufactured-gas works, including boiler coal	2	2	3	3
Cement mills	6-8	11	11	12
Smelters	.....	2	2	3
All other industries	7-14	6	13	13
Commercial and domestic heating	15–23	37	33	32
Steamship bunkers	1–2	0	0	0
Foreign exports and shipments to Alaska	1–2	4	1	2
Shipments to adjacent states	.....		1	1
State production, net tons	2,635,062[b]	1,812,104[b]	2,018,036[b]	1,572,057[c]

[a] Refer to Bibliography (3).
[b] Production reported by Bureau of Mines.
[c] Production reported by State Inspector of Coal Mines.

production, the actual tonnages have decreased. The most drastic curtailments in demand have occurred in coal used by railroads, briquette plants, coke plants, and steamships. Railroads and commercial and domestic heating, however, continue to be the most important consumer units.

The production of fuel briquettes began in 1911, and reached a maximum in 1917, with an output of 109,177 tons. The total reported production 1914 to 1938 amounted to 1,069,302 tons. In addition to coal briquettes, petroleum-carbon briquettes, made at oil-gas plants in Portland and Seattle, are sold for domestic use, and they are also replacing coke and coal in some water-gas processes. "Packaged fuel" made from outside coals and from petroleum coke is being produced and marketed in Spokane.

Pulverized coal has been used at power plants, smelters, cement mills, and miscellaneous industrial establishments since 1918. The quantity used reached a maximum of 350,000 tons in 1937; in 1938, sixteen per cent of the output of the state was fired in this form. Table 4 shows the consumption in 12 plants

TABLE 4 POWDERED-COAL CONSUMPTION IN NET TONS BY TYPES OF USERS[a]

Year	Cement plants	Smelters	Utilities, principally steam plants	Miscellaneous	Total
1927	225000	42000	50000	2500	319500
1928	175000	42000	51500	2000	270590
1929	167736	45862	52807	2600	269005
1930	186408	44204	56203	3882	290697
1931	138643	34842	80984	10898	265367
1932	59409	26057	80004	10675	176145
1933	33438	22223	77943	8710	142314
1934	67882	27454	60350	10266	165952
1935	60956	43202	56827	8419	169404
1936	195913	46095	63451	9240	314699
1937	224562	41837	73069	9709	349177
1938	140752	45203	58131	9434	253520

[a] Refer to Bibliography (3).

using Washington coal. Slack and buckwheat sizes produced in Whatcom, King, and Kittitas Counties supply the greater portion of this market.

The degree to which coal mined outside of the state, wood, fuel oil, and hydroelectricity affected the distribution and use of fuels in 1936 is shown in the analysis given in Table 5.

Price Schedule for Coals

Minimum prices for coal have been proposed and recommended by the Bituminous Coal Commission for various market areas in Washington. The entire schedule is too complicated to include in this review; instead, a summary, Table 6, will be given of prices of principal industrial sizes for shipment into the Seattle market territory. It is necessary to remember that the coals from Kittitas and Pierce Counties are bituminous; Thurston, Lewis, and Cowlitz Counties are subbituminous of low rank; Whatcom County coals are bituminous; and the King County coals are divided into the subbituminous, low-ash coals of the McKay district, the subbituminous of the Renton district, and the bituminous of the Cumberland district. Market prices may be expected to exceed the recommended minimum code figures.

Coals for Coking Purposes

Washington contains the only important coking-coal resource along the Pacific Coast. California and Oregon possess none, so far as present evidence is available; western British Columbia has coking coals but not of commercial significance at the present time; Alaska contains high-grade resources which are practically undeveloped and remote from transportation. Eastern British Columbia produces coke which is mainly consumed in the "Inland Empire" region of Washington and Idaho. It appears certain that attention must be focused on Washington in any consideration of future supplies of proved resource of this coal.

The principal coking coals are found in the Wilkeson-Carbonado-Fairfax district of the Pierce County field. This area has supplied the major production of coking coals and beehive coke. Portions of the beds in the Roslyn field of Kittitas County possess fair coking properties; and some eastern King County coals have fair to good coking qualities which make them potential sources for mixing or blending purposes.

Coke has been produced since 1880, in beehive ovens, and one by-product plant in Seattle, primarily operated for gas, produced coke from 1914 to 1937. This coke, together with gas-house coke, has been used in a wide variety of applications. Its severest handicap has been a moderately high percentage of ash, offset to a degree by low sulphur content. Some investigation is under way to produce better coke by improved preparation of coals and by mixing suitable coals of low ash content. The production of beehive coke from 1880 to 1937 was 1,631,319 tons; production of by-product coke 1914 to 1937 was 709,429 tons; a grand total of 2,340,748 tons. No coke was produced in 1938 and 1939.

In this brief review an attempt has been made to show that Washington possesses large resources of coal of various ranks and grades suitable for the needs of modern industry. Many of these coals are not of as good quality as coals from eastern mines; geologic conditions of deposition were different, with the result that the beds contain higher ash and generally are more difficult to mine and to clean. However, the fact that they have been successfully used for the various needs and demands of consumers over nearly a century indicates the fact that they have a place to fill in the field of fuel utilization. Many investigations have been made of the washability, friability, slacking, and grindability properties, and of agglutinating and coking characteristics of these coals. These are outside the scope of this paper, but the reports are available to the engineer who may wish to secure detailed information; a few references are cited in the Bibliography. It appears that the matter of economics of use rather than availability and suitability of supply is the determining factor from the engineering standpoint.

Effect of Water-Power Developments on Use of Coal

The completion of gigantic programs of water-power development inevitably will curtail the demand for coal used primarily as sources of heat and power. Other outlets such as coke and gas manufacture, hydrogenation and low-temperature carbonization, chemical uses in metallurgical and electrothermal processes

TABLE 5 COMPETING FUELS AND HYDROELECTRIC POWER, 1936[a]

Fuel	Quantity, coal, net tons; oil, bbl; or electricity, kwhr	Coal or coal equivalent, net tons	Percentage of total power	Percentage of Washington coal output
Coal from other states........	261,000	261,000	3.4	14.4
Coal from Canada............	90,000	90,000	1.2	5.0
Total competing coal.......	351,000	351,000	4.6	19.4
Washington coal.............	1,812,000	1,812,000	24.0	100.0
Total coal................	2,163,000	2,163,000	28.6	119.4
Wood.....................	....	500,000[b]	6.6	27.6
Fuel oil....................	9,400,000	2,547,000[c]	33.7	140.6
Hydroelectric power..........	3,273,148,000	2,357,000[d]	31.1	130.1
		7,567,000	100.0	417.7

[a] Refer to Bibliography (3).
[b] Refer to Bibliography (4). No estimate of the amount of wood fuel used more recently than that of 1924 is available; consequently, that figure is used to indicate the approximate magnitude of competition from wood in 1936.
[c] A conversion factor of 0.271 ton of coal per bbl of oil was used; this factor is based on a carefully estimated average Btu of 11,400 per lb, as-received basis, for Washington coal, and an average Btu of 147,000 per gal for fuel oil.
[d] A conversion factor of 1.44 lb of coal per kwhr, the average for central stations in the United States in 1936, was used.

TABLE 6 RECOMMENDED PRICE RELATIONSHIP FOR WASHINGTON COALS VIA RAIL TRANSPORTATION INTO SEATTLE MARKET AREA

Size group, in.	Kittitas County	Pierce County	Thurston, Lewis, and Cowlitz Counties	Whatcom County	King County: McKay District	King County: Renton District	King County: Cumberland District
Mine run	4.00[a]	3.90	2.75	....	....	....	3.50
$3^1/_2 \times 0$	3.75	3.90	2.50	....	....	....	....
2×0	3.15[b]	3.65	2.00[b]	....	3.75	....	3.00–3.25
$1^1/_4 \times {}^7/_8$	....	....	....	3.25	....	....	....
$1^1/_4 \times 0$	....	3.50	....	....	3.50	....	....
1×0	2.95[b]	....	1.50[b]	....	....	....	2.90–3.15
${}^7/_8 \times {}^3/_8$	....	....	....	3.00	....	3.25	....
${}^3/_8 \times 0$	2.75	2.85	1.00	1.50	2.85	1.50	1.75
${}^3/_{32} \times 0$	....	....	....	....	1.60	....	....

[a] Prices listed shall be increased 25 cents per net ton for washed sizes.
[b] Prices listed shall be increased 10 cents per net ton for washed sizes.
NOTE: All prices are net ton, 2000 lb, f.o.b. transportation facilities of the mines. All size designations are for round-hole screens. When coal is subjected to any chemical, oil, or waxing process, an additional charge of not less than 10 cents per net ton shall be added.

may in the future offset part of the losses. In any event, the resource of Washington coal appears to be a valuable asset in the expected growth of the Pacific Northwest.

Acknowledgment

In the presentation of this paper, the author has made use of material prepared by him for publication in various earlier reports. Certain data and tables, dealing with production distribution and use in Technical Papers of the Bureau of Mines, have been quoted directly. Acknowledgment is here made of this assistance.

BIBLIOGRAPHY

1 "The Coal Resources of the World," Twelfth International Geological Congress, Canada, 1913.

2 "Our Coal Supply: Its Quantity, Quality, and Distribution," by M. R. Campbell, Proceedings of the International Conference of Bituminous Coal, Carnegie Institute of Technology, 1926.

3 "Analyses of Washington Coals," Bureau of Mines, Tech. Paper 491, 1931; also supplement to this paper, U. S. Bureau of Mines, Technical Paper 618, 1940.

4 "The Fuel Market Situation in the Pacific Northwest," by Joseph Daniels, Monthly Bulletin, No. 149, Canadian Institute of Mining and Metallurgy, Sept., 1924, p. 569.

5 "Preliminary Tests of Thirteen Washington Coals in a Powdered-Coal Boiler Plant at the University of Washington," by G. S. Wilson, H. F. Yancey, and Joseph Daniels, University of Washington, Engineering Experiment Station, Bulletin 58, 1931.

6 "Friability, Slacking, Low-Temperature Assay, and Agglutinating Value of Washington and Other Coals," by H. F. Yancey, W. A. Selvig, and K. A. Johnson, U. S. Bureau of Mines, Technical Paper 512, 1932.

7 "Estimation of the Grindability of Coal," by H. F. Yancey, O. L. Furse, and R. A. Blackburn, Trans. American Institute of Mining and Metallurgical Engineers, vol. 108, 1934, pp. 267–294; also vol. 119, 1936, pp. 353–369.

8 "Coal in Washington—Distribution, Geology, Mining, Preparation, Uses and Economic Value of Coal Resources of Washington," by Joseph Daniels, University of Washington, Engineering Experiment Station, Report 3, 1934.

9 "Ball-Mill Grindability Indexes of Some American Coals," by H. F. Yancey and M. R. Geer, U. S. Bureau of Mines, Report of Investigations 3409, July, 1938.

10 "Burning of Various Coals Continuously and Intermittently on a Domestic Overfeed Stoker," by H. F. Yancey, K. A. Johnson, A. A. Lewis, and J. B. Cordiner, Jr., U. S. Bureau of Mines, Report of Investigations 3379, 1938.

11 "Physical and Chemical Properties of Coke Made or Used in Washington," by H. F. Yancey, R. E. Zane, R. W. Fatzinger, and J. A. Key, U. S. Bureau of Mines, Technical Paper 597, 1939.

Discussion

D. S. HANLEY.[3] The coal industry in the Pacific Northwest is fortunate in having a College of Mines at the University of Washington, with which is associated such men as the author and to whom the industry can go for advice and assistance. Professor Daniels has always been most cooperative in anything of a constructive nature affecting the coal-mining business in our state.

In my opinion, the coal industry of the State of Washington has failed to take the steps it should have taken years ago in the way of research and an energetic sales campaign not only to retain the business it enjoyed but to increase its markets. The inroads of fuel oil from California, commencing about the year 1911, have resulted in the loss of a substantial part of the coal market. Notwithstanding the fact of the large increase in population and in industrial activity, the total coal output of the state now is practically the same as it was 50 years ago. What was once one of the three or four largest industries in the state should not feel proud of its apathy in standing idly by and permitting its markets to be gradually taken away and absorbed by other fuels.

[3] Vice-President, Pacific Coast Coal Company, Seattle, Wash.

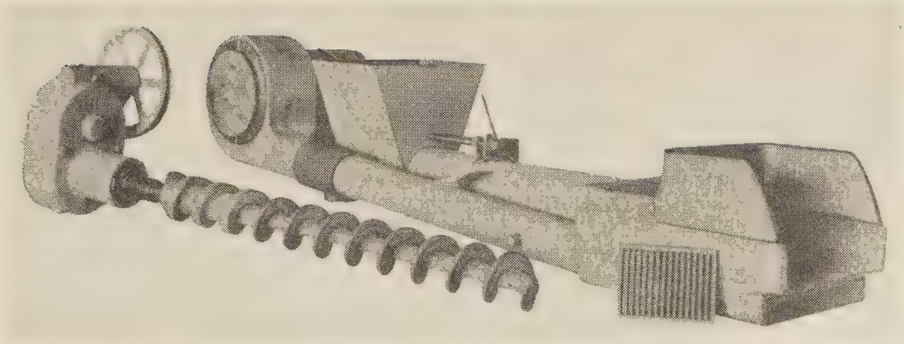

Fig. 1 Principal Parts of Overfeed Stoker, Including Reduction-Gear Box and Coal-Feed Screw; Fan Housing, Adjustable Intake Damper and Air Duct; Sliding Feed Gate, Underside of Grate and Retort

Burning Characteristics of Washington Coals on Domestic Overfeed and Underfeed Stokers

By H. F. YANCEY,[1] K. A. JOHNSON,[2] and J. B. CORDINER, JR.[3]

During the last 30 years, rapid advances have been made in the efficiency of fuel utilization in industry, but only during the last 10 years has comparable progress been realized in domestic heating. The most important advance in this field has been the development and wide application of the domestic coal stoker. The use of stokers for domestic heating originated in the Pacific Northwest, where two types, the underfeed and overfeed stokers are common. Elsewhere in the United States, the underfeed type predominates for domestic use. Studies[4] were made and reported upon in 1938 by the U. S. Bureau of Mines in cooperation with the College of Mines of the University of Washington on the subject of burning Washington coals on overfeed domestic stokers. Results were given of 25 combustion trials. Since then, the authors have conducted additional tests to include Washington and Oregon coals using both overfeed and underfeed stokers. This paper[5] describes the burning characteristics of caking and noncaking coals when fired on domestic stokers of both types installed successively in the same hot-water boiler. The results of burning trials with five coals selected from available data show the influence of variations in caking properties and ash-softening temperatures when used on both types of stokers.

THIS paper presents a comparison of the results of burning trials of coals, having different physical and chemical properties, on an overfeed and on an underfeed stoker. The coals were selected to demonstrate the influence of varying caking properties and different ash-softening temperatures on their performance with the two types of stokers. These two properties and size composition probably are the most important in affecting the behavior and suitability of coal for domestic-stoker burning. Caking is especially important in influencing the performance of coal on overfeed stokers, but is herewith considered for both types.

Coal Used

Five representative Washington coals, ranging in rank from subbituminous A to medium-volatile bituminous were fired. In caking properties, the coals ranged from nonagglomerating through weakly agglomerating to poor and good caking. In ash-softening temperature, they ranged from 2160 F to 2910 F. Ash contents ranged from 4.5 to 12.5 per cent.

Table 1 gives names and significant properties of coals tested. The description of caking or agglomerating properties is based upon the appearance of residue obtained by the standard method for the determination of volatile matter in the coal.[6]

The Harris and McKay coals were noncaking, but the other three coals, Roslyn, Roslyn-Cascade, and Wilkeson had caking strengths or agglutinating values of 700, 2300, and 10,240 g, respectively.

Comparative agglutinating values of coals from other states are as follows: Two samples from the Pocahontas No. 3 bed, West Virginia, had agglutinating values of 12,600 and 7130. One sample from the Pittsburgh bed in western Pennsylvania had an agglutinating value of 9810, while the Thick Freeport bed in the

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[4] "Burning of Various Coals Continuously and Intermittently on a Domestic Overfeed Stoker," by H. F. Yancey, K. A. Johnson, A. A. Lewis, and J. B. Cordiner, Jr., U. S. Bureau of Mines, Report of Investigations 3379, 1938.

Contributed by the Fuels Division and presented at the Fall Meeting, Spokane, Wash., September 3–6, 1940, of The American Society of Mechanical Engineers.

[5] The work upon which this report was based was performed under a cooperative agreement between the Northwest Experiment Station, Bureau of Mines, United States Department of the Interior, and the University of Washington, Seattle, Wash. Published by permission of the Director, Bureau of Mines.

[6] "Agglomerating and Agglutinating Tests for Classifying Weakly Caking Coals," by R. E. Gilmore, G. P. Connell, and J. H. H. Nicolls, Trans. American Institute of Mining and Metallurgical Engineers, Coal Division, vol. 108, 1934, pp. 255–265.

TABLE 1 NAMES OF COALS TESTED AND THEIR PROPERTIES

Coal	Rank	Caking properties	Ash-softening temperature, F
Harris	Subbituminous A	Noncoherent	2910
McKay	Subbituminous A	Slightly coherent	2160
Roslyn 5	High-volatile A bituminous	Weak agglomerate	2340
Roslyn-Cascade 1	High-volatile A bituminous	Poor caking	2635
Wilkeson-Wingate	Medium-volatile bituminous	Good caking	2330

TABLE 2 ANALYSES OF COALS USED IN STOKER TRIALS, AS-FIRED BASIS[a]

Coal	Harris	McKay	Roslyn 5	Roslyn-Cascade 1	Wilkeson-Wingate
Size, in.	$^3/_4$–$^3/_{16}$	$^7/_8$–0	$^3/_4$–$^1/_4$	$^3/_4$–0	$1^5/_8$–0
Analysis:					
Moisture, per cent	15.8	10.8	3.6	3.0	4.6
Volatile matter, per cent	32.9	38.2	38.0	37.0	24.7
Fixed carbon, per cent	38.8	46.5	46.5	48.7	58.8
Ash, per cent	12.5	4.5	11.9	11.3	11.9
Calorific value, Btu per lb	9850	11800	12390	12770	12930
Softening temperature of ash, F	2910	2160	2340	2635	2330
Agglutinating value, grams	0	0	700	2300	10240
Caking properties[b]	NAa	NAb	Aw	Cp	Cg

[a] Analyzed under supervision of H. M. Cooper, chemist, Pittsburgh Station, U. S. Bureau of Mines; ultimate analyses were made also but are omitted.
[b] NAa designates a nonagglomerate, noncoherent, volatile-matter residue; NAb, a nonagglomerate, slightly coherent residue that may be crushed by a 500-g weight; Aw, weak agglomerate; Cp, poor caking; Cg, good caking.

same general area showed a value of 6430. Samples of the Elkhorn bed in eastern Kentucky and the No. 6 bed in central Illinois had values of 4200 and 2120.

Table 2 shows the proximate analyses, calorific values, ash-softening temperatures, and agglutinating values,[7] or caking strengths of the five coals.

Stokers Used for Tests

Fig. 1 shows the principal parts of the overfeed stoker used in the work. The flat retort of this stoker gives a thin fuel bed typical of this method of burning. The available space on the retort is 9 in. wide × 8 in. long, giving a fuel-bed area of 0.5 sq ft. A sliding gate over the opening in the feed tube and the design of the screw allow the rate of feed to be varied between 10 and 25 lb of coal per hr.

Fig. 2 shows the underfeed stoker, installed in the hot-water furnace which was employed with both stokers, and the testing equipment. The retort of this stoker is of the usual pot type with an upper rim 11 in. in outside diam. The thickness of the fuel bed depends upon the type of coal being burned. A hydraulic mechanism allows the speed of the feed screw to be varied to feed between 11 and 27 lb per hr.

Furnace and Boiler

The boiler shown in Fig. 2, is of a size and type commonly used in residential heating by stoker or oil firing and comprises six vertical cast-iron sections. It has 52.6 sq ft of heating surface and a rated capacity of 565 sq ft of steam radiation or 905 sq ft of water radiation. The firebox is 23 in. wide, $20^1/_2$ in. long and 30 in. high from the top of the brick base to the crown sheet, giving a normal volume of 8.24 cu ft.

The distance from the grate of the overfeed stoker to the crown sheet was 26.5 in. The ashpit below the boiler in the brickwork was 4 in. deep and of the same cross-sectional area as the firebox. Ashes were removed through a door in the back section of the boiler. With the underfeed stoker, the distance from the top of the retort to the crown sheet was 22 in., but the floor of the refractory surrounding the retort was 2 in. below this, making the effective height to the crown sheet 24 in. The clinker was removed by means of tongs through the fire door.

Testing the Overfeed Stoker

The testing procedure for the overfeed stoker included a 2-hr adjustment period, an overnight hold-fire period, a 2-hr preheating period the following morning, and an 8-hr test period, during which the data were collected.

Immediately after the second 2-hr period, the stoker was stopped, the hopper and ashpit cleaned, a weighed quantity of coal was placed in the hopper, and the test was commenced. The useful heat recovered during the trial was measured by passing water through the boiler to two tanks set on dial-type platform scales, by means of which the heated water was weighed. The flow of water through the boiler was regulated to give a hot-water temperature of approximately 150 F at the outlet, while the temperature of the cold water fed to the boiler was about 50 F.

Continuously, throughout the test period, a sample of the flue gas was withdrawn from the smoke pipe at a constant rate, as determined by a flowmeter, and subsequently analyzed for carbon

[7] "Test for Measuring the Agglutinating Power of Coal," by S. M. Marshall and B. M. Bird, Trans. American Institute of Mining and Metallurgical Engineers, Coal Division, vol. 88, 1930, pp. 340–383.

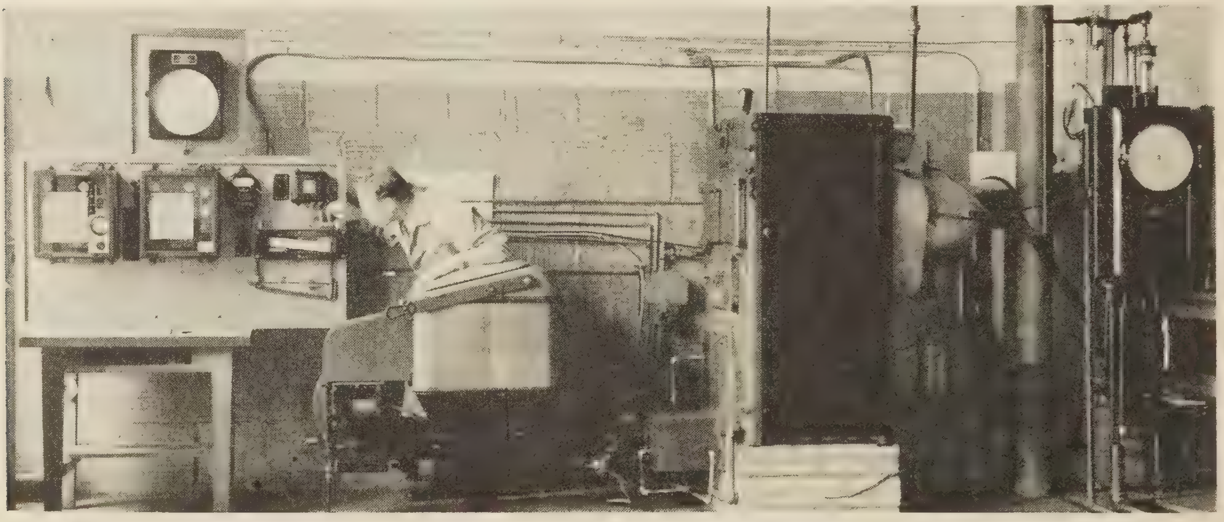

Fig. 2 Underfeed Stoker, Hot-Water Boiler, and Equipment Used in Coal-Burning Trials

dioxide, oxygen, hydrogen, carbon monoxide, methane, and unsaturated hydrocarbons. An absorption-type CO_2 recorder showed conditions during the test period.

The temperature of the flue gas was measured with a recording pyrometer, supplemented by the reading of an etched-stem mercury thermometer. The same method was used for determining the temperature of the hot water, except that a recording mercury thermometer was used instead of a thermocouple.

The condition of the fuel bed in the retort was noted from time to time through a Pyrex-glass observation window in the back door of the furnace. Comparative smoke readings were taken during the test at 15-sec intervals over a 30-min period, by comparing the density of the smoke with a Ringelmann chart.

At the end of the test, the soot was collected from the flue passages, smoke hood, and from the inside of the furnace by means of a vacuum cleaner.

Underfeed-Stoker Tests

The procedure for testing the underfeed stoker was similar to that for the overfeed stoker, with a few differences. Immediately after the second 2-hr period, the clinker was removed from the fuel bed and the thickness of the bed was recorded. The bed was observed periodically through the fire door. At the end of the test period, the clinker was removed. If necessary, enough loose ashes and coke or char were removed to give the same thickness of fuel bed as at the beginning of the test and appropriate adjustment was made.

Effect of Caking Properties

A comparison of the behavior of four of the five coals on both types of stokers, as indicated by the heat balance, is given in Table 3. The four coals tested were McKay, which is noncaking; Roslyn 5, weakly agglomerating with an agglutinating value of 700; Roslyn-Cascade 1, poor caking with an agglutinating value of 2300; and Wilkeson-Wingate, good caking with a value of 10,240. This is a range in caking properties from a coal which would produce only char on carbonization to one which would give a strong blast-furnace coke.

TABLE 3 PRINCIPAL RESULTS OF STOKER TRIALS OF CAKING COALS

Stoker	Overfeed			Underfeed		
Coal	McKay	Roslyn 5	Roslyn-Cascade 1	Roslyn 5	Roslyn-Cascade 1	Wilkeson-Wingate
Rate, lb per hr	13.3	13.4	14.2	15.5	16.7	17.0
Agglutinating value, g	0	700	2300	2020	3120	10240
Stack temperature, F	379	438	351	477	498	536
Carbon dioxide in flue gas, per cent	11.8	8.4	8.8	11.3	12.3	10.3
Heat balance:						
Efficiency, per cent	76.8	63.7	46.5	78.4	78.7	79.9
Losses, per cent						
Ashes	5.9	11.6	27.2	0.0	0.0	0.0
Soot	0.7	0.7	0.4	0.2	0.1	0.4
Dry flue gases	8.2	11.8	6.2	12.0	11.4	14.8
Moisture and hydrogen	5.5	5.0	4.2	5.4	5.2	4.4
Combustible in flue gases	0.0	0.5	6.5	0.0	0.4	0.0
Radiation and unaccounted for[a]	2.9	6.7	9.0	4.0	4.2	0.5
	100.0	100.0	100.0	100.0	100.0	100.0
Excess air, coal fired, per cent	48	77	27	58	45	70
Excess air, coal burned, per cent	59	102	79	58	45	71

[a] By difference.

Caking Coals on Overfeed Stoker

The most revealing item in the first part of Table 3, which deals with the overfeed burning of three coals that show increasing caking properties, is the efficiency. As caking increases, the efficiency decreases from 76.8 to 63.7 and to 46.5. At the same time, the loss of combustible in the refuse increases in the order, 5.9, 11.6, and 27.2, because the rate at which coal is supplied to the retort by the feed worm exceeds the burning rate; caking impedes the flow of air through the fuel bed.

All of the tests reported in this paper are continuous, that is, they do not simulate the off-and-on or intermittent operation of ordinary domestic-stoker operation. The latter type of burning also was used but is not reported in detail here. The ultimate effect, of course, is to lower the feed rate and allow more time for burning. The composition of the coal with respect to size is also an important factor in fuel-bed behavior, as is well known. For example, when the material finer than 4 mesh was removed from the 0 to $^3/_4$-in. Roslyn-Cascade coal (the one with the greater caking strength), it was burned continuously at about the same rate of feed with an efficiency of 57 per cent, compared to 46.5 per cent when unsized, and intermittently when sized with an efficiency of 65 per cent. This coal, however, would not be satisfactory for overfeed-stoker use and is not sold for that purpose.

Caking Coals on Underfeed Stoker

Table 3 also shows the results of continuous burning trials with the underfeed stoker on three caking coals, two of which, those of least caking strength, Roslyn 5 and Roslyn-Cascade, were burned on the overfeed. In contrast to the low efficiencies obtained with these two coals on the overfeed, they were burned with high efficiencies, 78.4 and 78.7 per cent, on the underfeed. The strongest caking coal, Wilkeson-Wingate, was burned with about equally high efficiency.

Differences in the caking properties of these coals affected conditions in the fuel beds. For example, as the agglutinating values of the coals increased, the resulting coke was less reactive, the formation of coke trees increased, and the fuel beds increased in thickness. Caking coals appeared to burn with thick fuel beds on the underfeed stoker, the thickness depending upon the natural accumulation of loose ashes, clinker, and coke. At the beginning of a test, just after a clinker was removed, the accumulation of loose ashes and coke gave a fuel bed that ranged in thickness from 3 to 7 in. above the top of the retort, according to whether the coal was weakly or strongly caking. As the test progressed, the thickness of the bed increased, owing to the formation of clinker. Just before the clinker was removed, either at the end of the test or at such time during the test as clinker removal became necessary, the thickness of the main part of the bed ranged from 7 to 9 in. The coke trees protruded occasionally as much as 9 in. above the main part of the bed.

Although large quantities of coke accumulated at various times during the burning trials, no difficulty was experienced in burning it, even with the strongest caking coal. The fuel bed went through irregular cycles of change. Starting with a low accumulation of coke, the coke built up in an irregular pile, sometimes with one or more coke trees extending almost to the crown sheet. This buildup of the fuel bed was accompanied by a period of slow combustion, during which the rate of combustion was slower than the rate of feed. The appearance of the fuel bed at this time usually was poor. As the cycle progressed, the coke trees fell and a period of rapid combustion reduced the bed to normal thickness.

Sherman and Kaiser, and Barnes,[8] all of the Battelle Memorial Institute, have presented a careful analysis of the formation and burning of coke in the underfeed fuel bed. Candee, working at Washington State College, designed a special retort with fuel-bed agitator to improve the burning of caking coals

[8] "Combustion of Bituminous Coal on the Small Underfeed Stoker," by R. A. Sherman and E. R. Kaiser, Trans. American Institute of Mining and Metallurgical Engineers, Coal Division, vol. 130, 1938, pp. 388–401. Also: "Fundamentals of Combustion in Small Underfeed Stokers," by C. A. Barnes, Bituminous Coal Research, Inc., Technical Report 4, 1938.

and to remove clinker or ash from the fuel bed automatically.[9] By better distribution of air to the fuel bed and breaking large pieces of coke to prevent blowholes, more satisfactory combustion was reported to have been obtained.

Burning of Noncaking Coals

Two coals, McKay and Harris, both of which are noncaking, were selected to show, first, the burning characteristics of noncaking coals on both overfeed and underfeed stokers and then to show the effect of a wide difference in ash-softening temperatures. Two different lots of each coal were used in the trials. The ash-softening temperatures of the two lots of McKay coal were 2340 and 2160 F. The variation in the softening temperature of the ash of the two lots of Harris coal was less, namely, 2910 and 2900.

Effect of Ash-Softening Temperatures

Overfeed Stoker. Table 4 shows the results of the tests of the two coals on both types of stokers. In considering the data for the overfeed tests, it should be remembered that the McKay coal is an unsized product, while the Harris coal is sized, as has been shown in Table 2. Despite this difference, nearly the same efficiencies, 76.4 and 75 per cent, were obtained. However, the loss of heat in combustible in the ashes amounted to 4.4 per cent with the unsized McKay coal and 2.1 per cent with the sized but higher-ash Harris coal.

Because of the differences in ash content and in ash-softening temperatures, the fuel beds of these two coals were different. Although the two coals were burned at nearly the same rate, the fuel bed with the McKay coal was only about 2 in. thick, as compared with a 5-in. bed for the Harris coal. The McKay coal, because unsized, gave an irregular bed, sometimes with large blowholes, whereas, the Harris fuel bed was uniform throughout the trial.

TABLE 4 PRINCIPAL RESULTS OF STOKER TRIALS OF NONCAKING COALS

Stoker	Overfeed		Underfeed	
Coal	McKay	Harris	McKay	Harris
Rate, lb per hr	15.9	14.8	16.9	15.6
Ash content, per cent	4.8	12.0	4.3	13.0
Ash-softening temperature, F	2340	2910	2160	2900
Stack temperature, F	471	422	475	425
Carbon dioxide in flue gas, per cent	13.5	10.1	12.2	9.3
Heat balance:				
Efficiency, per cent	76.4	75.0	75.0	73.6
Losses, per cent				
Ashes	4.4	2.1	0.0	0.0
Soot	0.4	0.6	0.4	0.3
Dry flue gas	9.1	10.9	11.0	11.8
Moisture	1.0	1.9	1.1	1.9
Hydrogen	4.4	4.8	4.8	4.3
Combustible in flue gases	2.0	0.0	0.5	5.4
Radiation and unaccounted for[a]	2.3	4.7	7.2	2.7
	100.0	100.0	100.0	100.0
Excess air, coal fired, per cent	24	70	44	79
Excess air, coal burned, per cent	30	75	45	80

[a] By difference.

Because of the wide difference in ash-softening temperature, the character of the clinker was different. McKay coal gave dense, well-fused clinkers; the Harris clinker was porous, friable, and fell off the end of the retort in small pieces. Sized McKay coal was also tested but, because the rate of feed was lower than in the other trials, detailed data are not given. The efficiency of this test (at 12.4 lb per hr), was 73.4 per cent and the loss of carbon in the ashes was 0.2 per cent. Conditions in the fuel bed were much more uniform than with unsized coal.

Underfeed Stoker. Coal from the McKay bed is used widely for underfeed domestic burning, but the Harris coal is sold only for overfeed use. Data in Table 4 show that both coals burned with approximately the same efficiency on the underfeed and that the efficiency was nearly the same as with the overfeed stoker.

Combustion of the McKay coal on the underfeed stoker is characterized by a thin fuel bed. Because of the low ash-softening temperature, dense clinkers are formed close to the tuyère ring at the top of the retort. The accumulation of clinker, because of the low ash content of the coal and the high density of the clinker is slow. Nevertheless, this type of clinker, if allowed to accumulate too long, checks the flow of air supplied by the fan.

That a heat-balance statement, showing a reasonably high efficiency, fails to disclose the general suitability of a coal for stoker-burning, is exemplified by the data for the test of Harris coal on the underfeed stoker. Five underfeed trials were made on the Harris coal and the results of only the best of these is shown. Hence, the data do not truthfully represent the difficulty experienced in burning this coal on the underfeed stoker. To obtain these results, it was necessary to use considerable excess air and to carry a thick fuel bed. Because of the high ash-softening temperature, 2900 F, only 6 lb of clinker strong enough to be removed with tongs were formed. The remaining 10 lb were present in the fuel bed as loose ashes. Thus, the high ash-soften-temperature, although advantageous with the overfeed stoker, renders this coal unsuitable for underfeed burning.

Summary and Conclusions

In general, it is concluded from the foregoing tests that the overfeed stoker is suitable for noncaking coals which have a wide range in ash content and ash-softening temperature. Even coals with very refractory ash fusing above 2900 F are suitable. No coals with an ash fusibility below 2100 F were tested. Because ashes are eliminated from the overfeed fuel bed as they are formed, large quantities of ash are easily discharged.

As is well known, caking coals are unsuitable for overfeed stokers. Weakly caking coals, even when burned in a sized condition, gave a low efficiency because excessive amounts of unburned coke were lost in the ashes. In contrast, weakly caking coals were burned with high efficiency on the underfeed stoker.

Two noncaking coals, one having a low ash content and low ash-softening temperature and the other having a medium ash content and high ash-softening temperature, proved satisfactory for overfeed burning, but the one with a medium ash content and high ash-softening temperature was unsatisfactory for underfeed burning, because the ash could not be fused to a clinker.

Both noncaking and caking coals may be burned satisfactorily on the underfeed stoker. Only one coal strong enough in caking properties to be used for manufacturing metallurgical coke was tested. This coal was burned in the underfeed stoker with a high efficiency, but some strongly caking and swelling coals are known to give difficulty. With respect to ash content and ash-softening temperature, the bituminous-type underfeed stoker ordinarily is limited to the burning of coals having low to medium ash contents and not too high ash-softening temperatures.

Acknowledgments

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[9] "The Development of a Domestic Stoker to Burn Washington Coals," by E. W. Candee, State College of Washington, Engineering Bulletin No. 56, 1938.

The Radiation of Furnace Gases

By H. C. HOTTEL[1] and R. B. EGBERT[2]

Data of various investigators on the emission and absorption of radiation by carbon dioxide and water vapor are reviewed critically. Recommendations of procedure in calculation of heat transmission by gas radiation are given, Fig. 3, with Equation [7] or [8] for carbon dioxide and Fig. 10 with Equation [17] for water vapor. Partial pressure P_w of water vapor, at a constant value of P_wL, affects gas radiation, but probably to a much smaller extent than has been reported in previous literature. A simplified procedure is presented to allow for the effect of the gas shape on radiant heat interchange.

The last fifteen years have witnessed a recognition by engineers of the importance of infrared radiation from gases in affecting heat transfer, especially at high temperatures where radiation from such gases as carbon dioxide and water vapor may be several times the heat transmission due to convection. Within the last ten years the basis of estimation of such heat transfer has changed from one dependent on inadequate measurements of the infrared absorption spectrum at room temperature, combined with a series of simplifying assumptions, to one dependent on direct measurement of total radiation. Since such experiments are somewhat difficult and the range of variables to be covered is great, it is not surprising that no complete agreement exists at present as to the procedure engineers should adopt in the calculation of radiant heat transfer from flue gases. It is the object of this paper to examine critically the data obtained by various investigators, to present some new data, and to consider whether it is possible to resolve the conflict of existing recommendations.

General Formulation of Heat Interchange by Gas Radiation

There is both an experimental and a theoretical basis for expecting emission and absorption of radiation by gases to have importance in any high-temperature heat exchanger involving the presence of heteropolar gases, i.e., gases whose molecules are composed of atoms carrying charges. Since all gases except the elementary gases such as hydrogen, oxygen, nitrogen, and argon are to some extent heteropolar, only these elementary gases are free from infrared absorption and emission bands. Infrared gas radiation of importance in heat transfer is due to changes in the energy levels of the molecule owing to its rotation and interatomic vibration.

The net interchange of radiation between a gas and some other body, due to one radiating (and absorbing) component of the gas, can be represented by the following general formula

$$q/A]_{net} = (E_G - A_{G,S}) = f_1(L, P_G, P_T, C, T_G) - f_2(L, P_G, P_T, C, T_G, T_S) \quad \ldots\ldots [1]$$

where

$q/A]_{net}$ = net heat transfer (Btu)/(hr)(sq ft)
E_G = emission from gas to other body
$A_{G,S}$ = absorption, by gas, of radiation from other body
P_T = total pressure (constant at one atmosphere for present field of interest)
P_G = partial pressure of the radiating constituent of the gas, atm
C = composition of the remainder of the gas
L = length of radiant beam through gas mass, ft
T_G = gas temperature, deg R
T_S = temperature of other body (generally a surface) with which gas is exchanging heat, deg R

The effect on total emission of the composition C of the diluent is but incompletely known. Measuring the infrared absorption spectrum of carbon dioxide, Hertz (1)[3] found that hydrogen as a diluent gas in place of air increased the absorption by a small but definite amount, while the use of nitrogen or oxygen in place of air as diluent made no appreciable difference. Hence, at least in the case of carbon dioxide, variation in composition of the diluent nonradiating fraction of flue gas, which consists mainly of nitrogen with varying amounts of oxygen, will not affect the radiation from the gas, and can be ignored.

The heat-transfer relation now simplifies to

$$q/A]_{net} = f_1(L, P_G, T_G) - f_2(L, P_G, T_G, T_S) \quad \ldots\ldots [2]$$

Even this simplification leaves a large number of variables. Schack (4) and Hottel (3), in the absence of direct data, made two assumptions. The first was that the number of molecules in the path determined E_G or $A_{G,S}$, i.e., that the effect of P_G and L entered as a single variable, the product P_GL. The validity of this relation, known as Beer's law, had been examined in considerable detail by Hertz (1) in connection with carbon dioxide, and less completely by von Bahr (2). Hertz found that at a fixed total pressure the absorption depended upon P_GL and was substantially the same whether pure carbon dioxide and a short path length or a lower partial pressure and longer path length were used, so long as total pressure was held constant by a nonradiating gas such as air. As already stated, the use of hydrogen instead of air resulted in a small but definite increase in absorption, about 2 to 5 per cent, depending upon the absorption band studied. Von Bahr examined a wide range of gases and compared the absorption of monochromatic radiation by pure gas at one atmosphere in 3-cm path lengths with absorption by the same gas at 1/11 atm partial pressure and a 33-cm path length, the total pressure being maintained at one atmosphere by air or hydrogen. Generally, her measurements were restricted to a single spectral region near the peak of an absorption band. She found that, at constant total pressure and temperature, the absorption was dependent on the term P_GL alone for all bands investigated for carbon monoxide, carbon dioxide, methane, ethylene, acetylene, methyl ether, ethyl ether, and for the 3.0μ band for ammonia. For the 6.3μ band of ammonia, 1/11 atm and 33-cm path length produced only 90–95 per cent as much absorption as one atmosphere of ammonia in a 3-cm path length. Becker (5) found for HCl at the 3.3–3.5μ band a somewhat lower absorption when partial pressure is down and total pressure is maintained with air than for pure HCl at one atmosphere and a correspondingly shorter length, at constant

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Contributed by the Heat Transfer Group and presented at the Annual Meeting, New York, N. Y., Dec. 2–6, 1940, of The American Society of Mechanical Engineers.

[3] Numbers in parentheses refer to Bibliography at end of paper.

P_GL. It appears, therefore, from the work of Hertz, von Bahr, and Becker that at a constant total pressure of one atmosphere Beer's law is adequate for carbon dioxide but may lead to error, probably small, for certain other gases, notably ammonia. Water vapor, on which no data of this type are available, is chemically more similar to ammonia than to the other gases studied.

The second assumption was that, at a given P_GL, the absorption by a gas at any temperature T_G of radiation from a surface at T_S was equal to the emission from a gas at T_S (an assumption which is exact when $T_G = T_S$, by Kirchoff's law). That is, at constant P_GL and surface temperature the absorption by the gas, of radiation from the surface, is independent of gas temperature, even though increasing the gas temperature at constant P_GL decreases the number of radiating molecules. As will be shown later, this assumption is not always justifiable.

These two assumptions permitted a further simplification of Equation [1] or [2] to the form

$$q/A]_{\text{net}} = f_1(P_GL, T_G) - f_1(P_GL, T_S) \quad \ldots\ldots\ldots [3]$$

that is, to a single function f_1 of two variables P_GL and T, evaluated successively at T_G and T_S.

Direct Measurements

The direct measurement of total gas radiation, made by sighting a total-radiation pyrometer through the gas onto some sort of background, is actually a measurement of net radiant heat interchange between the thermopile surface and what it sees, which latter may be gas or gas with a black-body background. Two methods of taking data are used. One employs a black body at room temperature as a background or makes use

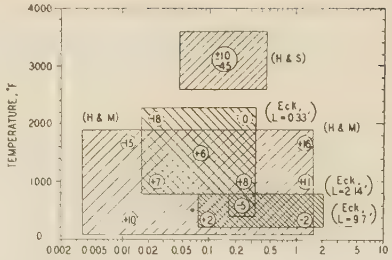

Fig. 1 Range of Variables Covered in Study of Carbon Dioxide Radiation

(Numbers in circles indicate percentage difference of different investigators, see text.)

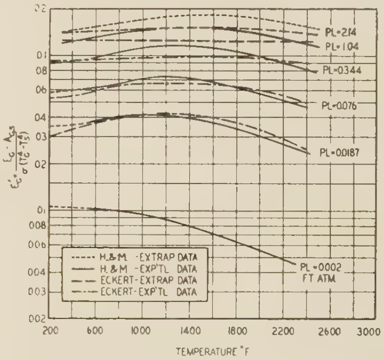

Fig. 2 Comparison of Data on Carbon Dioxide Emission

TABLE 1 SUMMARY OF EXPERIMENTAL MEASUREMENTS ON CARBON DIOXIDE

Author	Path length L, ft	Partial pressure P_C, atm	Gas temp t_G, F	Background temp t_S, F	Diluent gas	Knowledge of L	Uniformity of T_G	Stray radiation	Other comments
Hottel and Mangelsdorf, 1935	1.68	0.002 0.005 0.010 0.020 0.080 0.167 0.500 1.000	70 to 1910	—297 and 70 ± 2 to 2480	Dry CO_2-free air	Concentration traverse made. Sharp gradient within isothermal zone; established L to 2%	Uniform to 2 F	1.3% of black-body radiation at gas temperature	Radiometer sighted horizontally through horizontal tubes with nozzles at ends and hot-dry air protection at ends to prevent absorption at edges of zone. Mirror in radiometer only
Hottel and Smith, 1935	0.64 to 1.312	0.241 to 0.369	2568 to 3778	70 ± 2	$N_2 + O_2$	No concentration traverse. Reliance upon boundary between premix flame and air	Very narrow angle required by radiometer so that beam passed through isothermal gas zone. Sharp temp gradients at end. Temp measured by sodium-D line reversal method	Not determined. Should be negligible, as no hot surfaces present	Narrow-angle radiometer sighted through long Meker flame from combustion of CO-air and CO-air-oxygen mixtures. Mirror in radiometer only. Radiation is due to products of combustion only
Eckert, 1937	2.14	0.0355 0.062 0.13 0.25 0.50 1.00	212 to 752	75	N_2	No concentration traverse but a calculation of one made. Actually an error of 5 cm can be introduced under poor conditions according to experience of H.&M. with flow through similar nozzles	Good. Traverse given. No protecting air column at end except that due to partially preheated room air	Impossible to separate stray from mirror emission. Control experiment indicated total stray and emission from mirror was 1.8 to 2.6% of black body at gas temp	Apparatus involves two nozzles at bottom and mirror in top of vertical furnace. Plane mirror outside of furnace. Thermopile hence sees three mirrors in all. Originally, H_2 used as diluent but reacted with CO_2. To prevent heavier CO_2-N_2 mixture from mixing by gravity with air below furnace, high velocity through nozzles was used
	0.334	0.056 0.122 0.23 0.49 1.00	760 to 2300 390 to 2300	75	N_2	Same remarks as above	Same remarks as above	Stray = 1% of black body at gas temp	Apparatus similar in principle to H.&M. except that, instead of gas boundary formed by dry CO_2-free air at high temperature, partially heated unscrubbed atmospheric air is used
	9.7	0.46 0.23	212	75	H_2	No concentration traverse; no nozzle system to obtain sharp boundary	Uniform in furnace, gradient occurring over 10 cm length at ends	Total stray + emission from mirror + reflection of background = 6% of black body at gas temperature	Vertical furnace with mirror in top in hot zone. Background was flat surface

of a mirror so that the radiometer is its own background. This method yields emission characteristics only when the gas temperature is high enough to make negligible the term representing absorption, by the gas, of radiation from a black body at room temperature. The second method requires black bodies at different temperatures, ranging from liquid air to temperatures exceeding 2000 F, and permits separation of emission and absorption characteristics of the gas. Until this type of measurement is made there is a question as to the validity of using the net interchange factors, calculated from data obtained by the first type of measurement, as a basis for heat-transfer calculations.

The first experimental measurements were those of Lent and Thomas (6) on products of combustion of blast-furnace gas flowing in a duct four feet in diameter. A radiometer was sighted through the gas stream onto a water-cooled black surface. Uncertain path length and presence of considerable stray radiation make the data reliable to probably no better than 20 per cent.

Schmidt (7) next published the results of the first comprehensive investigation of water-vapor-radiation interchange between pure steam at different path lengths and temperatures and a blackened surface at room temperature.

Hottel and Mangelsdorf (8) studied steam-air mixtures, carbon dioxide-air mixtures, as well as steam-carbon dioxide-air mixtures, and measured both emission and absorption by using hot and cold black-body backgrounds. They varied the partial pressure and temperature of the radiating gas and kept the path length constant.

Hottel and Smith (9) measured emission from products of combustion of carbon monoxide and carbon monoxide-hydrogen mixtures at temperatures up to 3780 F.

Eckert (10), like Schmidt, measured the net interchange between the gas and a surface at room temperature, studying mixtures of carbon dioxide and nitrogen, carbon dioxide and hydrogen, and steam and nitrogen. Eckert varied the partial pressure and the temperature and varied the path length by the use of three different furnaces.

Eberhardt (11) measured the radiation from the flue gases in a steel reheating furnace, by sighting a radiometer across the gas within the furnace, through two openings, one in each side of the furnace, onto a black body. The gas contained both carbon dioxide and water vapor and had a path length of 14 ft. The temperature and the partial pressures of the radiating gases were varied.

Brooks (12) measured the emission and absorption of atmospheric air containing both carbon dioxide and moisture, at room temperature, and varied the path length.

Carbon Dioxide

The extent of the experimental investigation of carbon dioxide is indicated by Table 1 and Figs. 1 and 2. As already mentioned, Hertz (1) and von Bahr (2) found that Beer's law was valid for the various monochromatic absorption bands of carbon dioxide, and Eckert's results on three furnaces of different lengths support the same conclusion. Hence the radiant heat interchange between carbon dioxide and a bounding surface in industrial high-temperature heat-exchange equipment, which is generally operated at a fixed total pressure of one atmosphere, is dependent upon three operating variables, namely, gas temperature, surface temperature, and the product of path length by partial pressure of carbon dioxide, P_cL.

Fig. 1 indicates the range covered by the three most comprehensive investigations, namely, those of Hottel and Mangelsdorf (8), Hottel and Smith (9), and Eckert (10). Gas temperature is plotted against P_cL. Each cross-hatched rectangle represents the range in which emissivity can be obtained by direct interpolation of the actual data points of a particular investigator. As can be seen from Table 1, Eckert used a black-body background at room temperature only. Hence his measurements were reported as a pseudoemissivity ϵ_G', which is a net interchange factor between the gas and a black body at room temperature defined as follows

$$\epsilon_G' = \frac{(E_G - A_{G,R})}{\sigma(T_G^4 - T_R^4)} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

where

E_G = gas emission
$A_{G,R}$ = absorption, by gas, of room temperature radiation
T_G = gas temperature
T_R = room temperature
σ = Stefan-Boltzmann constant

As the temperature of the gas increases ϵ_G' approaches the true gas emissivity $\epsilon_G = (E_G/\sigma T_G^4)$, and above 1000 F the two can for all practical purposes be considered equal. For comparison with Eckert, the results of Hottel and Mangelsdorf have been converted to ϵ_G' wherever that quantity differs from ϵ_G. The numbers in the small circles in Fig. 1 represent the percentage difference between the recommendations of the two investigators, i.e.

$$\text{Number} = \left(\frac{\epsilon_{G}', _{\text{H.\&M.}} - \epsilon_{G}', _{\text{Eckert}}}{\epsilon_{G}', _{\text{H.\&M}}}\right) 100$$

Since the data of Hottel and Smith on carbon monoxide flames (top rectangle of Fig. 1) were used by Hottel and Mangelsdorf in establishing the high-temperature extrapolation of the latters' final recommended curves and since the flame measurements were accurate to about ten per cent, the agreement of the H.&M. extrapolation is likewise within ± 10 per cent of direct experimental data. This is indicated by the top number in the large circle. The lower number in the same circle is the percentage difference between Eckert's extrapolations and the H.&M. extrapolations. Fig. 2 likewise compares the two main groups of data; ϵ_G' is plotted as a function of temperature for several values of P_cL, with indication of whether each curve is based on data or extrapolations. An examination of both figures indicates that the recommendations of Eckert and of Hottel and Mangelsdorf agree to within 5 to 8 per cent in the range of temperature and P_cL where both investigators have experimental data. Outside this range at temperatures below 2000 F, discrepancies of 15 to 20 per cent exist. At high temperatures (about 3200 F) Eckert's recommendation is considerably higher.

Both Eckert and Hottel and Mangelsdorf took great care in measuring temperatures and kept errors caused by stray radiation down to less than 1.5 per cent of black-body radiation. The latter investigators determined the path length by a concentration traverse and used hot, dry, CO_2-free air as a windowless boundary to confine the radiating gas. Runs were made only when concentration gradients at the end were sharp and the temperature uniform through the center chamber. Although Eckert did not make such a concentration traverse, he used a system in which gas flow was probably somewhat steadier than that of Hottel and Mangelsdorf, and the concentration gradients at the boundary of his gas layer were probably satisfactory. Eckert allowed room air containing water vapor and carbon dioxide partially heated by the apparatus to form the boundary confining the radiating gas. As a result, concentration and temperature gradients occur simultaneously in the boundary layer, and the boundary gas contains some carbon dioxide and water vapor. This possibility of error may explain the discrepancies in the recommendations of the two investigators.

In addition to studies of emission, Hottel and Mangelsdorf

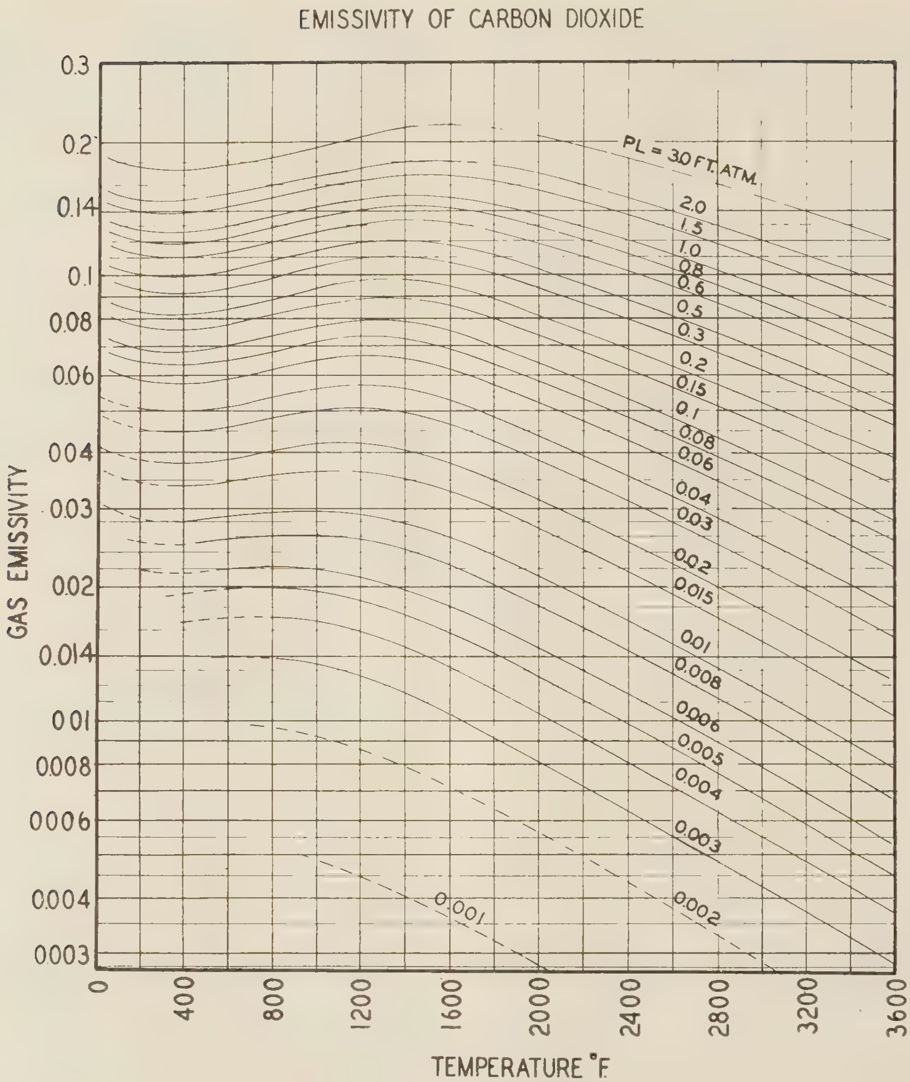

Fig. 3 Recommended Working Plot of Carbon Dioxide Radiation

made measurements of absorption, by the carbon dioxide, of black-body radiation from backgrounds at different temperatures. They found that, on maintaining the path length and partial pressure of carbon dioxide and the temperature of the source of black-body radiation all constant and varying the gas temperature, the absorption of radiation by the gas increased with an increase in gas temperature. Tingewaldt's (13) measurements of the absorption of monochromatic infrared radiation by carbon dioxide at various temperatures have confirmed the aforementioned result.

Although this finding prevented the simplification of the heat-transfer relation from the form represented by Equation [2] to that represented by Equation [3], an empirical relation between gas absorption and gas emission was established; consequently a single plot of emissivity in terms of T and PL is sufficient to determine both the total emission of radiant energy and the absorption of black-body radiation by carbon dioxide. Absorption is given in terms of emission by the relation

$$\alpha_{G,S} = \left(\frac{T_G}{T_S}\right)^{0.65} \epsilon_{S,P_cLT_S/T_G} \qquad [5]$$

where

$\alpha_{G,S}$ = absorptivity of gas at T_G for radiation from a black source at T_S

$\epsilon_{S,P_cLT_G/T_S}$ = gas emissivity at temperature T_S and at pressure-length product equal to P_cLT_G/T_S

A study of Table 1 and Figs. 1 and 2 does not permit an unequivocal decision as to which data to recommend for use. However, since the data of Hottel and Mangelsdorf are in good agreement with Eckert in the range covered by him and their extrapolation to high temperatures is in good agreement with Hottel and Smith's data, the H.&M. data have been used as a basis for constructing a working chart of emissivity versus temperature, for various values of P_cL. Such a chart appears as Fig. 3, to be used for evaluating terms in the expression for net radiant heat interchange between gas containing CO_2 and its bounding surfaces

$$q/A]_{CO_2} = \epsilon(E_G - A_{G,S})$$

$$= \epsilon\sigma(\epsilon_G T_G^4 - \alpha_{G,S} T_S^4) \qquad [6]$$

$$= 0.1723\ \epsilon \left[\epsilon_{G,P_cL} \cdot \left(\frac{T_G}{100}\right)^4 - \left(\frac{T_G}{T_S}\right)^{0.65} \cdot \epsilon_{S,P_cLT_S/T_G} \cdot \left(\frac{T_S}{100}\right)^4 \right] \ldots [7]$$

where

ϵ = emissivity (and absorptivity) of gray surface which bounds the gas.

Since the second or absorption term in the bracket is somewhat tedious to evaluate, it has been recommended (8) that when the gas temperature is high enough to make absorption much less important than emission (say, at least one third greater than surface temperature, absolute scale), the absorptivity $\alpha_{G,S}$ be assumed equal to gas emissivity at the surface temperature, E_{S,P_cL}. Then Equation [7] simplifies to

$$q/A]_{CO_2} = 0.1723\ \epsilon \left[\epsilon_{G,P_cL} \cdot \left(\frac{T_G}{100}\right)^4 - \epsilon_{S,P_cL} \cdot \left(\frac{T_S}{100}\right)^4 \right] \ldots [8]$$

This simplified form covers most furnace problems. Eckert has presented an analysis purporting to show that for cases in which gas temperature exceeds surface temperature the maximum error introduced by the use of Equation [8] instead of [7] is about 4 per cent; but the analysis is incomplete. If, for purposes of simplifying the analysis of error, one assumes $\epsilon_{S,P_cL_1}/\epsilon_{S,P_cL_2} = (P_cL_1/P_cL_2)^a$ and $\epsilon_{S,P_cL}/E_{G,P_cL} = (T_S/T_G)^b$, it may be readily shown that the error in use of Equation [8] instead of [7] is given by

$$\frac{q/A]_{\text{Eq. [8]}} - q/A]_{\text{Eq. [7]}}}{q/A]_{\text{Eq. [7]}}} = \frac{1 - (T_G/T_S)^{0.65-a}}{(T_G/T_S)^{4+b} - (T_G/T_S)^{0.65-a}} \ldots [9]$$

Fig. 3 indicates that b varies from +0.3 at 1000 F and high

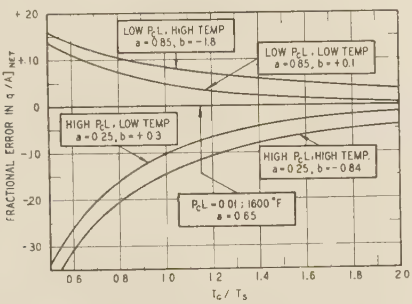

Fig. 4 Fraction Error in Calculation of Heat Interchange Due to Carbon Dioxide Caused by Use of Approximate Equation [8] Instead of [7]

P_cL's to −1.8 at 3000 F and low P_cL's; a varies from 0.85 at P_cL = 0.005 to 0.25 at P_cL = 2.0. Using Equation [9] and values of a and b limiting their range of importance, the percentage of error owing to use of Equation [8] has been calculated and is given in Fig. 4. It is apparent that caution should be employed in using the simplified form of equation.

Schack (14) has recently recommended an empirical equation for carbon-dioxide emission, obtained by fitting a simple power function of temperature and P_cL to the arithmetic mean of the Eckert and H.&M. data. Schack's equation is

$$E_G = 0.111 \sqrt[3]{P_cL} \cdot \left(\frac{T}{100}\right)^{3.5} \text{ Btu per sq ft per hr} \ldots [10]$$

or

$$\epsilon_G = 0.644 \sqrt[3]{P_cL} \Big/ \left(\frac{T}{100}\right)^{0.5} \ldots\ldots\ldots\ldots [11]$$

Inspection of Fig. 3 for which Equation [11] is proposed as a substitute makes it obvious that no such simple form of equation can be valid over a very wide range, since ϵ_G is proportional to the first power of P_cL at very low P_cL's and to a power of temperature varying from +0.3 at 1000 F and P_cL = 2 to −1.8 at 3000 F and P_cL = 0.01. However, in the temperature range 1000 to 2300 F and at values of P_cL between 0.02 ft atm and 1.0 ft atm Equation [10] gives values of emission which are within ten per cent of those calculated from the plot in Fig. 3. Outside this range the equation is greatly in error. For the absorption of black-body radiation by carbon dioxide Schack recommends the simplification already discussed—that $\alpha_{G,S}$ equals ϵ_G. His heat transfer equation then becomes

$$q/A]_{CO_2} = 0.111 \sqrt[3]{P_cL} \left[\left(\frac{T_G}{100}\right)^{3.5} - \left(\frac{T_S}{100}\right)^{3.5} \right] \ldots [12]$$

This equation may lead to an error of ± ten per cent ± error given in Fig. 4, even in the range for which Schack recommended it. By factoring $(T_G - T_S)$ out of Equation [12], one may obtain an expression for an equivalent or pseudoheat-transfer coefficient.

This is

$$h_{CO_2\ \text{rad}} = 0.0039 \sqrt[3]{P_cL}\ (T_{\text{ave}}/100)^{2.5}$$

in which T_{ave} is to an adequate approximation equal to the arithmetic average of gas and surface temperatures, degrees Rankine.

Water Vapor

The results on water vapor are less conclusive than those on carbon dioxide, and there is evidence that water-vapor emission at a fixed temperature and total pressure is not dependent solely upon P_wL, but rather on P_w and L separately. The chief investigations, summarized in Table 2, have been carried out in Danzig by Schmidt (7) and later by Eckert (10) under Schmidt's direction, and at The Massachusetts Institute of Technology by Hottel and Mangelsdorf (9). Also at M.I.T. unpublished data by Eberhardt (11) and Brooks (12) have been obtained. Fig. 5 indicates the range of variables covered by the various investigators. Since Beer's law has not been verified for water vapor, L and P_w must appear as separate variables. The three independent variables P_w, L, and T produce a three-dimensional figure, isothermal planes through which are presented in Fig. 5.

Schmidt measured interchange between a thermopile at room temperature and its field of view, namely, a jet of pure steam issuing from a nozzle at a relatively high velocity (about 60 fps) and at temperatures between 250 and 1760 F with a flat blackened plate at room temperature behind the steam jet. The partial pressure was kept constant at one atmosphere, and the path length L was varied by the use of three different-sized nozzles and by the use of mirrors. No concentration traverse was made to determine boundary effects, but these must have been appreciable, considering the well-known injection effects of a jet discharging at high velocity into a relatively stagnant fluid. The longest path length of 0.596 ft was obtained by sighting the radiometer across a 6-cm jet onto a mirror, back across the jet to a second mirror, and finally across the jet a third time onto a blackened flat plate, with the result that the beam of radiation passes through six boundary layers. A temperature traverse across the jet indicated an irregular variation—as much as 140 F at a mean steam temperature of 1090 F. No control experiment to determine stray radiation is indicated. Since Schmidt used a background at room temperature only, he expressed his results as the pseudoemissivity ϵ_G' already defined. He calculated the

TABLE 2 SUMMARY OF EXPERIMENTAL MEASUREMENTS ON WATER VAPOR

Author	Path length L, ft	Partial pressure P_w, atm	Gas temp t_G, F	Background temp t_S, F	Diluent gas	Knowledge of L	Uniformity of t_G	Stray radiation	Other comments
Schmidt, 1932	0.0325 0.0656 0.098 0.132 0.197 (0.394) (0.596)	1.0	250 to 1760	72 ± few deg (room temp)	None	No concentration traverse. Reliance on boundary of open jet. Two and 3 passes through jet to obtain L indicated in ( )	Poor. As much as 140 F irregular variation at higher temperatures (1000 F)	No control experiment indicated. Possibility of diffuse reflection of nozzle-wall radiation from flat-plate background	Radiometer sighted horizontally through vertical jet of steam discharging into cold air. Beam length doubled or trebled by use of mirrors, introducing edge-effects up to six times
Hottel and Mangelsdorf, 1935	1.68	0.005 0.010 0.020 0.040 0.080 0.167 0.500 1.00	70 to 1883	−297 and 70 ± 2 and 700 and 1840 and 2360	Dry CO_2-free air	Sharpness of gradient tested before each run by putting in CO_2 and analyzing. Runs only when gradient sharp. L good to 2%	Uniform to 2 F. Ends protected with hot, dry, CO_2-free air	1.3% of black-body radiation at gas temp	See comments in Table 1
Eckert, 1937	2.14	0.0305 0.061 0.122 0.260 0.534 1.00	250 to 750	75 (room temp)	N_2	See comments on Eckert's 2.14-ft furnace using CO_2, Table 1	Good. See Table 1	See Table 1, Eckert's 2.14-ft furnace. Possible error in method of applying correction	Apparatus has two nozzles, beam-entrance and beam-exit, at bottom of vertical gas furnace. Gold mirror inside furnace, plane mirror outside
	0.334	0.051 0.114 0.206 0.49 1.00	290 to 2300	Room temp	N_2	Same as above	Good. Same remarks as above	About 1% of black-body radiation at gas temp	See comments in Table 1 on Eckert's 0.334-ft furnace
	9.7	0.026 to 0.810	212	Room temp	N_2	See Table 1, Eckert's 9.7-ft furnace	Uniform in furnace. Gradient in ends occurring simultaneously with conc gradient. Varied from 70 to 212 F in 10 cm	See Table 1	Vertical furnace with mirror in top. No nozzles to produce sharp boundary. Background a flat surface. Possibility of fog formation at boundary from condensation of water vapor overflowing from apparatus

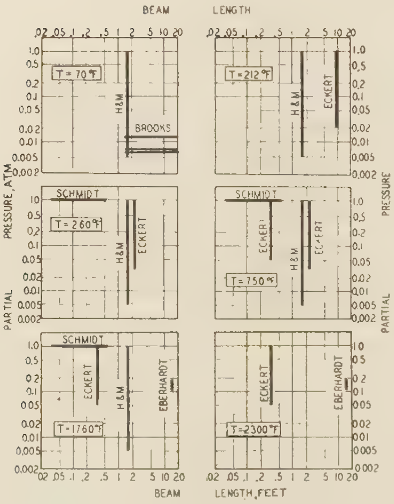

FIG. 5 RANGE OF VARIABLES COVERED IN STUDIES OF WATER-VAPOR RADIATION

emissivity of steam from monochromatic absorption measurements made by Hettner (15) on a pure steam column 3.5 ft long at 250 F, and used this value of emissivity to extrapolate his own results to high values of P_wL.

Hottel and Mangelsdorf measured the emission and absorption characteristics of steam-air mixtures, varying gas temperature, black-body temperature, and partial pressure and keeping the path length L constant at 1.68 ft. The temperature was uniform within 2 F over the entire path length. Before a run was made carbon dioxide was passed into the furnace to test the concentration gradient. When conditions of operation were such that the gradients were sharp, the system was thoroughly flushed free of carbon dioxide and water-vapor mixtures led into the furnace. A possibility of error arises here in that the conditions of operation which produced sharp concentration gradients with carbon dioxide might not have been maintained throughout the water-vapor runs. The stray radiation determined by a control experiment on dry carbon-dioxide-free air was about 1.5 per cent of black-body radiation, and was subtracted from all radiation measurements.

When the results of emission were compared with those of Schmidt, interesting discrepancies appeared. At small values of P_wL Schmidt's values of ϵ_G' are considerably larger than those calculated from Hottel and Mangelsdorf's measurements, while good agreement exists at high values of P_wL. These discrepancies indicated the possibility that P_wL was not a single independent variable but that increasing P_w produces a greater increase in emission than a corresponding increase in L. This possibility led Eckert under Schmidt's direction to make further investigations.

Eckert made measurements on three furnaces having different path lengths. One furnace was similar in design to that used by Hottel and Mangelsdorf, and confined gas having a path length of 0.334 ft. The gas temperature was varied between 290 and 2300 F, and the partial pressure of water vapor was varied between 0.051 and 1.0 atm, using nitrogen as a diluent gas. The stray radiation was small, about one per cent of black-body radiation, and the temperature was uniform throughout the gas column. No concentration traverse was made, and room air partially heated by the apparatus formed the windowless boundary between the radiating gas and the radiometer black-body system. Hence, possible sources of error include that of path length owing to possible uncertain boundary effect and that introduced by absorption and emission of radiation from the water vapor and carbon dioxide present in the air from the room.

A second vertical furnace contained a gold-plated mirror in its top in the heated zone. Radiation from a black body passed through one set of nozzles at the bottom of the furnace to the mirror, then back down through a second set of nozzles to a plane mirror, and thence into a radiometer. The path length was 2.14 ft, and the gas temperature was varied between 250 and 750 F. The emission from the hot gold mirror plus any stray radiation that was present was determined by a control experiment with

dry nitrogen in the furnace, and was used to correct the measurements. Air from the room, as before, formed the windowless gas boundary. A concentration traverse at this boundary was not made, though one was calculated.

A third furnace also contained a gold mirror and confined gas with a path length of 9.7 ft. This furnace was heated by condensing steam to 212 F and the gas, preheated to the same temperature, entered the top of the furnace in back of the mirror and overflowed out of the bottom, no nozzle system to get a sharp gas boundary being used. A water-cooled flat surface formed the background for the radiometer. The radiometer used in all of Eckert's work appears to be of good design, and was flushed with dry nitrogen gas. Eckert calculated all his results as the pseudoemissivity ϵ_G'.

In Fig. 6 the measurements of the different investigators are plotted as a function of P_wL for three different temperatures. Schmidt's results are higher than any of the others and the discrepancies increase as P_wL is decreased, but the data are in better agreement at high temperatures than at low. Eckert's results on his shortest path length agree with Schmidt's when pure steam is used but decrease more rapidly than Schmidt's with decreasing P_wL and are somewhat higher than the values calculated from the data of Hottel and Mangelsdorf. Eckert's values calculated from the data on his 2.14-ft furnace lie considerably below both his own values from the 0.334-ft furnace and those of Hottel and Mangelsdorf. The values calculated from the data on Eckert's largest furnace, L = 9.7 ft (taken at one temperature only), are lower than all other results. After he had examined his own data Eckert concluded that Beer's law does not hold for water vapor. Working at 212 F where he had data at a common value of P_wL and T from three furnace lengths plus Schmidt's data at P_w = 1, Eckert concluded that the four different measured emissivities could be expressed as a single function, the product (emissivity at the value of P_wL in question and at P_w = 1 atm) × (power function of P_w), i.e.,

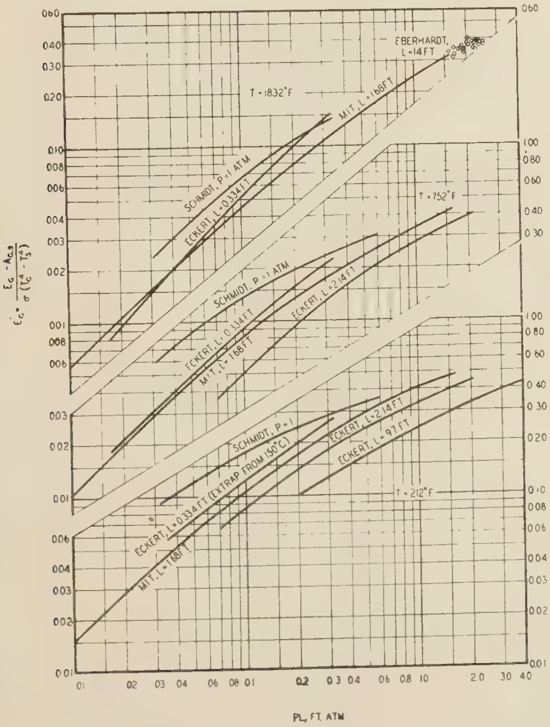

FIG. 6 COMPARISON OF DATA ON WATER-VAPOR EMISSION

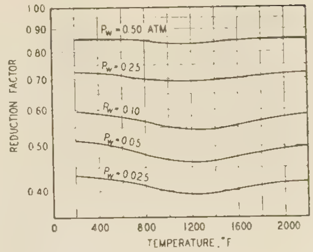

FIG. 7 ECKERT'S PROPOSED ALLOWANCE FOR EFFECT OF WATER-VAPOR PRESSURE ON EMISSION FROM WATER VAPOR AT CONSTANT P_wL

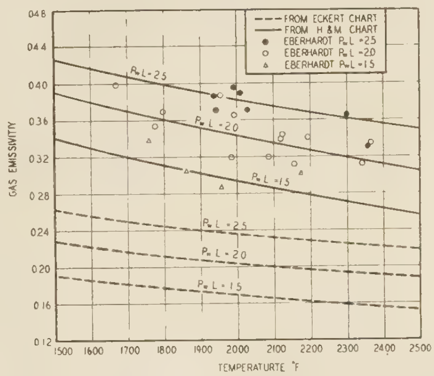

FIG. 8 COMPARISON OF DATA OF EBERHARDT ON WATER-VAPOR EMISSION WITH RECOMMENDED CURVES OF H.&M. AND OF ECKERT

$$\epsilon_G' = \epsilon_G']_{\text{at } P_w=1} \cdot (P_w)^x \qquad [13]$$

where the power x is a function of temperature only. At temperatures usually attained in industrial furnaces Eckert had data at one path length only, L = 0.334 ft, and hence could not verify this equation at high temperatures. However, he assumed that it was valid and used Schmidt's data in conjunction with his own to determine the factor $(P_w)^x$ at all temperatures. He presented this factor graphically as a function of temperature; his graph is reprinted here as Fig. 7.

Eberhardt (11) made gas-emission measurements from a steel reheating furnace in an industrial plant. He sighted across the furnace through square openings 15 in. in diameter with a radiometer which was carefully designed for minimum stray radiation and which required an object at 20 ft equal to the diameter of its mirror to fill the field of view. Eberhardt determined the temperature and concentration of carbon dioxide at various points along the line of sight and found that both were uniform for a distance of 14 ft, with sharp gradients in temperature and concentration occurring simultaneously at the edges. The carbon dioxide concentration was determined by Orsat analysis while the water-vapor concentration was calculated from the results of careful analyses of the fuel gas. The total radiation of the gas owing to both carbon dioxide and water vapor was measured with the radiometer. From the known path length of 14 ft and measured partial pressure of carbon dioxide the contribution of that constituent was calculated from Fig. 3 and subtracted from the total measured radiation. The remainder was radiation owing to water vapor alone.[4] The temperature varied between 1670 and 2370 F, while P_wL varied between 1.5 and 2.5 ft-atm. The results were converted to show the relation between ϵ_G and temperature at three fixed

[4] A small correction for superimposed radiation (see later discussion) was applied.

values of P_wL, by assuming that over the small range of P_wL involved the slope of the $\epsilon_G - P_wL$ relation was the same as in the H.&M. data. These converted results appear as data points in Fig. 8, along with the recommended curves of Hottel and Mangelsdorf (solid lines) and Eckert (dashed lines) corresponding to the same values of P_wL of 1.5, 2.0, 2.5 ft-atm.

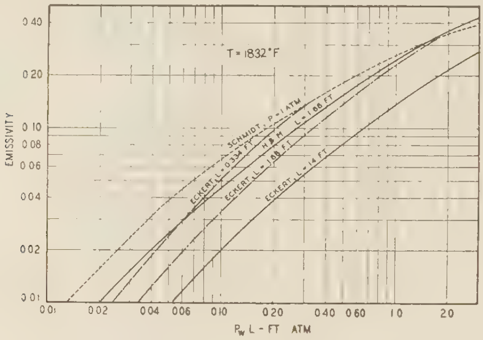

Fig. 9 Comparison of Recommended Curves for Water-Vapor Emission at 1000 C (1832 F)

It is apparent that Eberhardt's data are in excellent agreement with the solid lines, the maximum deviation of ± 10 per cent appearing to be random and due to indeterminate experimental errors. It is also apparent that the data are considerably higher than the curves recommended by Eckert, the calculation of which curves involves a correction downward owing to the low P_w even though P_wL is high.

Eberhardt's data were also corrected to a common temperature of 1832 F and plotted as points in Fig. 6. An extrapolation of the curve of Hottel and Mangelsdorf would pass through these points even though Eberhardt's data were obtained on a path length nine times as great and at correspondingly lower P_w's.

The emissivity of water vapor as a function of P_wL at 1832 F (1000 C) for three path lengths, $L = 0.334$ ft, $L = 1.68$ ft, and $L = 14$ ft, according to the recommendations of Schmidt, Hottel and Mangelsdorf, and Eckert is plotted in Fig. 9. Since both Schmidt and Hottel and Mangelsdorf assumed the validity of Beer's law, one curve suffices for all three path lengths. Eckert's recommendation yields three different curves when his correction for the effect of partial pressure is applied.[5] For the longest path length, $L = 14$ ft, values of P_w for the small values of P_wL were out of the range covered by Eckert's plot of correction factors. The values for the correction factor at 1832 F, according to Eckert's plot, can be calculated from the relation: factor = $(P_w)^{0.242}$. It is apparent that at the long path length of 14 ft Eckert's recommendation is quite low. This strongly suggests that his correction for pressure is excessive, at least for small partial pressures or high temperatures. Eckert himself suggests that the deviations from Beer's law which he found are due to association of water-vapor molecules. An increase in the partial pressure increases the association, and if associated water vapor has a higher emissivity than the unassociated state, then increasing the partial pressure at constant P_wL should increase the emissivity. At very low partial pressures or at high temperatures the association of water vapor becomes negligible. Therefore, if association is the cause of the deviation from Beer's law the law should be valid at very low partial pressures, rarely encountered in industrial practice, and at high temperatures such as encountered in most industrial furnaces. The agreement between Hottel and Mangelsdorf and Eberhardt as well as the fairly good agreement between Eckert's measurements at his highest temperatures on the 0.334-ft furnace and those of Hottel and Mangelsdorf indicate that this is the case.

Some measurements of the emission and absorption of radiation from atmospheric air containing moisture at room temperatures have been made by Brooks (12). A sensitive radiometer was sighted through laboratory air upon either of two black bodies, one filled with liquid air and the other with hot water. The distance between the black body and the radiometer was varied from 1.5 to 20 ft. Calibration of the radiometer was accomplished by permitting the radiometer to view the black body directly. Brooks made an attempt to correct for errors owing to inability to eliminate air with moisture and CO_2 between the radiometer and black body in the calibration of the radiometer, and for errors owing to boundary effects at the black bodies and radiometer. The boundary effect was particularly bad for the liquid-air-cooled black body as indicated by a visible fog which issued forth from it. Brooks' assumptions regarding the nature of these boundary effects were not sufficiently accurate to eliminate errors. About 15 per cent of the radiation was due to carbon dioxide in the air. When the radiation owing to carbon dioxide is subtracted from the measurements the remainder, radiation from water vapor, is about 40 per cent higher than calculations based on an extrapolation to room temperature of the H.&M. data at a P_L of 0.01 ft-atm, where the path length L was the same for both investigators. At Brooks' longest path length (20 ft) his measurements are in agreement with the H.&M. extrapolation. Brooks' measurements, while subject to considerable error, confirm Eckert's conclusion that at room temperature an increase in path length at constant partial pressure increases water-vapor emission less than does corresponding increase in partial pressure at constant path length. They also indicate that both the Eckert and the H.&M. extrapolations of emission measurements to room temperatures are possibly low. These discrepancies are of no consequence in furnace calculations.

Margaret Fishenden (16) in 1936 published the results of some radiation measurements on the products of combustion of city gas. Measurements on gas varying from 400 to 1600 F, at $P_wL = 0.205$ and $P_cL = 0.075$, yielded results from 5 to 21 per cent higher than calculations based on the H.&M. data and from 9 to 29 per cent higher than the Eckert data. The discrepancy in each case increased with temperature. Owing to stray radiation, uncertain path length, and temperature gradients along the line of sight of her radiometer Miss Fishenden's measurements are subject to considerable error. Recently she made measurement (17), yet to be published, on the absorption of radiation from hot black bodies by low-temperature steam-air mixtures, measurements which confirmed the inadequacy of Beer's law at low temperatures.

All these results point to the conclusion that at low temperatures Beer's law is not valid, and that some kind of correction factor must be used in conjunction with a single family of curves involving the three variables, emissivity, temperature, and P_wL. However, Eckert's proposed correction factor seems to be excessive at low temperatures as well as invalid at high temperatures. At very high values of P_wL, theory indicates that gas emissivity approaches unity and hence is independent of P_w at constant P_wL. Likewise if association is the cause of the experimental deviations from Beer's law then gas emissivity is independent of P_w at constant P_wL at very low values of P_w. Eckert's correction factor, a function of P_w and independent of P_wL, is such that unit emissivity cannot be attained at infinite P_wL unless $P_w = 1$, and it does not become constant when P_w is very small. One concludes that the correction factor is therefore theoretically unsound and is in error at low values of P_w and high values of P_wL.

[5] It is to be remembered that at this temperature Eckert has data at only one path length, namely, at $L = 0.334$ ft.

A more logical type of correction factor to be used in conjunction with the charts already published would be one which corrected P_wL instead of emissivity. One such form of correction might be

$$P_wL]_{P_w=1} = P_wL]_{\text{actual}}\,(1 - K + K\cdot P_w) \ldots\ldots [14]$$

This form of correction conforms to the conditions that emissivity be unity and independent of P_wL at infinite P_wL and that the correction become independent of P_w at very small values of P_w. Existing data are not sufficient to determine the factor K in the foregoing equations as a function of temperature. Eckert's data at 750 F and $L = 0.334$ ft (his most reliable furnace) and that of Hottel and Mangelsdorf can be brought together by Equation [14] with $K = 0.35$. This would give

$$\left(\begin{array}{l}P_wL \text{ for use with}\\ \text{charts based on}\\ P_w = 1, \text{ such as}\\ \text{Schmidt's}\end{array}\right) = (P_wL)]_{\text{actual}}\cdot(0.65 + 0.35\,P_w) \ldots [15]$$

and

$$\left(\begin{array}{l}P_wL \text{ for use with}\\ \text{chart based on}\\ L = 1.68\text{—that of}\\ \text{H.\&M. (Fig. 10)}\end{array}\right)$$

$$= 1.56\left\{\sqrt{1 + 1.97\,(P_wL)]_{\text{actual}}\,(0.65 + 0.35\,P_w)} - 1\right\} \ldots [16]$$

Since many other functions besides (14) exist which conform to the limits imposed by theory, equations such as [15] and [16] are certain to be replaced by better recommendations when adequate data become availale. Such an experimental study is now under way at M.I.T.

With the extensive and somewhat conflicting data on water-vapor emission in mind, particularly the Eberhardt data taken in the range of P_w and T encountered in furnace practice, the authors make tentative recommendations for calculations: For temperatures above 1200 F use the chart, Fig. 10, based on the H.&M. data, without any correction for deviations from Beer's law. For lower temperatures a correction should probably be used, and [16] is recommended temporarily. In the vicinity of 212 F Eckert's correction may be used, though we feel it overcorrects.

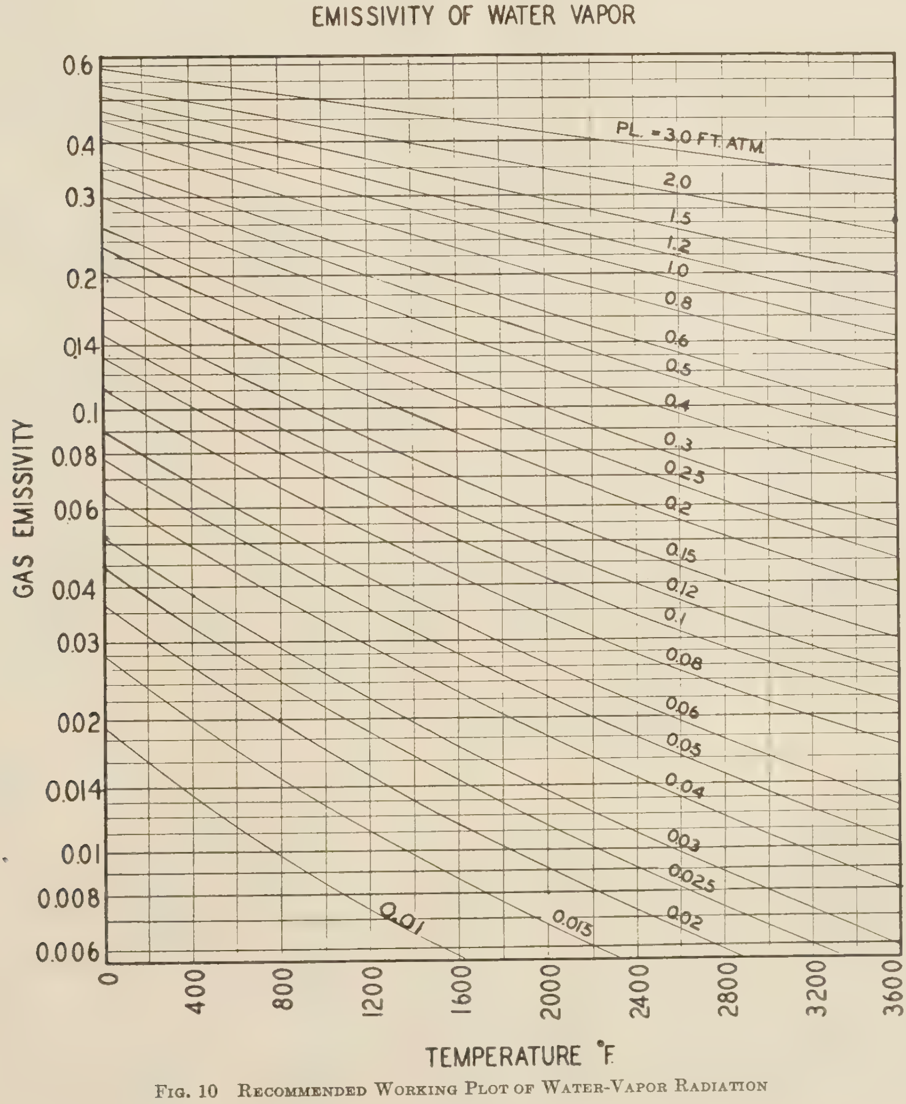

Fig. 10 Recommended Working Plot of Water-Vapor Radiation

The only measurements on the total absorption of black-body radiation by water vapor are those of Hottel and Mangelsdorf. They found that absorption by a gas at constant P_wL of radiation from a black body at a given temperature T_S is independent of gas temperature. Hence the absorption is equal to the emission the gas would exhibit if at the temperature T_S of the body. The

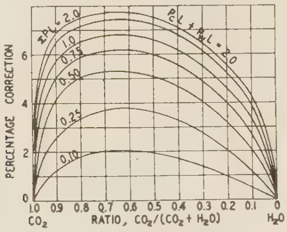

FIG. 11 CORRECTION DUE TO SUPERIMPOSED RADIATION FROM CARBON DIOXIDE AND WATER VAPOR

equation for radiant heat interchange between gas containing water vapor and its bounding surfaces is then of the form of the simplified carbon dioxide equation

$$q/A]_{H_2O} = 0.1723\left[\epsilon_{G,P_wL}\left(\frac{T_G}{100}\right)^4 - \epsilon_{S,P_wL^*}\left(\frac{T_S}{100}\right)^4\right] \quad [17]$$

where ϵ_G and ϵ_S are read from Fig. 10 at t_G and t_S and at a common P_wL.

Schack (14), using the arithmetic mean of the H.&M. data and those of Eckert, has developed equations for water-vapor emission similar to those already presented for carbon dioxide. He assumes that the emission increases with partial pressure at constant P_wL according to Eckert's recommendation. His equation is

$$q/A]_{H_2O} = 1.08\, P_w^{0.8}L^{0.6}\left[\left(\frac{T_G}{100}\right)^3 - \left(\frac{T_w}{100}\right)^3\right] \text{Btu per sq ft per hr} \quad [18]$$

Between 900 and 2400 F and $P_wL = 0.02$ to $P_wL = 1.0$ ft-atm this equation gives values of emission that are from 15 per cent lower to 15 per cent higher than those calculated from Fig. 10 when the path length is 1.68 ft. At other path lengths larger deviations may be expected due to the probably excessive correction for effect of P_w. The equation is recommended only where errors of 20 per cent can be tolerated. As with carbon dioxide, Equation [18] may be converted to yield a pseudo-heat-transfer coefficient by factoring out $(T_G - T_w)$. It is

$$h]_{H_2O\ rad} = 0.0324\, P_w^{0.8}L^{0.6}T_{ave.}^2 \quad [19]$$

MIXTURES OF CARBON DIOXIDE AND WATER VAPOR

The total emission from a gas containing both carbon dioxide and water vapor is less than the sum of the emissions due to the carbon dioxide and to water vapor, each evaluated as though the other were not present, because the emission and absorption bands of the two gases overlap. The magnitude of the difference between actual emission and the evaluated sum is a function of PL, temperature, and relative concentrations of the two gases. This difference is designated by the symbol K defined and by the following equation

$$E_{c+w} = E_c + E_w - K$$

Hottel and Mangelsdorf made the only extensive measurements of emission and absorption of radiation by mixtures of carbon dioxide, water vapor, and air. However, the accuracy of the determination of K was low since it involved differences of quantities of similar magnitude.

Eckert (10) made calculations of this difference K from the monochromatic absorption data. These calculations are in fair agreement with the measurements of Hottel and Mangelsdorf. This correction term varies from zero at small PL's to a maximum of 10 per cent of the total radiation at the highest PL's for which emission measurements have been made. Hence it is not necessary to know K accurately when probable errors in the existing recommendations for gas emission are considered. Fig. 11 presents the difference factor expressed as the percentage K' by which $(E_c + E_w)$ must be reduced to give E_{c+w}, plotted as a function of the ratio $\dfrac{P_c}{P_c + P_w}$ for several values of $P_cL + P_wL$. Since K' varies with temperature and the chart is presented for use at all temperatures, K' may be 50 per cent in error but this introduces at most a 4 per cent error in the calculation of the total radiant heat transmission from gases containing both carbon dioxide and water vapor.

EFFECT OF GAS SHAPE

The use of a definite value of P_GL in calculating interchange between a gas mass and its bounding surface presupposes a gas shape for which path length L is constant in all directions through the gas. The only shape for which that limitation is applicable is a hemisphere of gas radiating to a spot on the center of its base. For actual gas shapes a suitable mean value of L must be obtained, the radius of an "equivalent" hemisphere. This problem has been presented in some detail (3), and for various gas shapes of industrial importance mean values of L have been given. Table 3, column 2, gives references for the various shapes studied. Hottel and Port (21) have shown that at very low values of P_GL where E_G (or ϵ_G) approaches proportionality

TABLE 3 BEAM LENGTHS FOR GAS RADIATION

Shape	Bibliographic references	Characterizing dimension, D	Factor by which D is multiplied to obtain mean beam length L: When $P_GL = 0$	For average values of P_GL
Sphere	(18), (3)	diam	2/3	0.60
Infinite cylinder, radiating to walls	(18), (3)	diam	1	0.90
Rt. circ. infinite cylinder, rad to spot on center of base	(10)	diam	...	0.90
Rt. circ. cylinder, ht = diam; rad to whole surface	.....	diam	2/3	0.60
Same; rad to spot on center of base	(10)	diam	...	0.77
Infinite cylinder of half-circular cross section; rad to spot on center of flat side	(10)	radius	...	1.26
Space between inf. parallel planes	(3), (20)	separating distance	2	1.8
Cube	(3)	edge	2/3	0.60
1 × 2 × 6 rectangular parallelopiped, radiating to		shortest edge		
2 × 6 face	(3), (21)		1.18	1.06
1 × 6 face	(21)		1.24	
1 × 2 face	(21)		1.18	
all faces	(21)		1.20	
Space outside infinite bank of tubes with centers on equilateral triangles; tube diam = clearance	(3), (10)	clearance	3.4	2.8
Same, except tube diam = one half clearance	(3), (10)	clearance	4.45	3.8
Same, except tube centers on squares, tube diam = clearance	(10)	clearance	4.1	3.5

to P_GL, the value of L for *any* gas shape radiating to its bounding walls approaches as a limit the simple expression, four times the mean hydraulic radius of the shape, i.e., four times the gas volume divided by the area of the bounding walls. For the range of P_GL encountered in practice, the mean path length L is always less. A study of rectangular parallelopipeds of widely varying dimension ratios led to the conclusion that a satisfactory approximation consists in taking 85 per cent of the limiting value, four times mean hydraulic radius. This simple rule works quite well for other gas shapes, as borne out by a comparison of the last two columns of Table 3, giving values of L for various gas shapes for $P_GL = 0$ and for P_GL in the industrially important range. Since in the latter range E_G (or ϵ_G) varies as about the 0.3 power of P_GL, a 10 per cent error in choice of L produces only a 3 per cent error in the calculation of heat transmission.

BIBLIOGRAPHY

1 "Über das ultrarot Absorptionspektum der Kohlensäure in seiner Abhängigkeit von Druck und Partialdruck," by G. Hertz, Berichte der deutschen Physikalischen Gesellschaft, Verhandlungen, vol. 13, 1911, p. 617.

2 (*a*) "Über die Einwirkung des Druckes auf die absorption ultraroter Strahlung durch Gase," by Eva von Bahr, *Annalen der Physik*, vol. 29, 1909, p. 780.

(*b*) "Über die Einwirkung des Druckes auf die Absorption ultraroter Strahlung durch Gase," by Eva von Bahr, *Annalen der Physik*, vol. 33, 1910, p. 585.

3 "Heat Transmission by Radiation From Non-Luminous Gases," by H. C. Hottel, Transactions of the American Institute of Chemical Engineers, vol. 19, 1927, p. 173.

4 (*a*) "Der Wärmübergang in technischen Feurungen unter dem Einfluss der Eigenstrahlung der Gase," by A. Schack, *Mitteilungen der Wärmestelle*, vol. 55, 1923, p. 63.

(*b*) "Über die Strahlung der Feurgase und ihre praktische Berechnung," by A. Schack, *Zeitschrift für technische Physik*, vol. 5, 1924, p. 266.

5 "Über die Druckabhängigkeit der Absorption des Chlorwasserstoffs im Ultrarot," by G. Becker, *Zeitschrift für technische Physik*, vol. 34, 1925, p. 255.

6 "Versuche über die Eigenstrahlung der Gase," by H. Lent and K. Thomas, *Mitteilungen der Wärmestelle*, vol. 65, 1924, p. 208.

7 "Messung der Gesamtstrahlung des Wasserdampfes bei Temperaturen bis 1000 C," by E. Schmidt, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 3, 1932, p. 57.

8 "Heat Transmission by Radiation From Non-Luminous Gases II." "Experimental Study of Carbon Dioxide and Water Vapor," by H. C. Hottel and H. G. Mangelsdorf, Trans. American Institute of Chemical Engineers, vol. 31, 1935, p. 517.

9 "Radiation From Non-Luminous Flames," by H. C. Hottel and V. C. Smith, Trans. A.S.M.E., paper PRO-57-4, 1935, p. 463.

10 (*a*) "Die Wärmestrahlung von Wasserdampf in Mischung mit nichtstrahlenden Gasen," by E. Schmidt and E. Eckert, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 8, 1937, p. 87.

(*b*) "Messung des Gesamtstrahlung von Wasserdampf und Kohlensäure in Mischung mit nichtstrahlenden Gasen bei temperaturen bis zu 1300 C," by E. Eckert, *Forschung auf dem Gebiete des Ingenieurwesens*, Beilage no. 387 (1937).

11 "Heat Transmission in Continuous Metal-Heating Furnaces," by J. E. Eberhardt, Sc.D. thesis, The Massachusetts Institute of Technology, 1936.

12 "Measurements of Emissivity of Air From Path Lengths Up to 6 Meters," by Brooks and Hottel, private communication.

13 (*a*) "Die Absorption der Kohlensäure in Gebiet der Band $\lambda = 2.7\mu$ zwischen 300 und 1100 C Abs," by C. Tingewaldt, *Physikalische Zeitschrift*, vol. 35, 1934, p. 715.

(*b*) "Die absorption der Kohlensäure in Gebiet der Band $\lambda = 4.3\mu$ zwischen 300 und 1000 C absolute," by C. Tingewaldt, *Physikalische Zeitschrift*, vol. 39, 1938, p. 1.

14 "Die Strahlung der Feuergase," by A. Schack, *Archiv für das Eisenhüttenwesen*, vol. 13, Dec., 1939, p. 241.

15 (*a*) "Über das ultrarot Absorptionspecktrum des Wasserdampfes," by G. Hettner, *Annalen der Physik*, vol. 55, 1918, pp 476–497 and 545–560.

(*b*) "Über Gesetzmässigkeiten in den ultraroten Gasspektren und ihre Deutung," by G. Hettner, *Zeitschrift für technische Physik*, vol. 1, 1920, p. 345.

16 "Radiation From Nonluminous Combustion Gases," by M. Fishenden, *Engineering*, vol. 142, Dec., 1936, p. 684.

17 Private Communication from M. Fishenden, July, 1940.

18 "Die Gasstrahlung bei der Strömung im Rohr," by W. Nusselt, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 70, 1926, p. 763.

19 "Die Berechnung der Strahlung von Gasräumen," by E. Schmidt, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 77, 1933, p. 1162.

20 "Absorption und Strahlung russfreier Gase," a chapter in "Der Chemie Ingenieur," by M. Jacob, Leipzig Akademische Verlagsgesellschaft M.B.H., vol. 1, pt. 1, p. 301.

21 "Heat Transmission by Radiation From Gases," by F. J. Port, Jr., Sc.D. thesis, The Massachusetts Inst. of Tech., 1939.

Fig. 1 Chickamauga Development—A Typical Multipurpose Project

Kaplan Turbine Installations of the Tennessee Valley Authority

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Eight Kaplan turbines recently installed in the Pickwick Landing, Guntersville, and Chickamauga plants of the Tennessee Valley Authority have a total capacity of 300,000 hp at rated head, and in physical size are among the largest constructed. The paper reviews the function of Kaplan turbine plants in the Tennessee River development as a whole, the determination of turbine requirements, the power-station arrangement, and noteworthy features of turbine design, construction, and erection.

General Description

The development of the Tennessee River, in accordance with the terms of the Tennessee Valley Authority Act, comprises a series of multipurpose projects for the provision of a channel for 9-ft navigation in the river from Paducah, Ky., to Knoxville, Tenn., the control of destructive floodwaters in the Tennessee and Mississippi River basins, and the generation of hydroelectric power.

The prescribed navigation improvement is accomplished by means of locks and a continuous succession of pools, the minimum levels of which are governed by the drawdown which will afford a minimum but adequate navigation channel at the next project upstream, while the surcharge levels for flood control were fixed with reference to the resultant damage to cities, railroads, highways, and land. To augment the flow of the Tennessee River during the dry season and for the retention of headwater floods, storage projects are also provided at strategic locations on certain of the tributary streams.

The cycle of operation of the reservoir system is dictated by flood-control considerations during the winter and spring months, and by navigation and power requirements during the summer and autumn. This mode of operation is feasible because large floods on the Tennessee River, as at Chattanooga and below, occur only between the middle of December and April and are due primarily to the transit of typical seasonal storms along the river basin, starting at the western end and moving in a direction from southwest to northeast.

Accordingly the basic program is to deplete the storage reservoirs on the tributaries to low level by December 15, and to retain water in these reservoirs at safe rates in the interval between December 15 and April 15 and, subsequently, at higher rates governed by stream flow and use. On the other hand, the main river reservoirs are operated during the flood season with particular reference to immediate needs at Cairo, Ill., and the lower Mississippi River, and are depleted in the fall to a level consistent with good navigation; surplus water from the river reservoirs being discharged between flood crests on the Mississippi.

The major physical features of the various component projects are summarized for convenient reference in Fig. 2 and Table 1. With the completion of Kentucky, Watts Bar, and Coulter Shoals, the development will afford 650 miles of high-grade waterway, 9,000,000 acre-ft of flood storage (sufficient to reduce flood crests on the Mississippi 2 ft between Cairo and the Arkansas River), and 1,800,000 kw ultimate hydroelectric capacity.

Determination of Plant Capacity

In project planning, the ultimate capacity to be provided at each individual plant should be determined in advance, so that power-station intakes and draft-tube foundations may be provided during the initial construction to accommodate the ultimate number of units required.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

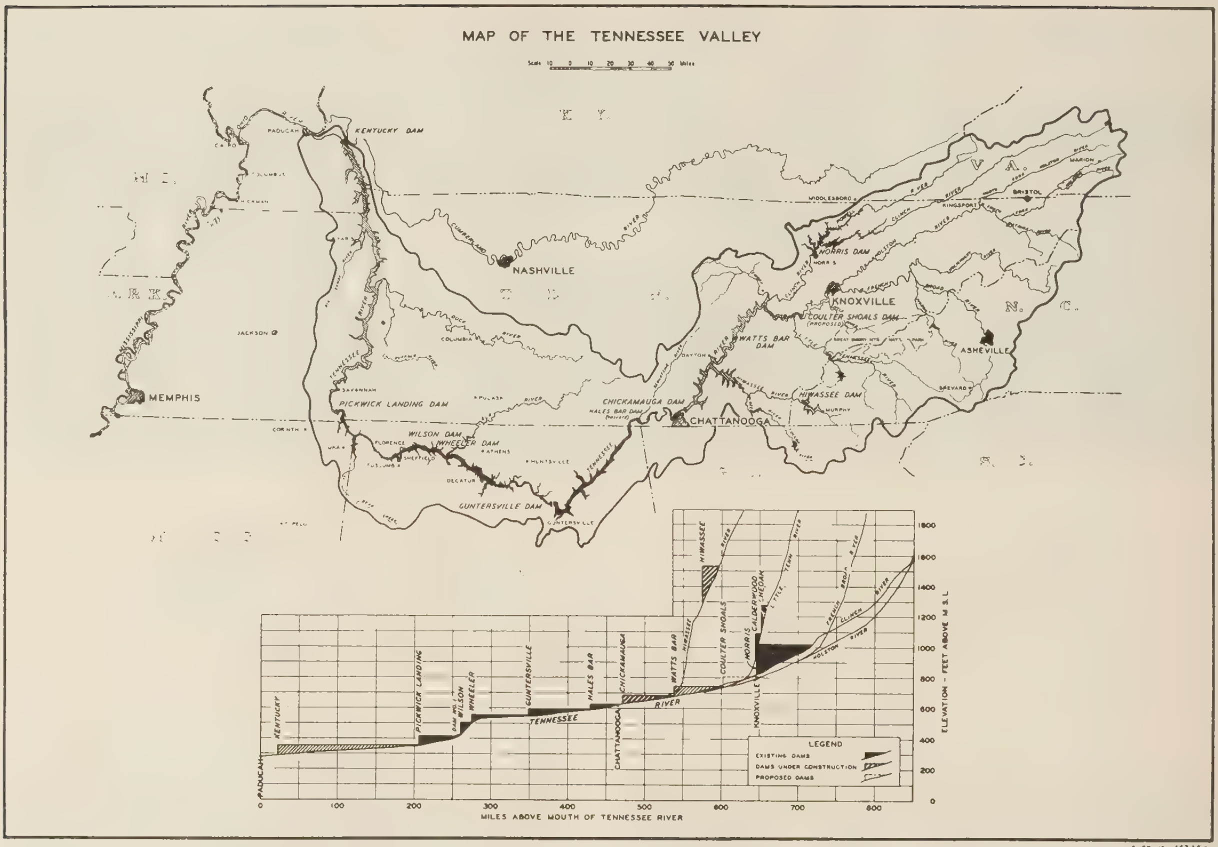

FIG. 2 MAP OF THE TENNESSEE VALLEY

With the regulated flow at each plant established by operation of the reservoir system in the combined interest of flood control, navigation, and power generation, the determination of ultimate plant capacity for any particular project must be approached from the standpoint of the power system as a whole. The method employed has been to determine the ultimate primary or continuous energy output of the entire system of plants, based upon the regulated flow at each site, and to establish the corre-

TABLE 1 PROJECT FEATURES

	Navigation		Reservoir						Power					
Project	Size of Lock Chamber (feet)	Maximum Lift of Lock (feet)	Area at Top of Gates (acres)	Volume at Top of Gates (acre-feet)	Controlled Flood Storage (acre-feet)	Length of Spillway (feet)	Spillway Capacity (second-feet)	Backwater Length (miles)	Rated Head (feet)	Head for Best Efficiency (feet)	Present[4] Plant Capacity (kw)	Ultimate[4] Plant Capacity (kw)	Effective Capacity During High Flow - 1926-1927 Flood (kw)	Type of Turbines
Kentucky[1]	110x600	73	256,000	6,100,000	4,570,000	960	1,100,000	184.4	48	51	--	160,000	74,000	Kaplan
Pickwick Landing	110x600	63	46,[illegible]00	1,091,000	418,000	880	910,000	52.7	43	56	72,000	216,000	36,000	Kaplan
Wilson	60x300[2]	90	16,200	600,000	--	2,212	629,000	15.5	95 & 92	95 & 92	184,000	444,000	419,000	Francis
Wheeler	60x360	53	68,300	1,150,000	429,000	2,400	687,000	74.1	48	48	64,800	259,200	245,000	Propeller
Guntersville	60x360	45	70,700	1,019,000	282,000	720	625,000	82.1	36	37	72,900	97,200	69,000	Kaplan
Hales Bar	60x267	37	5,800	126,000	--	1,200	--	39.8	35	35	50,500	50,500	20,000	Francis
Chickamauga	60x360	58	37,200	655,000	353,000	720	600,000	59.0	36	48	81,000	108,000	67,000	Kaplan
Watts Bar[1]	60x360	70	41,600	1,132,000	370,000	800	550,000	72.4	52	57	90,000	150,000	150,000	Kaplan
Coulter Shoals[1]	60x360	70	13,900	336,000	130,000	800	500,000	50.0	65	70	--	96,000	64,000	Kaplan
Norris	--	--	40,160	2,567,000	2,020,000	300	54,000	72.0	165	180	100,800	100,800	100,000	Francis
Hiwassee	--	--	6,280	438,000	365,000	224	130,000	22.0	190	200	57,600	115,200	115,000	Francis
Blue Ridge[3]	--	--	3,290	197,500	183,000	110	55,000	10.0	147	--	20,000	20,000	20,000	Francis
Ocoee No. 1[3]	--	--	1,380	76,700	25,900	362	--	7.5	110	--	18,000	18,000	18,000	Francis
Ocoee No. 2[3]	--	--	--	--	--	--	--	--	250	--	18,800	28,200	18,000	Francis
Great Falls[3]	--	--	2,2[illegible]0	55,100	49,900	450	150,000	--	142	--	29,400	29,400	29,000	Francis

1 Under construction.
2 Two lock chambers.
3 Acquired by purchase of completed projects; dependable flood storage not fully determined.
4 Generating capacities are based upon actual performance which exceeds guaranteed performance. Table 2 gives guaranteed capacities.

sponding ultimate peak capacity required for the entire system, on the basis of an assumed annual load factor of 60 per cent, together with an allowance of 15 per cent for machine outages and an additional allowance to compensate for the loss of capacity from decreased head at the river plants during extreme floods. Additional capacity at 100 per cent load factor is also provided to carry such high-grade secondary energy as may be available 75 to 80 per cent of the time.

The minimum ultimate capacity assigned to any particular plant must be at least large enough to utilize the entire regulated flow at the site at a constant rate of demand or, in other words, 100 per cent load factor. The additional system peak capacity required for variable load demand is apportioned among the various projects largely in inverse ratio to the unit incremental capacity cost. The general effect of this method of apportionment is to provide peaking capacity at the plants having the higher heads, although consideration must be given to possible pondage limitations and to the reduction of head during extreme floods.

TABLE 2 TURBINE DESIGN DATA

	Guntersville	Chickamauga	Pickwick Landing
Maximum headwater elevation - feet	605	701	430
Normal headwater elevation - feet	594	682	413
Minimum headwater elevation - feet	590	673	408
Normal tailwater elevation - feet	555	635	356
Minimum tailwater elevation - feet	550	628	356
Maximum tailwater elevation - feet	600	697.5	422
Maximum head - feet (net)	42	52	60
Minimum head - feet (net)	5	3.5	5.5
Head for best efficiency and speed - feet	37	48	56
Rated head - feet (net)	36	36	43
Rated horsepower	34,000	36,000	48,000
Maximum horsepower	39,000 at 39 ft	42,000 at 40 ft	55,000 at 47 ft
Generator continuous rating - 60° C, 0.90 power factor - kva	27,000	30,000	40,000
Generator capacity - 60° C, continuous at 0.90 power factor - kw	24,300	27,000	36,000
Rated speed - rpm	69.2	75	81.8
Specific speed at rating - rpm	145	161	163
Head for runaway speed - feet	42	52	60
Runaway speed - rpm	189	218	200
Turbine manufacturer	S. Morgan Smith	Baldwin Southwark	Allis-Chalmers
*Generator manufacturer	General Electric	Allis-Chalmers	Westinghouse
Number of units, present	3	3	2
Number of units, ultimate	4	4	6
Value of sigma at rating	1.12	1.32	0.90
Diameter of runner at throat - inches	265	264	292
Number of blades	5	5	6
Blade adjustment	Automatic oil pressure	Automatic oil pressure	Automatic oil pressure
Rated discharge - cfs at rating	9,500 at 36 ft	10,200 at 36 ft	11,200 at 43 ft
Peripheral coefficiency at rating	1.66	1.79	1.98
Peripheral speed, turbine, runaway fpm	13,100	15,070	14,900
Peripheral speed, turbine, normal fpm	4,810	5,180	6,180
Discharge, coefficient of gates as orifice = C in $CA\sqrt{2gh}$ at rating from model test	.404	.430	.471
Elevation centerline of distributor - feet	558	632	358.583
Elevation centerline of runner - feet	549.875	623.04	349.50
Spacing center to center of units - feet	78'0"	80'0"	80'0"
Weight of rotating element - turbine and generator - pounds	973,000	936,000	917,000
Head for maximum hydraulic thrust - feet	42	52	60
Maximum hydraulic thrust - pounds	914,000	1,052,000	1,500,000
Total load on thrust bearing	1,887,000	1,988,000	2,416,000
Type of thrust bearing	Kingsbury	Kingsbury	Kingsbury
Capacity of thrust bearing - pounds	2,000,000	2,075,000	2,700,000
Type of generator setting	Umbrella	Umbrella	Umbrella
WR^2 of turbine and generator (lb-ft^2)	81,200,000	81,700,000	80,500,000
Type of scroll case	Concrete	Concrete	Concrete
Governor manufacturer	Woodward	Woodward	Allis-Chalmers
Gate servomotor:			
Capacity - ft-lb (oil pressure 300 p.p.s.i.)	368,000	517,000	772,000
Operating pressure - p.p.s.i.	250-300	250-300	250-300
Minimum time to close gates - seconds	8	8	8
Speed droop adjustment	5%	5%	5%
Blade servomotor:			
Capacity - ft-lb (oil pressure 300 p.p.s.i.)	610,000	635,000	887,000
Minimum time to open or close blades - seconds	10	10	10
Maximum time to open or close blades - seconds	40	40	40
Type of draft tube	Elbow	Elbow	Elbow
Splitter	Yes	No	Yes
Velocity through intake trashracks, gross area, fps at rated discharge	4.1	4.2	5.0
Velocity at draft-tube exit - fps at rated discharge	7.1	7.4	7.2
Scroll case:			
Clear width between main piers - feet	66'0"	66'0"	69'0"
Thickness main piers - feet	12'0"	14'0"	11'0"
Number intermediate piers	2	2	2
Thickness intermediate piers	6'6"	6'6"	6'6"
Offset centerline of turbine from centerline of scroll case	5'0"	3'0"	5'3"
Height of intake openings - feet	43'0"	46'0"	40'0"
Draft tube:			
Clear width between main piers	66'0"	66'0"	69'0"
Thickness main piers	12'0"	14'0"	11'0"
Number intermediate piers	2	2	2
Thickness intermediate piers	6'0"	6'0"	6'6"
Elevation lowest point of tube	497	572	294
Horizontal length of draft tube	85'0"	85'0"	85'0"
Height of draft-tube openings - feet	24.54	25.54	23.0

*All generators 3-phase, 60-cycle, 13,800-volt

Note: Generating capacities listed in this table are guaranteed capacities. The actual capacities obtained in operation are given in table 1.

Selection of Turbines

In the integrated power system, the Pickwick Landing, Guntersville, and Chickamauga plants will be operated at high capacity factor during periods of ample flow in the main river, and the tributary plants at Norris and Hiwassee will, as a general rule, be operated intermittently. Conversely, during periods of low flow in the main river, the tributary plants will operate at high capacity factor, while main river plants like Pickwick Landing, Guntersville, and Chickamauga will be assigned to service at relatively low capacity factor. As shown in Table 2, the generating units for these plants will be required to operate over a great range of load and head conditions; and, since the maximum head in all cases is less than 60 ft, movable-blade propeller turbines of the Kaplan type, with their characteristic high efficiency over a wide gate range, are ideally suitable.

In the case of Chickamauga, which is typical of the run-of-river plants, the estimated continuous power available from the regulated flow is about 50,000 kw, and the ultimate installation to meet system-capacity requirements about 100,000 kw. In view of the immediate power-market conditions and present and future operating characteristics, it was decided that an ultimate installation of four 25,000-kw units, with three units installed initially, would afford the required degree of flexibility. In field operation the turbines exceed the guaranteed ratings so that the four units will have an actual capacity of 108,000 kw.

Referring again to Table 2 and Fig. 3, the basic requirement of the turbine purchase specification is a machine capable of 36,000 hp at a head of 36 ft, or during flood periods, with best efficiency and speed selected for 48 ft head, which obtains during the major part of the period of normal operation. With reference to cavitation, the bidder was required to state guaranteed horsepower outputs for the schedule of headwater and tailwater elevations given in Table 3.

At an early stage of the project design and well in advance of inviting bids, the manufacturers were given complete information pertaining to head and tailwater elevations and other governing physical features, and were requested to comment on pre-

TABLE 3 CAVITATION AND EFFICIENCY TESTS AT BALDWIN LOCOMOTIVE WORKS OF CHICKAMAUGA TURBINE

(Runner, 5-blade, 264 in. 75 rpm; 11-in. model runner for cavitation tests; 16-in. model runner for efficiency tests)

Tailwater Elevation	Headwater Elevation	Head Feet	Guaranteed Horsepower	Allowable Horsepower from Cavitation Tests	Allowable Horsepower from Model Efficiency Tests
630	674	44	42,000	44,100	49,500
630	678	48	42,000	45,900	56,300
630	682	52	42,000	48,100	63,300
634	674	40	40,750	41,500	42,600
634	678	44	42,000	46,500	49,500
634	682	48	42,000	49,900	56,300
638	672	34	33,200	35,100	35,500
638	677	39	40,000	42,000	41,300
638	682	44	42,000	48,000	49,500

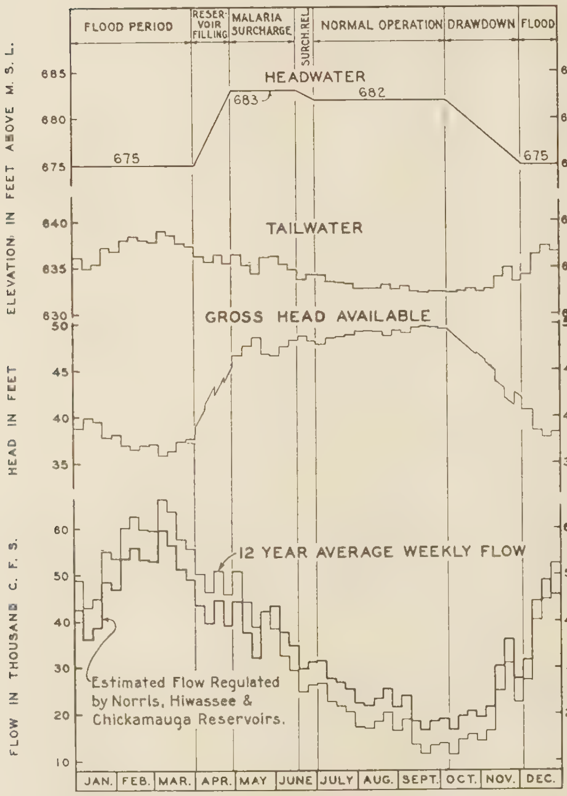

FIG. 3 RESERVOIR OPERATION DIAGRAM OF CHICKAMAUGA PROJECT

liminary turbine specifications. By thus working in close co-operation with the turbine manufacturers, it was found possible definitely to specify the speed of operation, the elevation of the center line of the distributor, the unit spacing, and the depth of draft tube, so as to afford a fixed common basis for bidding, and yet allow each bidder reasonable latitude for employing his own characteristic design.

Model Testing

To avoid the difficulty of measuring large prototype discharges and to insure an ample margin of safety against excessive cavitation, acceptance of the turbines with reference to efficiency and cavitation was based upon laboratory tests of homologous model runners complete with homologous scroll cases and draft tubes. Acceptance with respect to capacity was based upon actual prototype performance.

The Moody formula was employed to compute the efficiency of the prototype turbines from the model-test results

$$\mathrm{Eff}_2 = 100 - (100 - \mathrm{Eff}_1)\frac{(D_1)^{1/4}}{(D_2)}\;\frac{(H_1)^{0.01}}{(H_2)}$$

D_2 = diameter of prototype runner
D_1 = diameter of model runner
H_2 = head on prototype turbine
H_1 = head on model turbine
Eff_2 = efficiency of prototype turbine
Eff_1 = efficiency of model turbine

The correction for head difference is comparatively unimportant and has been omitted from specifications for turbines for plants now under construction. The correction for diameter is of governing importance and, in the case of the 16-in. model runners for the 264-in. Chickamauga turbines, converts a model efficiency of 87 per cent to a prototype efficiency of 93.6 per cent. The accepted method of employing the formula is to compute the correction for the point of maximum efficiency of the model and to add this single percentage to all the model-test results.

Table 3 is abstracted from the cavitation-test results of the Chickamauga model tests conducted at the laboratory of the Baldwin Locomotive Works. It will be noted that cavitation is a limiting factor at the higher heads for which the tailwater is relatively low, but that, at the lower heads and higher tailwater elevations, the capacity is limited solely by the ability of the runner to deliver power.

Power-Station Arrangement

Fig. 4 shows the general cross section of the Guntersville power station, which is typical for all three Kaplan plants. Because of low entrance velocities and care taken to inhibit the formation of eddies, the corresponding head losses are relatively small; and the intake structure is comparatively short and simple, designed essentially to give proper direction to the flow filaments entering the scroll case. On the other hand, the velocity of exit from the runner is relatively high; a considerable proportion of the total available energy still remains in the discharge leaving the wheel; and for proper efficiency a long draft tube is necessary to regain this energy. Under the hydraulic requirements mentioned, structural economy is readily obtained by designing the intake and draft-tube substructure as a monolith and utilizing their combined mass to sustain the hydrostatic load.

Particular attention has been given to the economic elevation of turbine runner. Minimum turbine and generator costs are obtained by setting the turbine runner so far below minimum tailwater that the wheel diameter is no longer limited by cavitation requirements, but is established solely by the maximum power that the turbine runner is capable of delivering. However, except for the case in which solid rock is covered by an exceptional depth of overburden, minimum structural costs result from keeping the runner as high as possible with respect to tailwater, so as to reduce the volume of rock excavation, the hydrostatic load on the structure, and the yardage of concrete required for stability. Since the size of waterways is a function solely of the discharge necessary to produce the rated power at the rated head, variations in wheel diameter to meet the cavitation requirements, corresponding to various assumed runner elevations, need cause no change in unit spacing, but merely affect the diameter of the upper portion of the draft tube in the vicinity of the runner. In case the manufacturer does not have model-test results available for the desired ratio of diameter of throat ring to size of draft-tube water passages, advance model tests at the purchaser's expense may be warranted. Another consideration is that the larger wheels set at higher elevation have in-

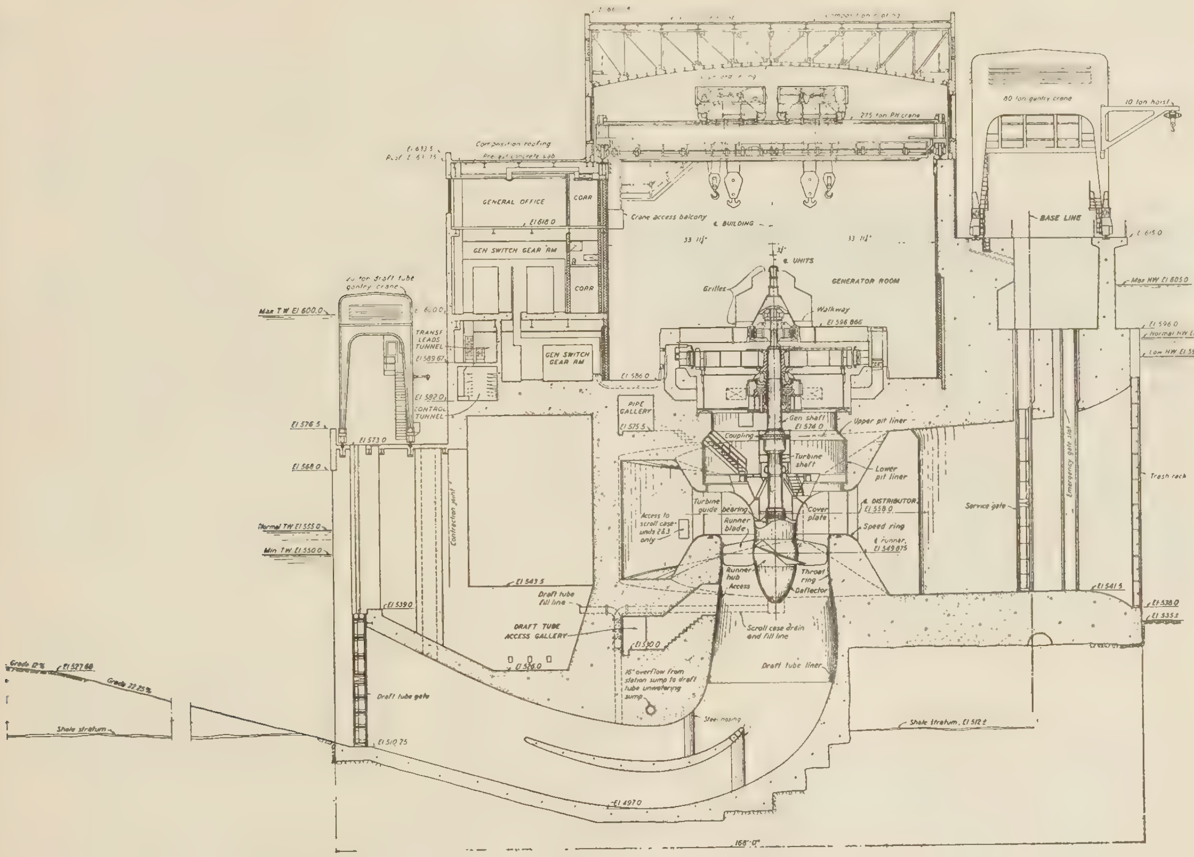

FIG. 4 CROSS SECTION OF GUNTERSVILLE POWER STATION
(Turbine capacity: 34,000 hp, 36 ft head, 69.2 rpm. Generator capacity: 27,000 kva, 13,8000 volts, 3 phase, 60 cycles.)

creased capacity at reduced head during flood conditions. Obviously, no sweeping generalization can be made, and the economic elevation of runner setting must be studied in close cooperation with the manufacturers for each particular plant so as to yield the highest ratio of net annual return to annual fixed charges.

The adopted disposition of mechanical and electrical auxiliaries is believed to utilize the limited space immediately adjacent to the generating units to best advantage so as to reduce the size of service bay to a minimum. The electrical bay is located downstream from the generator room where a foundation is provided by the long piers of the draft tube. For maximum convenience in operation, and to give the shortest and simplest arrangement of high-pressure oil piping to the Kaplan head and the wicket-gate servomotors, the governor-actuator cabinets of the duplex type are located on the main floor of the generator room between the companion pair of generating units. With the governor actuators on the main floor, space remains in the main draft-tube piers, at the elevation of the inspection tunnel where there would otherwise be excess concrete, for a compact arrangement of draft-tube unwatering pumps and operating valves. This location of unwatering pumps in the main piers between units not only eliminates space which would otherwise be required in the service bay, but also materially reduces the length of suction lines and friction-head loss in unwatering the draft tubes.

Because of the characteristic location of Kaplan runners from 10 to 12 ft below average low tailwater, and the necessity of providing ready facilities for inspection and maintenance, the draft-tube unwatering system, including draft-tube stop logs and a gantry crane, represents in itself a sizable investment. The unwatering pumps for each plant have an aggregate rated capacity of 10,000 gpm at rated head and an actual capacity of about 16,000 gpm under average conditions. The pumps are the deep-well type with low-level runners, so as not to require priming, and are capable of unwatering the draft tube completely in from 1 to 2 hr.

TURBINE DESIGN AND CONSTRUCTION

In accordance with general practice, turbine contracts of the Tennessee Valley Authority stipulate that the manufacturer shall prepare and be responsible for the design, including the determination of scroll case and draft-tube waterways, in conformity with the governing physical conditions and general requirements enumerated in the purchase specification. The authors will describe briefly, from the standpoint of the purchaser's engineers, certain of the design elements covered in the specification, with the request that the manufacturers' engineers amplify the treatment of design and construction features in the subsequent discussion.

Welded Construction. In view of the increasingly successful use of welded-plate construction in the heavy-machinery industry, the Authority's specifications afforded bidders the option of employing either cast-steel or welded-plate construction for the speed ring, head cover, lower guide-vane ring, discharge ring, gate-shifting ring, and wicket gates. Plate construction for the draft-tube liner and the upper pit liner is, of course, standard practice. The specifications required that the design of welded joints and con-

nections and the fabrication of welded-steel parts conform to the Boiler Construction Code of The American Society of Mechanical Engineers, Section VIII, for Unfired Pressure Vessels. It was also stipulated that the welding conform to paragraph U-69 of the same code, and be stress-relieved, excepting the draft-tube liner, upper pit liner, and minor details, welding for which was specified to conform to paragraph U-70. The contractor for the Guntersville

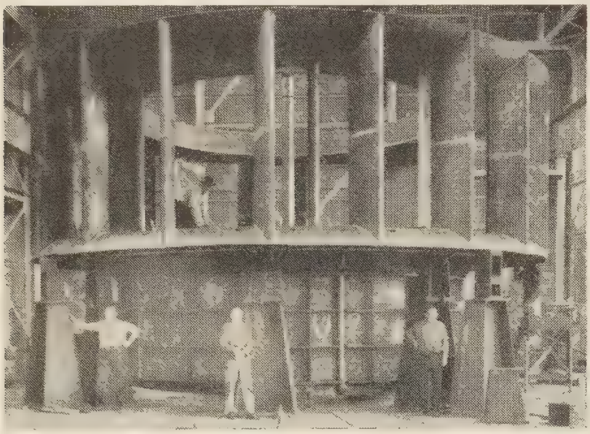

Fig. 5 Welded Speed Ring; Guntersville Project

turbines, the S. Morgan Smith Company, employed welded construction very extensively and with very satisfactory results. Fig. 5 shows the welded speed ring of the Guntersville turbine.

Turbine Shaft. Fig. 6, showing the cross section of the Chickamauga turbines, is fairly typical of all three plants. Subsequent to the design of the Pickwick Landing machines, improved facilities at the steel-forging plants made it practicable to specify that the upper end of the turbine shaft be enlarged in forging so as to form the operating cylinder for adjustment of the runner blades, and flanged at the upper end for coupling with the generator shaft. The lower flange of the generator shaft forms the upper cover of the blade servomotor cylinder. This particular feature permitted the elimination of two large steel castings for the servomotor, which were formerly interposed between the turbine and generator shafts, and two corresponding intermediate flanged bolted joints, leaving only a single flanged bolted coupling to be gasketed tight against possible oil leakage. The simplified detail has proved entirely satisfactory and much superior during shop assembly and field erection and alignment. The bottom connection of the oil-pressure-supply lines, extending inside the generator shaft from the Kaplan head above the generator down to the blade-shifting servomotor, is made by screwed, rather than by flanged fittings, which were formerly used for the purpose and required that the coupling between the turbine and generator shafts be opened 10 or 12 in. to permit making the connection. The screwed fitting permits insertion or removal of the oil-supply pipes from above without dismantling the coupling.

Turbine Guide Bearings. The main turbine guide bearings are the water-lubricated adjustable type with shoes of lignum vitae or a molded plastic material, such as bakelite or Insurok, while the corresponding shaft sleeves are a special corrosion-resistant steel. Water-lubricated bearings were selected in preference to oil-lubricated bearings for three principal reasons: (1) Optimum location of the bearing for its primary function of support, immediately adjacent to the turbine runner with no stuffing box and oil chamber interposed between the runner and bearing; (2) most accessible and convenient location of stuffing box above the guide bearing, where adjustments can be made while the unit is in operation; and (3) elimination of the hazard of burning out the bearing due either to the leakage of water past the stuffing box or flooding of the turbine pit during the rather frequent condition of high tailwater.

In general, the problem of providing adequate lateral support is much more acute for propeller runners than for Francis wheels, owing principally to the relatively greater diameter and weight due to lower heads, and to the inherently greater hydraulic instability under low-gate conditions. It is characteristic of Francis wheels that the runner is located in elevation at about the center line of the distributor; and, consequently, the oil-lubricated type of bearings for such machines may be located above the head cover in a relatively free-draining position without involving an excessive distance between the bearing and the runner to be supported. On the other hand, because of the characteristic design of propeller-type turbines, in which the runner is placed 6 or 8 ft below the center line of the distributor, the oil-lubricated type of bearing, if adopted, must be placed in a relatively small, conical chamber down below the head cover in order to keep the distance between the bearing and the runner to be supported within acceptable limits. In the event of a defective stuffing box, even a low rate of leakage might soon fill the surrounding space and cause the bearing to burn out.

Runner-Hub Lubrication. Correct lubrication of the mechanism within the Kaplan runner hub, which is continuously submerged under a head of from 10 to 60 ft, is an important element of design. The runner-blade trunnions, operating in bronze bushings under bearing pressures of between 2000 and 3000 psi, must be provided with an unfailing film of the proper lubricating oil, uncontaminated by grit and free from any appreciable amount of water. In addition, the hub oil has a second essential function as a protective coating and inhibitor of corrosion for the many steel surfaces of the internal mechanism.

By means of a revolving oil chamber, located on the main shaft just below the generator coupling, a continuous oil pressure of from 8 to 10 psi is maintained, sufficient to insure a steady supply to the trunnion journals. The revolving reservoir conserves an appreciable volume of oil displaced during each downward movement of the main operating shaft into the runner hub.

The runner-blade trunnions are sealed by means of a chevron type of packing, retained in the packing space under pressure between two stainless-steel rings. The arrangement is adequate to prevent either the infiltration of water and grit along the blade trunnions during high tailwater conditions or the loss of oil from inside the hub during operating conditions of high vacuum adjacent to the blades. Leakage of oil from the hub is restricted on the average to about $^1/_4$ gal per day, and the design is so rugged that a tight seal is assured for several years of operation. The revolving reservoir is equipped with a sight-gage glass, by means of which, when the unit is stopped, the quantity of oil in the runner hub can be immediately determined and make-up oil added, if necessary.

Simple and effective means are provided for draining any accumulation of leakage water which would, if neglected, endanger the safety of the trunnion bearings and cause corrosion of the operating mechanism. Because it is heavier than the lubricating oil, any leakage of water along the trunnions will collect in the bottom of the hub when the unit is stopped. In the case of the Guntersville design, which is representative in principle for all three manufacturers, drainage of water from the bottom of the hub, under the static head of oil in the annular reservoir, is accomplished by drilling out the center of the blade-operating shaft so as to form a pipe communicating with the bottom of the hub and terminating at the upper end in a test connection, readily accessible at the exterior surface of the turbine main shaft above the stuffing box

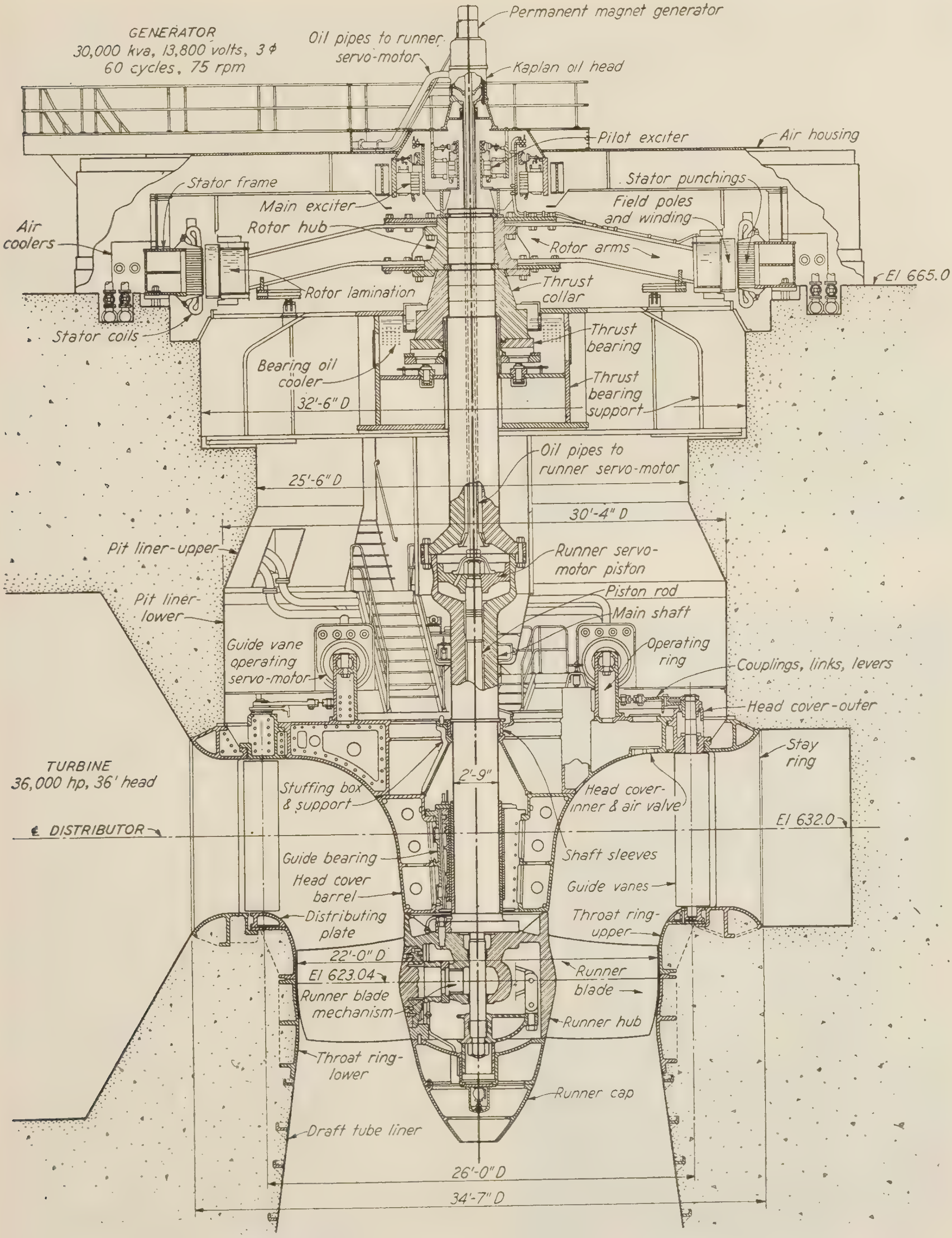

Fig. 6 Cross Section of Chickamauga Turbine

Alignment of Turbine and Generator Shafts. In conformity with the purchaser's specifications, turbine and generator shafts, complete with thrust-bearing collars and generator rotor hubs, are fitted, coupled, and given a rotating-alignment check at the manufacturer's plant prior to shipment to insure that the combined shafts are straight and that the face of the thrust collar lies in a plane perpendicular to the combined-shaft axis.

Before operation of the unit, a field alignment check of the entire rotating assembly of the turbine and generator is made to demonstrate that the axis of rotation of the combined shafts does not depart from the vertical more than 0.003 in. at any point, also that the maximum diameter of circle described by any point on the shaft in rotating about that axis is not more than 0.01 in. This so-called rotation check is made with all turbine and generator guide bearings backed out, and with the entire rotating assembly simply suspended from the thrust bearing. The rotating assembly is turned, by means of a block-and-tackle arrangement attached to the rotor spokes, through four intervals of 90 deg each and back to the original position. At each position, micrometer tram measurements are made between the turbine and generator shafts and four fixed suspended plumb wires. Any corrections necessary to plumb the shafts with respect to the vertical can readily be made either by the adjusting studs under the Kingsbury bearing shoes or by the insertion of adjusting plates under the thrust-bearing supporting beams. If the throw of the shafts exceeds 0.01 in. on the diameter at any point, indicating that the thrust collar is excessively out of perpendicular to the shaft axis, the requisite field adjustments are more difficult; and, consequently, every check should be made at the factory to insure that this feature is acceptable before shipment.

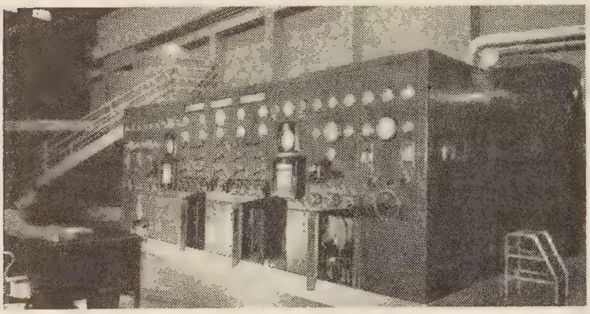

Fig. 7 Governor-Actuator Cabinet; Pickwick Landing Project

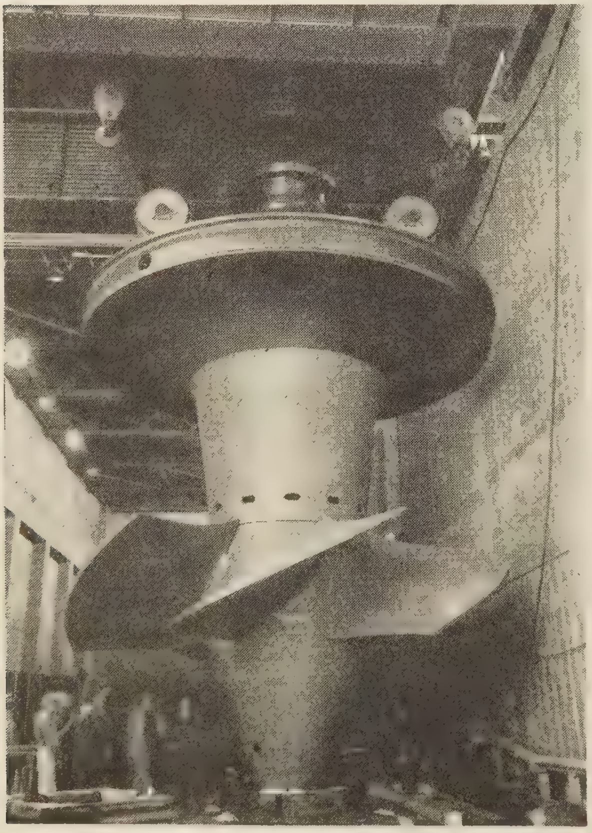

Fig. 8 Head Cover and Runner Assembly; Pickwick Landing

Governors. The governors are of the cabinet actuator type with the entire mechanism, including the governor-flyball elements, relay valves, motor-driven oil pumps, Kaplan valves, and Kaplan controls, enclosed in a compact cabinet, the base of which forms the sump tank. The governor controls and the requisite gages and temperature and pressure indicators are mounted upon the front face of the cabinet.

In the Guntersville and Chickamauga plants, where three units have been installed initially, duplex actuators are provided in a single combined cabinet located between the units. Owing to the necessity of providing clearance at the temporary end wall of the power station, individual cabinet actuators are provided for the third units and planned for the fourth units. At the Pickwick Landing plant two units were installed initially and provided with a duplex actuator.

The governors are equipped with motor-driven flyballs, gear-type motor-driven oil pumps, cable-type restoring mechanisms, and welded-steel oil piping with flanged connections only where necessary for ease in assembling and dismantling. The operating oil pressure is between 275 and 300 psi. The oil piping and restoring cables to the wicket-gate servomotor in the turbine pit are located in floor trenches entering the cabinet from below. The piping and restoring cables to the Kaplan head above the generator are located beneath the generator walkways and enter the actuator cabinet from above.

Conclusion

Experience at the Pickwick Landing, Guntersville, and Chickamauga plants indicates that Kaplan turbines in the range of physical size from 22 to 24 ft may be satisfactorily designed, constructed, and erected to meet the exacting demands of variable head and load service. Engineers are frequently confronted with the question of whether still larger units might be economical. There are many obstacles to increased size, such as the difficulty of obtaining large steel castings for runner hubs and blades without a prohibitive percentage of rejections, and the difficulty of designing large complicated concrete substructures to withstand the attendant indeterminate shrinkage stresses without consuming a prohibitive interval of time for dissipation of setting heat during construction operations. It also appears that, although the practical limit of refinement in machine-shop tolerances for such large heavy machinery has already been reached, still further refinement would be necessary in the case of larger machines to obtain, during field erection, the degree of accuracy in turbine and generator-shaft alignment required for the satisfactory operation of such large heavy rotating masses. In the opinion of the authors, the Pickwick Landing, Guntersville, and Chickamauga machines approach the maximum practicable size of Kaplan turbines which may be fabricated and erected with present facilities.

Discussion

J. M. Mousson.[3] The authors state that efficiency tests on the prototype units are not contemplated to avoid the difficulty of measuring large unit discharges and that the step-up, based on laboratory tests, is being used as a criterion for the acceptability of the units. It would appear, however, that the reliance on the Moody step-up formula exclusively is not a guarantee for proper prototype performance because: (1) It was developed based on experience with Francis runners. (2) It rests exclusively on one manufacturer's laboratory conditions and may not be applicable unconditionally to those of others. (3) Even under favorable conditions field experience thus far gained with propeller-type units of the fixed-blade or adjustable type indicates that no further increase in step-up may be expected above 150 in. runner diam. Some engineers even contend that with these types of turbines a gradual decrease in step-up could be expected above the maximum occurring at about 150 in. diam.

It is believed that the complete negation of prototype testing is a serious handicap to progress in the art, particularly serious when this viewpoint is held by engineers associated with such a vast enterprise as the T.V.A. No one will contend that the period of development of the propeller-type turbine is over and, therefore, continuous and substantial efforts are yet to be made and are imperative to achieve efficiencies closer to the ideal.

In addition, it must be emphasized that the over-all costs per horsepower installed are an important factor. Certain expensive features adopted by one or the other turbine manufacturer remain yet to be justified from an economic point of view. However, no justification can be obtained without prototype testing. This was ably pointed out some time ago by one of the authors' associates[4] to the effect that the economics of draft-tube splitters should be carefully investigated and that there was a real opportunity as well as a necessity to do so with the completion of Wheeler and Pickwick Landing Dams. This opportunity is now even more striking with Guntersville and Chickamauga in operation, both plants having Kaplan turbines of nearly identical dimensions but only the draft tubes of the former development are provided with draft-tube splitters.

A recent feature of Kaplan turbines, referred to in the paper, is the adoption of cables for the restoring mechanisms instead of the rigid rods previously used. It would be interesting to know whether or not the prestressed cables stretch appreciably in service and, if so, what the permanent elongating characteristics are with relation to time.

Based on past experience, the correct cam design controlling the gate-blade relation cannot be obtained through model testing and some form of index testing is required on the prototype units. It would be of value to know what types of index method are being employed on the various installations of the T.V.A.

F. Nagler.[5] The authors present comprehensively significant engineering data which cannot help but answer questions occurring to many engineers interested in hydroelectric power.

It would be of interest if the authors would comment a little further on Table 3. It is not evident whether the tests for cavitation were made at the heads listed in the third column, or whether the cavitation coefficients were arrived at by testing the models at some other heads and simply applying the results so obtained. It is noted that the authors state the head difference, in connection with the application of the Moody formula, is comparatively unimportant, from which it might be assumed that the head differences for the cavitation tests may closely approximate those to be experienced in the powerhouse.

The writer would like to ask whether some of the hydraulic-thrust figures shown in Table 2 have been checked by field measurements. The writer has quite a collection of such data, obtained by calibrating the deflection of the bridge by means of the fairly well-known weight of rotor, shaft, and runner, and then using this calibrated bridge to determine the additional hydraulic thrust. If such figures are available for any one of the three plants shown in Table 2, they would be of considerable interest.

The care taken by the engineers of the T.V.A. in prechecking the turbine and generator shafts for alignment is particularly noteworthy. This seems to be an increasingly desirable practice. Did the authors find that shop tests showed comparable straightness and truth with the rotational check made in the field?

It would also be of interest to have further comment on, or comparison of, the cable-type restoring mechanisms with the older torsional-shaft types. They seem to possess such advantages in cheapness and flexibility of installation that any operating disadvantages observed should be brought out.

R. E. B. Sharp.[6] The optional use of either cast or welded-plate steel as the material for the speed or stay ring, as well as for the other parts mentioned by the authors, is a sensible provision in turbine specifications. The turbine manufacturer is thus permitted to take advantage of his preferred design and shop practice and in some cases to improve deliveries by the use of that material most readily available. The use of plate steel is preferable for those parts of the water passages subject to cavitation, such as the throat or discharge ring, due to its greater resistance to this action. Runner blades of steel plate, welded to a skeleton frame, would undoubtedly resist the cavitating action better than cast-steel blades, but the matter of strength and cost would be formidable problems.

As brought out by the authors, the use of a water-lubricated bearing results in a minimum amount of overhang of the runner below the bearing. This is a desirable feature as affecting the critical speed of the shaft. With both runner and generator rotor overhung beyond the only two guide bearings provided, it is essential that these calculated deflections be a minimum.[7]

The actual amount of hydraulic thrust on the turbine runner can be readily determined in the field, by application of the principle that the deflection of the beams supporting the thrust bearing is proportional to the load. The deflection due to the known total weight of the revolving parts is measured with the turbine shut down, and again when in operation. The relation between the hydraulic thrust so measured, the runner-blade pitch at various radii, and the power developed can be used as a check on the effective-flow distribution through the runner.

The intermittent turning of the runner blades under load through only a small percentage of a total revolution, with frequent periods without movement, prevents the maintenance of an effective oil film at the loaded portions of the blade bearings, with the result that a high coefficient of friction must be overcome by the blade servomotor, even though the runner hub is filled with oil. This coefficient is undoubtedly lower than it would be were the load on the blades not of a very live nature with some vibration to permit to some extent the seepage of an oil film into the desired locations.

[3] Hydraulic Engineer, Safe Harbor Water Power Corporation, Baltimore, Md. Mem. A.S.M.E.

[4] "Economic Aspects of Energy Generation," a Symposium, Trans. A.S.C.E., vol. 104, 1939, pp. 942–1008; discussion by R. M. Riegel, pp. 1014–1015.

[5] Chief Engineer, Canadian Allis-Chalmers, Ltd., Toronto, Canada. Life Member A.S.M.E.

[6] Chief Engineer, I. P. Morris Dept., Baldwin Southwark Divsion, The Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

[7] "Lateral Vibration of Shafts," by L. F. Moody, *Product Engineering*, vol. 6, Feb., 1935, pp. 57–60; March, pp. 98–100; April, pp. 142–143.

The ideal method of lubrication with bushed bearings, such as are used, would be to have a continuously operating high-pressure oil pump which would provide an oil film at the desired points. The mechanical complications of such an arrangement with the possibility of derangement and consequent loss of such an oil film have so far militated against the use of such a feature.

The use of antifriction bearings in the hub, of the roller or ball type, would greatly reduce the friction load. The lack of adequate bearings of this type of stainless steel, such as would be necessary to take care of possible water entrance into the hub, has been a deterrent in the adoption of this type of bearing. The nature of the live load on the blades, in conjunction with only a partial revolution with long periods at one position, and a consequent Brinelling action of the balls or rollers on the races, is an undesirable feature. While Terry has adopted with success this type of bearing in his design of automatic adjustable-blade runners, the writer is of the opinion that additional successful service experience is necessary to justify this type of bearing, particularly in view of the major operation required in the renewal of such bearings, involving the removal of the runner from the unit. On the other hand, in spite of the high coefficient of friction with bronze-bushed bearings of the type which have become standard in Kaplan runners, no bearing renewals have yet become necessary to the writer's knowledge although, in some instances, difficulty has been encountered due to the bushing turning with the blade shank in the runner hub. To prevent this action from occurring, it is good practice, by the use of dry ice, to shrink the bushing into the hub and retaining rings with resulting actual knitting of the bushing into the surrounding cast steel, which much more effectively prevents relative motion than any press fit. The writer believes that the use of molded plastic material for the bushings in the runner hubs, with some reduced friction coefficient, has possibilities, as this material in certain instances has been found superior to bronze in its ability to withstand high bearing loads.

Authors' Closure

In connection with Mr. Mousson's discussion, it should be emphasized that the remarks of the authors with reference to model testing were purposely limited to commercial acceptance tests for Kaplan turbines. The results of research activities, including liberal use of prototype testing to explore debatable features of economic design, will be made available at some future date.

The authors are under no delusion that the Moody formula is an instrument of extreme precision, but simply take the position that, in the present state of the art of current-meter gaging and the attendant possibility for interminable controversy, advance model testing furnishes a practical workable device having general commercial acceptability as a basis for contract. In this connection it is pertinent that manufacturers are not yet prepared to guarantee Kaplan-turbine efficiencies in excess of the order of 89 per cent. Whether or not the step-up relation holds within narrow limits is not material, because the tests exceed the guarantees by a comparatively wide margin; for instance, model tests on a 16-inch runner show an efficiency of 87 per cent, which steps up to $93^1/_2$ per cent for a 22-ft prototype, exceeding the guaranteed efficiency of $89^1/_2$ per cent by a 4 per cent margin. Conceding Mr. Mousson's statement that the Moody relation is not dependable for prototype diameters exceeding 150 in. (although the authors know of no concrete evidence to support such a contention), it is reassuring to note that a 38-in. prototype of the 16-in. model would have a Moody efficiency of $89^1/_2$ per cent, and a 150-in. prototype would show $92^1/_2$ per cent, which exceeds the guarantee by 3 per cent. The point is that, under the present range of contract efficiencies which are available to the purchaser of Kaplan wheels, absolute refinement in stepping up efficiency acceptance tests is of merely academic interest.

TABLE 4 HYDRAULIC THRUST DETERMINATIONS

	Wheeler	Guntersville	Chickamauga
Dead load: rotor, shaft, and runner, lb...	803,000	973,000	936,000
Deflection, generator bridge, dead load, in..	0.018	0.031	0.028
Maximum deflection when operating, in...	0.042	0.053	0.053
Hydraulic thrust, computed from deflection, lb..............................	1,070,000	687,000	836,000
Hydraulic thrust, estimated by manufacturer, lb.............................	1,190,000	914,000	1,052,000
Gross head during tests, ft..............	46.7	42.6	49
Maximum head assumed for thrust estimate, ft............................	52	42	52
Gate opening for maximum thrust, per cent..............................	100	60	40
Blade tilt for maximum thrust, per cent....	Fixed	55	5

Mr. Mousson asks what amount of stretch has been found in the cable-type restoring mechanisms for Kaplan turbines. Over the first year of operation the permanent elongation or stretch appears to be in the neighborhood of $^1/_{16}$ in., which is insignificant in a 40-ft length with a movement of 24 to 36 in. Readjustment is very simple and requires about two minutes. There is some elasticity to these cables, and tests indicate that they may fail to show gate movements of 0.2 of 1 per cent and less.

To determine the proper shape of the cam controlling the gate-blade relation, Winter-Kennedy-type taps are used, readings being taken with two or even three sets of taps giving different coefficients, at five to seven different blade tilts and at least six or seven different gate openings, resulting in 30 to 36 test points at each head. From these data, by plotting curves of $KW/D^{1/2}$, the optimum blade angle for each gate opening can be readily established, independent sets of Winter-Kennedy points being used to confirm this determination. While different sets of taps sometimes show slightly different characteristics, the corresponding determinations of the proper blade angle usually coincide within very narrow limits.

Referring to Mr. Nagler's comments, the cavitation tests shown in Table 3 of the paper were made at the new cavitation laboratory of the Baldwin Southwark Corporation, with heads varying from 15 to 27 ft, averaging about 20 ft. The cavitation coefficients so obtained were applied without correction to the prototype conditions. The model efficiency tests were conducted under even lower heads, between 2 and 4 ft, and the corrections for head in the Moody formula were neglected in stepping up these efficiencies. The horsepower of the model was computed directly, no correction being made for increased efficiency.

The hydraulic-thrust figures contained in Table 4 have been obtained by measuring generator bridge deflections with dead load and with hydraulic thrust.

Shop checks on the individual and the combined shafts have been reliable, especially for checking coupling alignment. However, with generator construction such as used at Guntersville and Chickamauga, where the generator thrust collar is a separate piece, shrunk and keyed to the generator shaft, several cases have occurred which indicate that these collars shift under load, sometimes throwing the rotating parts out of true position. The writers now specify integrally forged thrust collars as the best assurance against such type of misalignment.

Mr. Sharp's suggestion of the possibility of using some form of molded plastic material for the runner hub bushings is interesting, especially if it would permit water lubrication. Maintaining oiltight seals around the runner-blade trunnions, with provision for refilling with oil, and draining out any infiltrating water add considerably to the cost and complicates the design. With molded plastic bearings and either bronze or stainless sleeves provided on the trunnions, the hub could remain full of water, thus reducing the cost and possibly eliminating the use of oil, particularly in the smaller sizes of units.

The authors wish to thank those who have discussed the paper for their valued comment.

A Study of the Development of Skill During Performance of a Factory Operation

By RALPH M. BARNES,[1] IOWA CITY, IOWA, and J. S. PERKINS,[2] CHICAGO, ILL.

While in general the many studies into the nature of skill have been concerned with the total time required to accomplish a given task and the influence on time values of varying conditions pertaining to a specific operation, this paper is primarily devoted to a time study of the elements entering into the performance of an industrial task. The investigation constitutes a pioneer effort to study the effect of practice on a typical factory operation, conducted under laboratory conditions.

The work was undertaken jointly by the Western Electric Company and the University of Iowa. The equipment was made, the tests were run, and the data were compiled in the industrial engineering laboratory at the University of Iowa. The statistical work, tabulation of results, and other activities incidental to the preparation of the final report were handled in the offices of the Western Electric Company.

Introduction

OF CURRENT and outstanding interest in the industrial world of today are those problems related to the teaching, the acquisition, and the measurement of skill. The importance of the subject is reflected in the many studies which educational institutions, industrial organizations, and psychological laboratories have made relative to the nature of skill.

All of these investigations are concerned with the time required to accomplish a specific task and with the manner in which varying conditions can affect such time values. However, practically all investigations to date have been concerned with the total or cycle time required for performance of a specific task, and have given only limited attention to the time required for performing the various therbligs[3] of which any task consists. The present investigation has taken this further step, and has been directed primarily at study of the change in therblig time values resulting from practice in performing an industrial task.

The various aspects of the study on which information was sought are as follows:

1 To study the effect of practice on a typical factory operation carried on under laboratory conditions.

2 To study the learning curves of the various elements of the operation as they were performed by each of the different subjects.

3 To study the consistency between subjects in learning the same element.

4 To study the effects of "speeding" and "soldiering."

5 To study dispersion and its relation to the average performance time.

6 To study the effects of fumbling on the normal learning curve.

7 To study the several ways in which a transport load and pre-position element was performed.

8 To examine the effectiveness of several rating techniques on data of known quality.

9 Finally, to study the effect of practice on the relation of eye movements to hand motion.[4]

Transition From Purpose to Method

In order to study the effects of learning on the elements of an operation or industrial motion cycle, it was regarded as most feasible to set up under laboratory conditions the operation of feeding parts to a punch press. This motion pattern was selected primarily because:

1 It is an extremely common motion pattern in industry, being very similar to punch-press work, as well as to other operations such as feeding parts to tapping machines.

2 It is short and thereby offers the opportunity for a high degree of learning in comparatively little time.

3 It is a complex operation requiring coordinated action of both hands, the eyes, and one foot (for pressing a pedal). This complexity offers opportunity to study the acquisition of skill over a variety of elements.

The part being fed into the punch press was a relay spring, Fig. 1.

The operation was performed by grasping a relay spring from the supply tray by the left hand. The left hand then turns the spring so that it may be grasped properly by a pair of tweezers. The tweezers are held in the right hand and, accordingly, then the part is passed from the left hand to the tweezers in the right hand. The right hand locates the part in the die, releases hold of the part, and returns for another part. As the right hand reaches for the next part, a pedal is pressed which ejects the part at the back of the die.

Fig. 2 shows one cycle of the operation. Frames 3 to 6 show the left hand reaching for a selected relay spring. Then the attention shifts to the die to locate the part held by the right hand, frames 6 to 12. During this time (frames 6 to 12), the left hand is pre-positioning the next part for the tweezers. Now, reverting to the part held in the tweezers; when this part is located in the die the tweezers in the right hand reach for the part held by the left hand. As the right hand is reaching, the pedal is pressed, ejecting the part from the die. The attention at this time is directed in transferring the part from the left hand to the tweezers in the right hand, frames 13 through to frame 2.

Fig. 3 is a schematic diagram showing the paths of the hands, and the points of fixations of the eyes as laid out over the workplace.

Laboratory Equipment

Equipment Used in Performing Motion Cycle. The equipment

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[2] Industrial Engineer, Western Electric Company.

[3] Work has been arbitrarily divided into eighteen common elements called therbligs. For further information, refer to "Motion and Time Study," by Ralph M. Barnes, second edition, John Wiley and Sons, Inc., New York, N. Y., 1940, chap. 6.

Contributed by the Management Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[4] Because of the limitations of space not all of these nine aspects of the study are presented in this paper. For full details, refer to University of Iowa Studies in Engineering, Bulletin 22, University of Iowa, Department of Publications, Iowa City, Iowa, 1940.

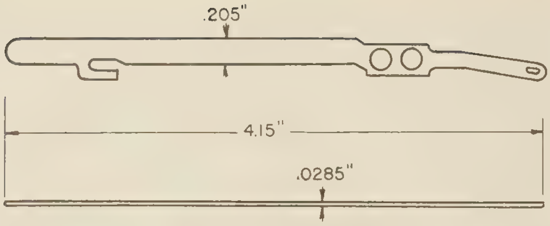

Fig. 1 Relay Spring

was so made and arranged as to impose on the subjects the same demands and limitations surrounding the motion pattern of punch-press work in an industrial shop. The part Fig. 1, used in the laboratory study, was in the same condition as when used in the shop. It was irregular in shape but all in one plane.

The laboratory equipment does not "process" the part. The part is merely located in the die, as though to be formed, and then ejected without being formed. A total of 250 parts were thus used over and over again during the run of the experiment.

A pair of aluminum punch-press tweezers, $^3/_{32}$ in. thick and $7^1/_2$ in. long, exactly the same as used in the shop for this operation, were used in grasping the part from the left hand and then placing the part in the die. In order to record the opening and closing of the tweezers, a 6-v circuit was completed at the instant the tweezers closed. This circuit, working through an electric relay, operated one of the pencils on the kymograph. By making a series connection to both points of the tweezers, the kymograph pencil Fig. 4 would jog laterally on the tape when the tweezers closed; when the tweezers released the part opening the circuit, the pencil would jog back to its original position.

Fig. 2 One Cycle of the Operation

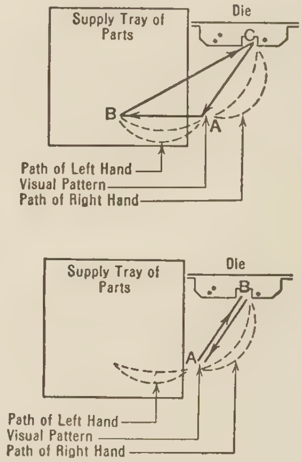

Fig. 3 Comparison of Visual Patterns

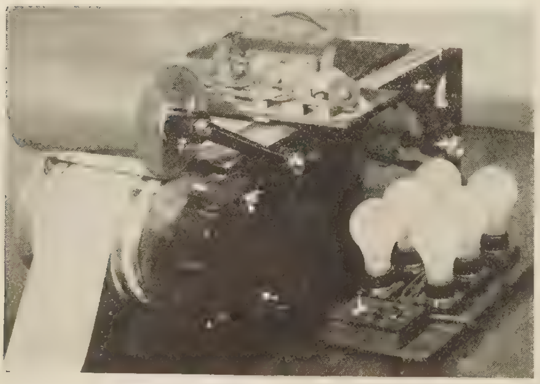

Fig. 4 Electrically Operated Kymograph Measures and Records Time

The die used in the study was made in the University of Iowa, mechanical-engineering laboratory, and was similar to common shop dies. Three, bullet-nosed pilot pins were used to locate the part in the die. The die operated by a cord attached to the foot pedal in such a way that, when the subject pressed the foot pedal, as a shop operator would to trip a press, the die rotated 180 deg, ejecting the part with a loud thump as the die hit the stop.

The pedal also operated a Veeder counter, used primarily to keep a count of the cycles of practice during the practice periods.

Equipment Used for Measuring and Recording Time. The kymograph, shown in Fig. 4, is a device for measuring and recording time with great accuracy. This machine was built in the industrial-engineering laboratory at the University of Iowa. Narrow paper tape is drawn across the kymograph table and under solenoid-operated pencils by means of two rollers driven at uniform speed by a synchronous motor.

Opening and closing electric circuits by means of photoelectric cells and switches causes the pencils to move laterally, making jogs in the lines drawn on the moving strip of paper, Fig. 5. The tape travels at a uniform velocity of 1914 in. per min.

Four pencils were used for recording time in this study. The work of the left hand was divided into three elements, the work of the right hand into three elements, and the work of the foot into two elements. Fig. 5 is a reproduction of the record made by the solenoid-operated pencils for one cycle of the operation.

Camera and Projecting Equipment. Motion pictures were taken by the Eastman Special and Bell and Howell 70-E cameras. Color pictures were taken of one subject for demonstration purposes and of another for frame-by-frame analysis.

The pictures were taken at 1000 frames per min with the camera always started on a full-wound spring. These cameras were checked, showing that an error of not more than 2 per cent occurred from full-wound spring to the automatic stopping point.

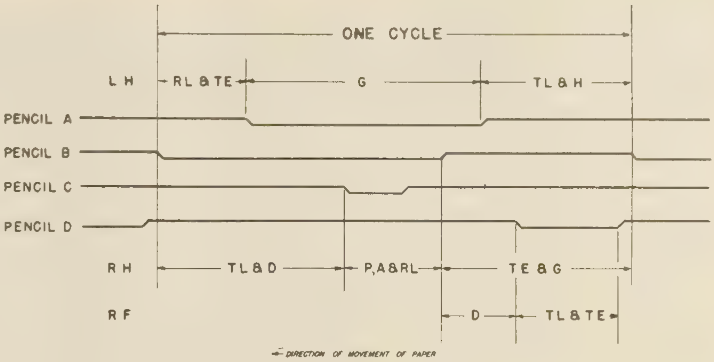

FIG. 5 REPRODUCTION OF ONE CYCLE OF RECORD MADE BY SOLENOID-OPERATED PENCILS ON KYMOGRAPH

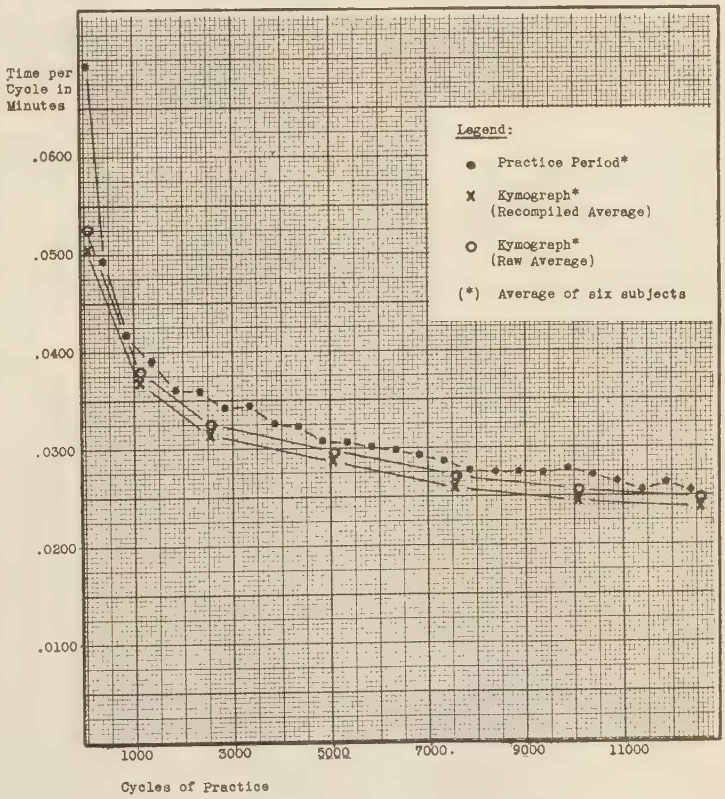

FIG. 6 LEARNING CURVES; COMPARING KYMOGRAPH DATA WITH PRACTICE-PERIOD DATA

SCHEDULE OF PRACTICE AND SAMPLING PERIODS

It is generally recognized that short and frequent periods of instruction and practice produce greater skill in shorter time than longer and infrequent periods. This consideration was regarded as important in arranging the schedule which was for 500 cycles a day, 5 days a week, Monday through Friday inclusive for the 5-week run.

The practice periods were divided into two parts of 250 cycles each, the time being taken on each of the two runs. On days scheduled for timing by the kymograph and camera, the first run as a practice run was for 250 cycles, the second run was for 200 cycles, and the third run was for 50 cycles. When obtaining a sample of 50 cycles by means of the kymograph it was necessary to have the subject actually perform about 75 cycles. The unrecorded practice amounts to not more than 2 per cent. It is, however, about the same for each subject and for the purpose of this experiment believed to be of minor importance.

Definitions. "Practice" normally refers to the period of work in which the subject is diligently applying himself to learning the motion cycle, as described.

"Sampling" refers to the use of the kymograph as a means of timing elements of cycle. It also includes taking moving pictures both for analysis in terms of the therblig times and as a means of sampling for demonstration purposes.

THE SUBJECTS

The six subjects who performed the work in these experiments were all students at the University of Iowa and undertook this work voluntarily in the interest of furthering a promising experiment. A consideration of the physical make-up of these students leads to the conclusion that there is no significant difference between them and employees encountered as applicants for industrial work of this sort.

INTERPRETATION OF RESULTS

Elimination of Abnormal Observations. In the raw data, it was noted that there were occasional observations far removed in value from the rest of the group. Such values have little effect on the average but have considerable effect on the standard deviation. To avoid the effect of these abnormal values, a refining process was applied to the data to eliminate each value more than two standard deviations removed from the arithmetic mean. The procedure was to:

1 Compute the arithmetic mean.

2 Compute the standard deviation.

3 Eliminate from data all values more than two standard deviations distant from the arithmetic mean.

4 Recompute the arithmetic mean; this figure is referred to as the "recompiled average."

5 Recompute the standard deviation of the refined data; this figure is referred to as the "recompiled standard deviation."

The average of the recompiled cycle times is also shown in Fig. 6, and demonstrates that the extreme values were of the greater magnitude. The difference between the raw data and the recompiled averages ranges between 2.8 and 3.7 per cent.

Analysis of Therblig Combinations. Definitions for therblig combinations are given in Table 1. The data collected in this experiment permit study of the manner in which the time required to perform individual therbligs or combinations of therbligs varies as skill is acquired. This study permits a consideration of the manner in which the over-all learning curve is built up from the learning curves of the elements within the cycle.

Some of the learning curves take the general form of total cycle-time curves encountered in the literature. In other cases, the final proficiency attained is but slightly different from the pace set at the very beginning of the study. Finally, in those therbligs which are not necessary to the attainment of the best cycle time, it is found that there is no real consistency by subjects and that the tendency, if any, is to retain the starting time and perhaps starting method rather than to improve the time for such unimportant therbligs.

For some therbligs there is a comparatively wide range of dispersion among the subjects and for others there is but little variation in the final time required for performance. The therbligs will now be considered individually. Cycle-time learning curves are given in Fig. 7.

TABLE 1 DEFINITION OF THERBLIG COMBINATIONS: FOR KYMOGRAPH STUDY

Member	Therbligs	Definition	Beginning of measurement	End of measurement
Left hand (LH)	RL&TE	Acts of relinquishing control and reaching for next part	From instant tweezers are closed on part	Until hand interrupts beam of light in reaching for a part on supply platform
	G	Act of obtaining control of a part	From instant hand interrupts beam of light	Until hand has complete control of part, allowing beam to activate photocell as hand is drawn from supply table
	TL,PP&H	Act of turning part ready to be grasped, carrying it into transfer area and holding it while tweezers close on it	From instant hand leaves beam of light	Until tweezers are closed on part
Right hand (RH)	TL&D	Motion carrying part toward die	From instant tweezers are closed	Until part and tweezers interrupt beam of light located over and at front edge of die
	P,A&RL	The acts of directing visually part to nest; assembling part in nest, and opening tweezers	From instant beam of light over nest is broken	Until tweezers open
	TE&G	Acts of moving tweezers for next part and gaining control of part	From instant tweezers open	Until tweezers are closed on next part
Right foot (RF)	PD	A delay in action of pedal movement	From instant tweezers open	Until die begins to discharge part[a]
	TL&TE	Acts of pressing pedal down and withdrawing foot	From instant mercury switch completes circuit	Until mercury switch opens circuit

[a] The mercury switch is activated by the pedal which ejects the part and also rotates a fast-moving cam, thereby tipping the mercury switch. This delay also occurs in the reverse process when opening the circuit, thus effecting a compensating factor.

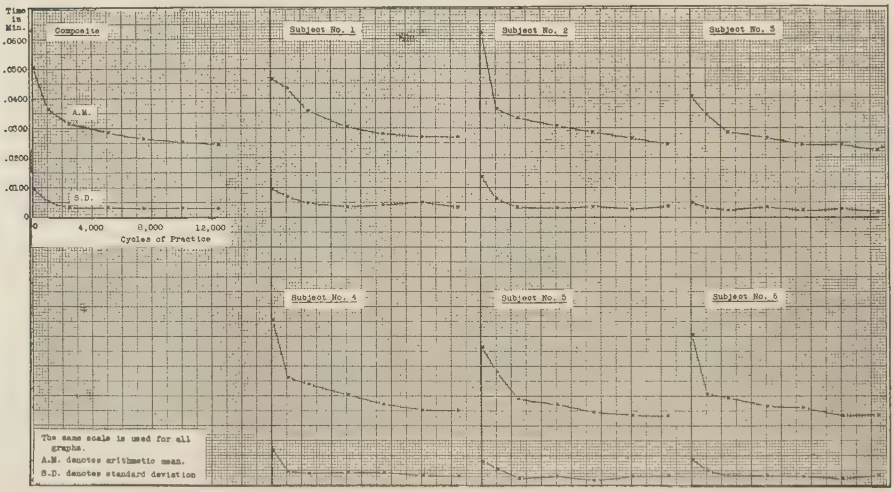

FIG. 7 CYCLE-TIME LEARNING CURVES; COMPOSITE AND INDIVIDUAL GRAPHS; ARITHMETIC MEAN AND STANDARD DEVIATION

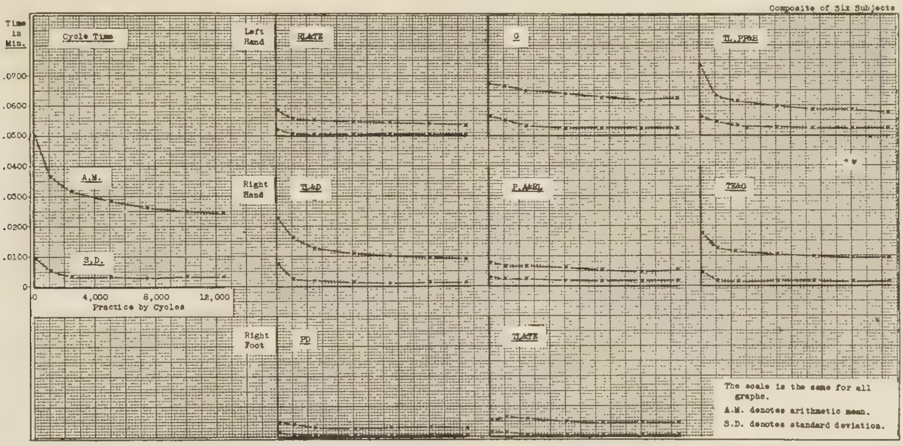

Fig. 8 Learning Curves of Cycle Times and Elements; Composite of Six Subjects; Arithmetic Mean and Standard Deviation

Left-Hand (LH) Operations

Release Load and Transport Empty (RL&TE). This is the element of the left-hand releasing the part previously grasped by the tweezers and then reaching to the supply tray for the next part. The trend of the average time from first sample to last sample shows that this element is one in which practice has considerable effect on the performance time. The time was reduced to 45 per cent of the starting time.

This is an interesting observation, especially in consideration of what the subject was doing. It is during this element that he

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	59.82	33.78	44.54	52.23	45.82	36.53	45.45

looks at the supply of parts, selects a particular part, and then considers how he will transport load and pre-position the part before he grasps it. There are four ways of pre-positioning a part, depending upon which of the four possible ways the part is fortuitously reposing in the supply tray. It seems that the poor initial time of some of the subjects may be the result of indecision during the process of determining how to pre-position.

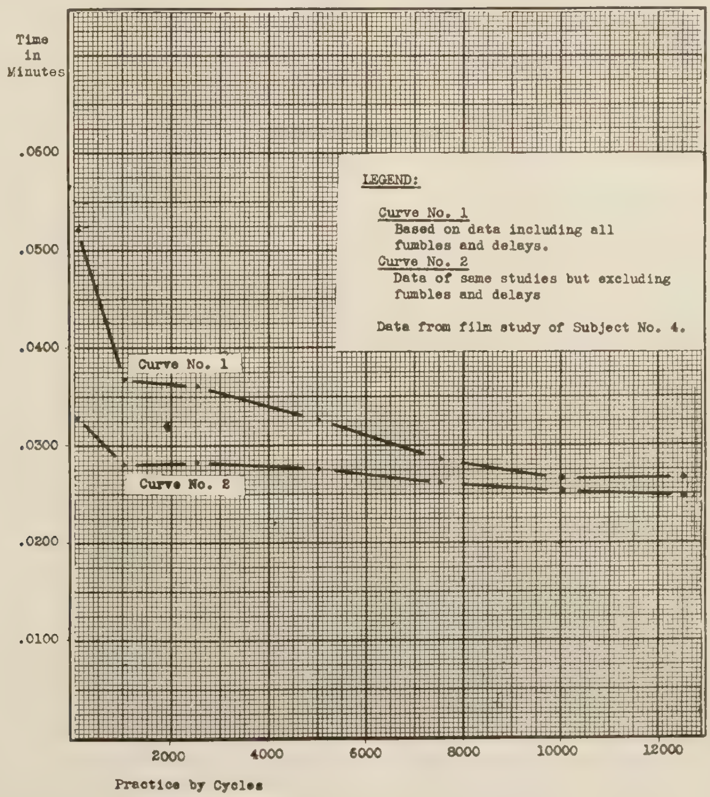

Fig. 9 Curves Showing Effects of Fumbles and Delays on Learning Curve

As soon as the subject completes this selecting part of the task, he looks to the die to direct the right hand, so he then immediately has another mental problem. Since these mental problems are solved in chronological order, mental agility is an important factor in the TE therblig time. The decision time is probably the predominant part of the element time at the outset, but reduces to become a minor influence in the later stages.

A study of the standard deviation shows a reduction in the early stages and a rise toward the final stages in the study. The reason for this is not readily evident but, undoubtedly, the phenomenon carries some significance because it is contrary to normal expectancy. The subjects generally show the same trend both in change in the average time and in the standard deviation.

Grasp (G). Because of the nature of the timing apparatus, the grasp includes a little of the transport motion. Other than this, the subject's visual attention at this stage of the cycle is in directing the right hand to place the part in the die.

Three of the subjects required more time at the end of the study for this element than they did at the beginning.[5] The general trend, however, does show a slight reduction in time. The average of the individual scores shows that the final stage amounted to 65.65 per cent of the time for the initial performance.

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	106.93	68.86	66.60	54.26	54.01	43.21	65.65

In studying the trends of the standard deviation of the learning curve there is a common trend for the final performance to have a lesser deviation than the first performance.

Transport Load, Pre-Position and Hold (TL,PP&H). This element is the one in which the part is carried by the left hand to the tweezers; it includes turning the part ready to grasp and is held in place until the tweezers close.

There is a strong tendency rapidly to decrease the performance time. In considering the "last sample to first sample ratio" the reduction is to 38.12 per cent of the starting time.

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	35.54	19.15	48.81	35.53	27.11	62.57	38.12

A study of the graphs Fig. 8, indicates a rather strong tendency to conform to the normally conceived learning curve.

Right-Hand (RH) Operations

Transport Load and Delay (TL&D). This is the element of the cycle in which the right hand carries the part held by the tweezers toward the die. It does not include placing the part in the die.

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	43.04	33.49	50.45	37.83	43.72	38.79	41.23

The table shows a fairly consistent reduction in time and the graph shows the curves to be quite normal and without peaks or flutter, averaging 41.23 per cent of the starting time.

It is during this phase of the cycle that the attention of the eyes is directed toward the selection of a part, and also in directing the left hand to the part. Also, during this element the eyes shift to the die and thus the attention is shifted to the direction of the right hand. As the part held by the tweezers penetrates the beam of light in front of the die, the element is terminated.

Considering what is going on during the element, the mental task of selecting, the directing of a hand, the shifting of the eyes to the die and the shifting of the attention to a different problem, the consistency of the progress is surprising, as is also the consistency of the standard-deviation learning curve. It is also noted that the coefficient of variation is particularly low for this element.

Furthermore, this particular element is one in which, possibly, a visual pattern change might be manifested. After a few thousand cycles of practice, subjects would now and then use only two fixations to a cycle while at other times they use three fixations per cycle, according to the original instructions.

Position, Assemble, and Release Load (P,A&RL). This element is for the short motion in locating the part in the die.

The impression from first observation of the data is that this series is a most erratic group of learning curves. Two of the subjects, after several hundred cycles of practice, took more time than for the initial sample. Five of the subjects took more time in the last sample than they did in the previous sample. In addition, there were other peaks and mounds. However, the final study shows that, on the average, the subjects were performing in 66.7 per cent of their starting time.

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	67.03	52.69	52.81	90.54	74.91	62.18	66.69

In the section of therblig findings based on the film data, it was found that the RL was performed very consistently, consequently if the RL were deducted, the remaining P and A must be the erratic elements.

In reviewing what is taking place, it should be noted that it is this phase of the cycle in which the part is placed over the pilot pins. And although the pilot pins facilitate the positioning, in many cases the subjects do not locate the part exactly right in the first locating movement, and small adjustment movements are necessary. If the subject tries to perform this motion too fast, he probably will misplace the part and waste time in recovering it. Less over-all time would be required if the subject worked more slowly but exactly.

The standard-deviation learning curve is likewise erratic for each subject and also leaves much to be explained. The coefficient of variation for this therblig exceeds that of any others.

Transport Empty and Grasp (TE&G). This element follows the release of the tweezers at the die. The TE&G includes the reaching of the right hand and tweezers to the transfer area, and the closing of the tweezers on the part.

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent	83.62	43.94	60.61	47.32	46.86	49.42	55.3

Right-Foot (RF) Operations

Pedal Delay (PD). This element is defined as the delay between the opening of the tweezers and the closing of the mercury switch actuated by the foot pedal.

	Subjects						
	1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent..............	68.43	99.84	62.52	14.34	60.20	74.23	63.26

The inconsistency of data in the learning trend for each subject as well as the inconsistency between subjects shows that these curves involve a phenomenon distinctly different from that previously encountered.

The subjects were instructed to press the pedal as soon as the part in the die was released. However, since a fast follow-up in the foot action after the release does not necessarily reduce the cycle

[5] All subjects to a minor extent performed slight variations in the cycle at the outset of the study. However, practice had the effect of making all subjects perform more like the instructed method.

time, the subjects all floundered around, trying faster and slower times as they pleased. More than 50 per cent of the time the foot was not busy, so it really made little difference how fast or slow the subjects coordinated the hand RL with the pedal TL. Accordingly, each subject took whatever time suited him.

It is interesting to note that only one subject early in the study seemed to find a rhythm which she tended to maintain throughout the study.

Transport Loaded and Transport Empty (TL&TE). In somewhat the same respect, the TL&TE is considered. This element is the time to press and release the pedal.

	Subjects 1	2	3	4	5	6	Avg
Ratio of final performance time to original time, per cent.......	118.07	121.38	68.91	118.68	67.70	42.02	89.46

There is no consistency within the individual studies, except in the case of one subject who tended to perform in about the same time throughout the study; and there is no consistency between subjects. This element is analyzed as one which is not important in cycle-time performance, consequently, the subjects perform it just as they wish to.

Fumbles and Delays

In analyzing the motion-picture film of one of the subjects, fumbles or delays were noted in varying frequency among the several therbligs. Relating the number of fumbles and delays to the total number of cycles shows clearly which are the troublesome elements.

Any hand movement not performed in accordance with instructions has, for the purpose of this study, been considered a fumble. Any retardation not necessitated by the methods as prescribed has been considered a delay. For example, when the left hand holds the part for the tweezers to grasp, the therblig is an instructed hold; however, were there a fumble in bringing the part to the grasping area, causing the hand to wait for the part, then the delay occurring in the grasp therblig of the right hand would be included as a delay. In this way a fumble or delay by one hand can result in a delay for the other hand. Table 2 shows in which therbligs and to what extent the trouble occurs by percentage, at the various stages of acquisition of skill.

TABLE 2 PERCENTAGE OF THERBLIGS IN WHICH FUMBLES OR DELAYS OCCURRED

After practicing, cycles	Left hand RL	TE	G	TL & PP	H	Right hand H	Tl, P&A	RL	TE	G
50	0.0	87.5	25.0	50.0	0.0	0.0	50.0	6.3	56.3	0.0
1050	0.0	6.7	40.0	93.3	0.0	0.0	73.3	0.0	0.0	6.7
2550	0.0	0.0	76.7	56.7	0.0	0.0	70.0	0.0	0.0	23.3
5050	0.0	0.0	26.7	54.8	0.0	0.0	74.2	0.0	0.0	29.0
7550	0.0	0.0	15.4	11.1	0.0	0.0	11.1	0.0	0.0	18.5
10050	0.0	0.0	15.1	5.9	2.9	0.0	5.9	0.0	0.0	8.8
12550	0.0	0.0	23.1	19.2	0.0	0.0	15.4	0.0	7.7	3.9

Effects of Fumbles and Delays on a Learning Curve

Fumbles and delays, which occur due to the lack of experience, have much to do with the form of a learning curve. The film analysis permits the graphing of two learning curves as follows:

1 One showing the normal curve, i.e., including all fumbles and delays.

2 The other showing a learning curve which excludes fumbles and delays, Fig. 9.

By excluding these obvious fumbles and delays, the learning curve is surprisingly flat. At the outset, the cycle time, excluding fumbles and delays, amounts to 63 per cent of the total time and, at the finish of the study, the data excluding fumbles and delays amounts to 93 per cent.

There follows a mathematical analysis of the differences of the two learning curves:

Cycle time at outset................	0.0521 Min
Cycle time at finish................	0.0272 Min
Improvement......................	0.0249 Min

The improved performance can be traced to the following overlapping causes:

Reduction in fumbles and delays...........	0.0168 Min
Faster performance.......................	0.0081 Min
Total................................	0.0249 Min

This analysis indicates that of the improvement obtained, about two thirds may be assigned to the elimination of obvious fumbles and delays and only about one third may be assigned to faster movements and better coordination. This proportion was somewhat surprising and may require reconsideration of some of the concepts of learning. It may develop that the phenomenon known as rhythm, coordination, deftness and smoothness, as distinguished from awkwardness, may consist not of a general status but of the absence of specific and definable fumbles and delays.

Change in Visual Pattern

During the early part of the learning period, it was observed that occasionally the subjects did not perform the regular visual sequence. While normally the subjects were required to perform according to the instructions, this departure from the standard method was regarded as having special significance, and the subjects were allowed to develop this departure in a natural manner.

According to the instructions, three visual fixations were to be performed in conjunction with the hand movements of each cycle. The first fixation was to occur as the left hand reached to grasp the part from the supply tray; the eyes then were focused on the supply tray during the selection of a part. The second fixation occurred as the right hand placed the part in the die, and at this time the eyes were focused on the die. The third fixation occurred as the tweezers in the right hand grasped the part being held by the left hand; the eyes then were focused on the part, Fig. 3. After several thousand cycles of practice, the tendency of the subject was sometimes to use three fixations and sometimes to use two, as distinguished from the exclusive use of either two or three. This mixture may in reality have been an intermediate stage of shifting from the use of three fixations to two fixations.

It is natural to question whether the subjects should have been instructed to use a two-fixation pattern. In response to this question, there is well-founded opinion that to do this would have caused considerable trouble. In the early training, looking in sequence toward the three places was essential.

When the subjects used the two-fixation pattern, the paths of the hand movements were the same; the only obvious difference was the points of fixations. The eyes would first fixate on the part during the transfer from the left hand to the tweezers in the right hand. The second fixation would occur when the eyes were focused on the die when placing the part in the die, as illustrated in Fig. 3.

Comparison of Time for Two- and Three-Fixation Methods

The film study provides quantitative data showing what proportion of cycles was being performed by the two-fixation method, and also a comparison of the average time values for each of the two methods during the transition.

	After practicing							
Cycles	5050		7550		10,050		12,550[a]	
No. of fixations	3	2	3	2	3	2	3	2
Percentage of sample	90.0	10.0	81.0	19.0	56.3	43.7	60.0	40.0
Time (avg), min	0.0323	0.040	0.0281	0.038	0.0279	0.0299	0.0363	0.028

[a] Because of scanty data taken at the 12,550 period, the 10,050 stage is generally referred to because it is statistically more significant.

In summary, it is observed after 10,050[a] cycles of practice:

1 That 44 per cent of the sample consists of the two-fixation method.

2 That a trend of the average time per cycle for each of the two methods indicates that, if the study had been carried on longer, that the two-fixation method would have occurred still more frequently.

3 That the two-fixation method would have taken less time than the three-fixation method, if the study had been carried on further.

This is regarded as a rather original finding and is important because:

1 The subjects (with one exception) were not aware that this change was taking place. They believed that they were following the instructions (and they actually were so far as their hands were concerned) but somehow they were not conscious of the visual pattern change that was taking place until toward the end of the study.

2 Instructions by industrial engineers seldom include directions for the visual pattern. The fact that it appears natural for an operator to change from one method to another without showing apparent trouble such as an irregularity in the cycle-time learning curve is a valuable finding. It is recognized that changes of hand movements alone will cause various effects on the learning curve, but the data do not seem to manifest any of the effects that the introduction of a change in method would be expected to show. Also, it is of practical importance to the industrial engineer to know that he can instruct a worker at the beginning to perform three fixations, and mention that ultimately only two fixations should be included. This may effect a faster learning period.

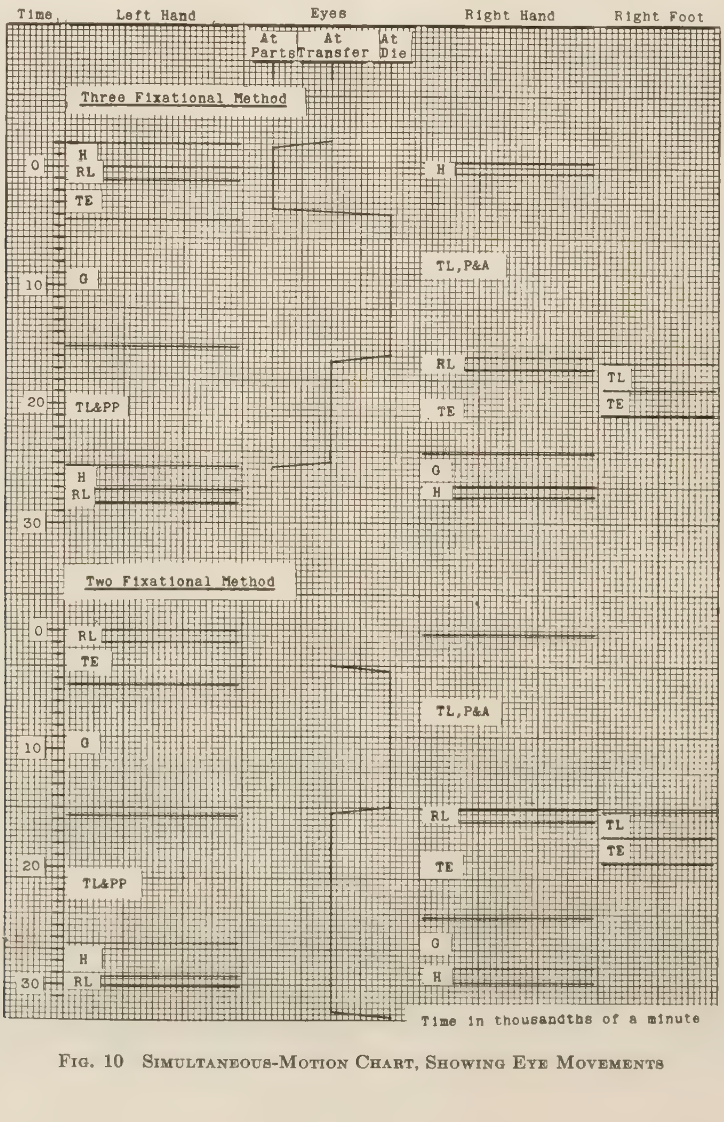

FIG. 10 SIMULTANEOUS-MOTION CHART, SHOWING EYE MOVEMENTS

TABLE 3 PERCENTAGE CHANGE FROM FINAL NORMAL RUN TO SPEED RUN

Subject no.	Left hand RL&TE	Left hand G	Left hand Tl, PP&H	Right hand TL&D	Right hand P,A& RL	Right hand TE&G	Cycle time	Right foot PD	Right foot TL&TE
1	— 3.33	— 7.90	— 3.18	—17.82	— 7.96	+ 0.65	—7.49	+ 3.33	—11.54
2	+ 0.07	—19.42	+39.01	— 9.55	— 6.15	+13.72	+0.29	— 1.83	—11.18
3	— 2.46	— 3.45	+13.15	— 4.30	+ 7.29	+ 6.75	+0.85	— 3.35	+12.66
4	— 8.98	— 5.24	+ 0.38	—18.40	+ 7.11	+ 4.05	—5.41	+37.35	— 4.89
5	—13.08	+ 2.07	—10.70	—19.79	+20.01	— 0.69	—4.18	—33.83	+ 0.89
6	+ 5.51	+ 9.55	—16.66	— 8.75	+10.89	+ 7.99	—1.87	— 5.48	+16.27
Weighted avg, per cent change.......	— 3.95	— 4.63	+ 2.82	—13.18	+ 5.15	+ 5.21	—3.12	— 7.08	— 1.80

NOTE: A negative sign indicates that the speed run was faster than the normal run, and a positive figure indicates that the "speed run" was actually slower.

TABLE 4 PERCENTAGE CHANGE FROM NORMAL SAMPLE TO SOLDIERING SAMPLE

Subject no.	Left hand RL&TE	Left hand G	Left hand TL, PP &H	Right hand TL&D	Right hand P,A& RL	Right hand TE&G	Cycle time	Right foot PD	Right foot TL& TE
1	+22.75	+18.27	+118.63	+ 43.91	+152.32	+23.74	+59.05	+ 13.08	+34.01
2	+24.90	+ 8.23	+ 90.59	+ 18.44	+ 29.41	+35.20	+31.42	+ 1.62	—12.35
3	+25.36	+47.31	+ 46.82	+ 62.24	+ 16.19	+42.82	+42.80	+ 55.52	+44.61
4	+39.27	+30.58	+160.09	+180.34	— 18.86	+41.17	+76.24	+3349.39	— 7.22
5	+12.71	+23.27	+ 33.97	+ 30.59	+ 1.83	+31.90	+20.90	+ 36.79	+41.29
6	+ 7.83	+ 6.20	+ 10.70	+ 14.70	— 2.05	+19.51	+10.96	+ 23.19	+38.51
Weighted avg, per cent change......	+21.84	+21.94	+ 74.75	+ 56.86	+ 27.23	+32.15	+40.99	+ 195.13	+25.48

NOTE: A positive sign indicates more time during soldiering than during last normal run.

Study of Relation of Eye Movements to Hand Movements

The film data permit making a simultaneous-motion chart, Fig. 10, of the events as they occurred after practicing 10,050 cycles. It shows the relation of one hand movement to the other, the relative duration of the visual fixations, and the instant the eyes started to shift. It also includes the action of the leg movement.

The two-fixation cycles are segregated from the three-fixation cycles, and this offers additional information regarding the manner in which the work was performed.

It was noted that in all therbligs requiring eye fixation, the eyes left the scene of action before the action was completed, Fig. 10. In making the transfer of the part from the left hand to the tweezers, the eyes moved away before the tweezers closed on the part. Likewise, when the left hand went to grasp a part from the supply tray, the eye moved away before the left hand had completed the grasp. Finally in locating the part in the die, the eyes moved away before the tweezers released the part.

From the viewpoint of the industrial engineer, the practical result is that the eye precedes the associated TE movement of the hand. It is from this viewpoint that quantitative measurements were made.

These findings should be considered in the preparation of a synthetic motion cycle. A careful analysis of eye movements and eye fixations should be considered. One could not say that the eyes always leave so many thousandths of a minute ahead of a TE or a TL, because sometimes it may be necessary to check visually the part in the die after the hand is away from the die. However, a careful analysis of the visual work in conjunction with the hand work would make possible a reasonably accurate forecast of the coordination to be found between these members in a motion pattern.

Speed Run

At the end of the last normal run each subject performed 15 cycles as fast as he could. This is called the speed run. It was explained to each subject that data were wanted which would reflect his most skillful performance at maximum effort. To be sure that fatigue would not affect the results, only 15 cycles of data were to be taken after the subject had gained "momentum" by performing a few cycles. Table 3 has been prepared to show the percentage change from the final normal run to the speed run.

The following conclusions were drawn from Table 3:

1 The subjects averaged 3 per cent faster performance for the speed run.

2 Performance during the regular kymograph periods, although without specific directions for speed performance, was evidently at slightly slower speed than top speed.

3 It appears that the subjects all could perform the TL&D element faster at will, but it was at the expense of the other two elements for that hand.

4 The six subjects do not seem to follow a common trend in the changes caused by the speed run.

Soldiering

If each subject, after acquiring a known degree of skill, decided to restrict his output, what would happen to the elemental time values? In order to explore this question the subjects agreed that they would hold back from top performance but only to such an extent that the tendency to hold back would not be apparent. The amount that each held back was a problem each individually solved. Table 4 was prepared to show the percentage change from the final normal sample to the soldiering sample.

The following conclusions are drawn from these data:

1 The extent to which the subjects increased performance time and yet considered it undetectable varies widely ranging from 11 to 76 per cent, averaging 41 per cent.

2 The averages of all elements are affected to some extent, the subjects averaging an increase of more than 20 per cent for each element.

3 The element showing the greatest increase in time was the TL,PP&H. This element provided a very plausible excuse in performing slowly, because, as one observed the workers, he could see that the act involved thinking of how to turn the part was causing the worker in this element to make slow movements.

4 There appears to be no common pattern or trend in soldiering.

Conclusions

Ordinarily a study of the conclusions reached in an investigation of this type will give to the reader a fair understanding of the problems encountered in the study. However, because of the pioneering nature of this study and because of the recondite nature of the findings, a study of the conclusions alone appears in this case to be inadequate, unless supplemented with some associated study of the details. For these reasons also, a special

effort has been made to include, in the body of the University of Iowa Bulletin,[4] much of the original data and all essential details.

Summarized Findings of the Investigation

The learning curves for the various therbligs show marked differences in characteristics. Some of them fall quite rapidly and others quite slowly. Clearly then the learning curve for the total cycle will be determined by the proportions in which the task contains rapidly learned and slowly learned therbligs. In the case of therbligs having no bearing on the cycle time, there is no definite tendency to reduce the therblig time as practice continues.

The elimination of noticeable fumbles and delays was in this study a greater factor in improving the cycle time than was the increase in speed alone. Here again, some therbligs are more susceptible to fumbles and delays than others, and the cycle-time performance will be influenced by the kind of therbligs which go to make up the cycle.

The data are inconclusive as to whether the learning curve for any one therblig will be alike or unlike for all subjects. The data are also inconclusive as to whether any one subject will adopt a predictable pattern for handling the various therbligs.

It was found that the visual pattern used by the subjects changed during the course of the study, and for the most part this change took place unknown to the subjects themselves. It was also found that the eyes can and do leave the scene of action before the action is completed.

An effort by the subjects to slow up their pace as far as possible without the change being detected resulted in changes in time for all therbligs, but not uniformly. This same lack of uniformity was evident during an effort by the subjects to "race."

Discussion

W. R. Mullee.[6] Many of us in the field of time-and-motion study have wondered how long it takes an operator to learn. We have been unable to ascertain the facts because we could not standardize the conditions as well as in the laboratory. This paper clearly indicates that time studies should not be taken on an operator before he has practiced the operation perhaps 1000 cycles.

In some plants where standards are set from time studies rather than from data, it can easily be seen that the resulting time standard will be too high. A time study which is taken at the beginning of the training period will be about $1^1/_2$ times as high as that obtained after 1000 cycles, and almost twice as high as that taken after, say, 10,000 cycles. This throws a great burden on the time-study investigator in attempting to rate such a wide variation in performance. Many of us feel that time studies cannot be rated accurately for performances which vary more than 20 per cent from normal for any given work element. The authors' experience seems clearly to indicate the desirability of not taking studies until the operator has become sufficiently skilled to eliminate at least most of the fumbling, etc.

It would be very interesting if another experiment were carried out to determine rapidity of learning when both hands are used simultaneously. From observations made in the plant, it seems evident that a longer period is required under such conditions.

Another interesting variation would be to determine the effect on learning if the number of therbligs to be learned were increased, i.e., if the cycle were made longer by introducing additional work elements. Also, to compare the rate of improvement under such conditions with that shown in this paper.

Summing up the discussion, the paper seems to prove the desirability of establishing time standards from data as compared with individual time studies; and indicates that jobs should be planned to avoid, as much as possible, conditions which might cause fumbles. Two thirds of the improvement noted was in the elimination of fumbling and one third was due to faster movements and better coordination.

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A New Steam Engine and Boiler

By S. L. G. Knox,[1] Englewood, N. J., and J. I. Yellott,[2] Chicago, Ill.

The paper presents descriptions of a new type of steam engine and a forced-circulation boiler which were designed by Mr. Knox and tested by Professor Yellott. The engine has a maximum capacity of 90 hp and is a high-speed compound reversible unit with uniflow exhaust. The description relates particularly to the unusual features of the inlet valves. The boiler employs a new principle for obtaining forced circulation. An impeller placed longitudinally in a lower drum exerts a pumping action which results in a uniform and rapid circulation through a number of riser tubes which comprise the walls of the combustion chamber. The paper is presented as a progress report. It is the authors' purpose by this means to present new and improved ways of accomplishing familiar ends, rather than to give an exhaustive discussion of the development of the particular units involved.

The current interest in high-pressure prime movers and forced-circulation steam generators gives timeliness to the following brief description of a compact steam plant which embodies both of these features. Originally designed to provide an efficient, powerful, and directly reversible unit for mobile purposes, the plant utilizes a high-speed compound uniflow engine, for which highly superheated steam is supplied at pressures up to 700 psi by a unique forced-circulation boiler. In its present state, the plant is noncondensing, but the engine may be discharged into a condenser and utilize to good advantage the additional energy thus made available.

The first part of the paper will be devoted to a description of the unique features of the engine. Results are also presented of a test run on the engine at the Stevens Institute of Technology.

The second section of the paper will describe the steam generator, which combines the merits of a multitubular water-level boiler with the advantages of forced circulation. Test results are also presented.

The Knox Engine

The prime mover of this plant is a high-speed compound uniflow engine of the steam-reverse type.[3] It is designed to operate at 1000 rpm, taking steam at 700 psi 750 F, and exhausting either to atmosphere or to a condenser. In brief, the engine is double-expanding, with one high-pressure and two low-pressure cylinders. The inlet valves are of the piston type, with certain unique variations which will be described later. These valves are actuated by an auxiliary crankshaft, which is driven by gears from the main crankshaft. Exhaust from all cylinders is accomplished by uniflow ports, with auxiliary exhaust provided by the piston valves.

The outstanding feature of the engine is the valve gear, by which a number of useful functions are performed without introducing more moving parts than the minimum number required for a simple nonreversing engine with the same number of cylinders. Most important of these functions are the provision of expansion and the automatic increase of torque at starting and low speeds, without additional mechanism. The result of the special features of the engine is a steam consumption of 14.6 lb per bhp-hr (noncondensing), and an engine efficiency of 53.3 per cent, obtained under test when the engine was loaded to 70 hp or about 75 per cent of its capacity.

The relative size and general appearance of the unit are shown in Fig. 1. The general plan of construction is shown in Fig. 2,

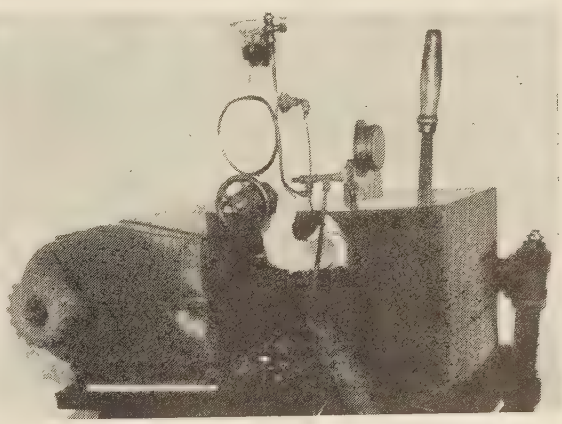

Fig. 1 The Knox Engine

which gives the principal details of construction of the engine, but omits the passages by which the steam makes its way from the valves to the cylinders. The single high-pressure cylinder, $3^1/_4 \times 4^1/_2$ in., is located between the two low-pressure cylinders, $4^1/_2 \times 4^1/_2$ in., as shown in the cross section of the cylinder block, Fig. 3.

Steam is introduced through a control valve which serves both as a throttle and as a steam-distributing valve. The direction of rotation of the engine is changed from forward to reverse simply by pushing the control lever to its forward or reverse position.

Sequence of Valve Events

When running forward, the steam passes from the control valve, designated as V in Figs. 2 and 3, to the inner edges of the high-pressure valve, shown as a in Fig. 3. This valve, as well as the other two valves b and c, is a piston valve. These differ from the usual piston valves in two important respects: First, the valve has a rolling motion as well as a reciprocating motion; thus the path of any point on the valve is an ellipse. The second unique feature is the serrated edges, which can be seen in Fig. 4. The combination of this unusual motion with the shape of the valve edges gives rise to the characteristics which will be discussed.

The high-pressure cylinder discharges into a receiver space, which is composed of the unused volume within the engine casting. The discharge takes place through exhaust ports in the middle of the cylinder which are uncovered by the piston in the con-

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[3] A steam-reverse engine is one in which the direction of rotation is reversed by changing the direction of flow of the steam. This is accomplished by using one set of ports in the valves as steam-admission ports in one direction, and as exhaust valves in the other direction.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

ventional uniflow manner. Auxiliary exhaust, necessary to keep compression down to a reasonable amount, is provided by the piston valves mentioned previously.

As is explained later, each valve is connected to two cylinders; the outer edges to one cylinder, the inner edges to another. If the direction of rotation of the engine is such that the outer

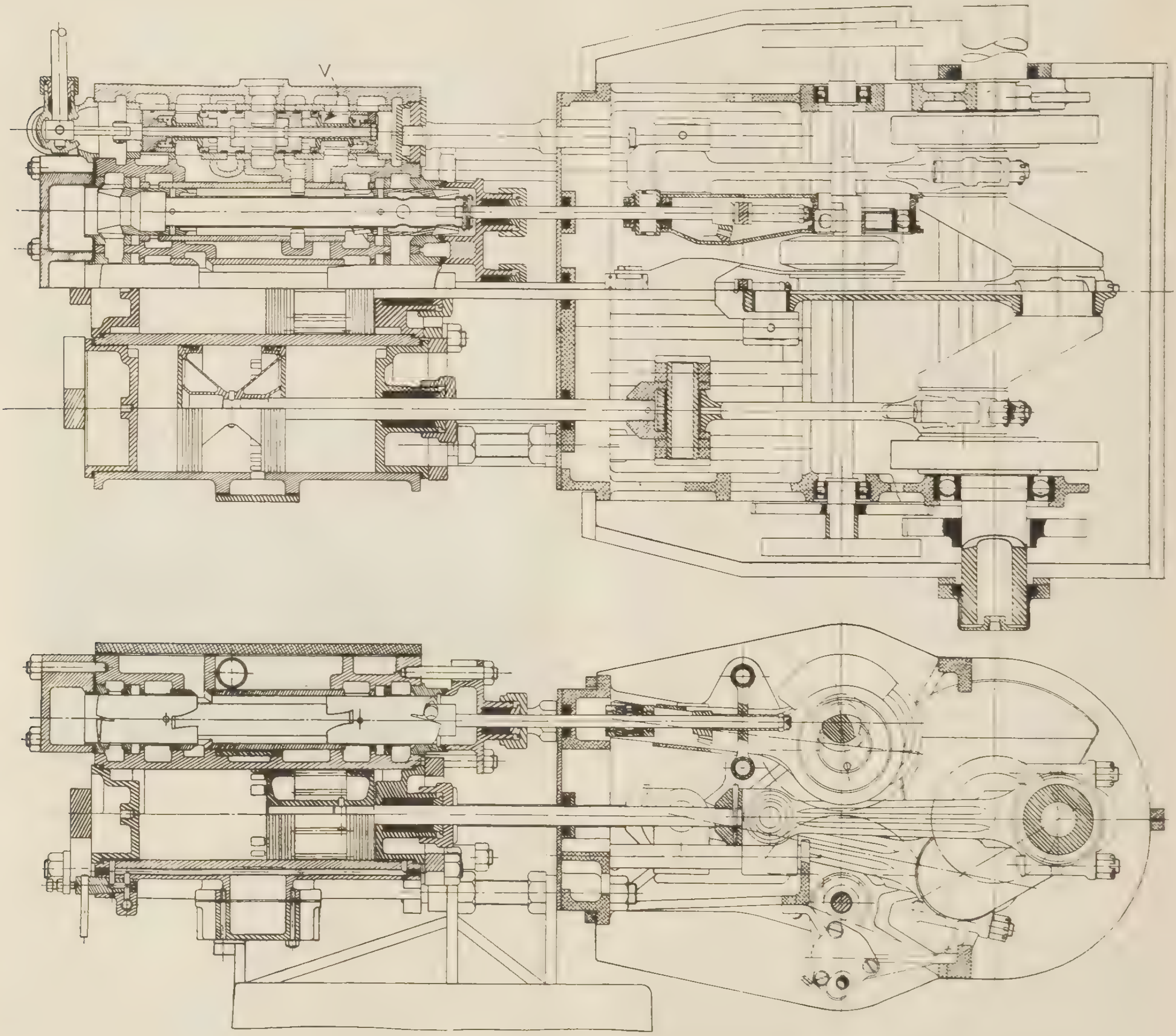

Fig. 2 Plan and Cross Section of the Engine

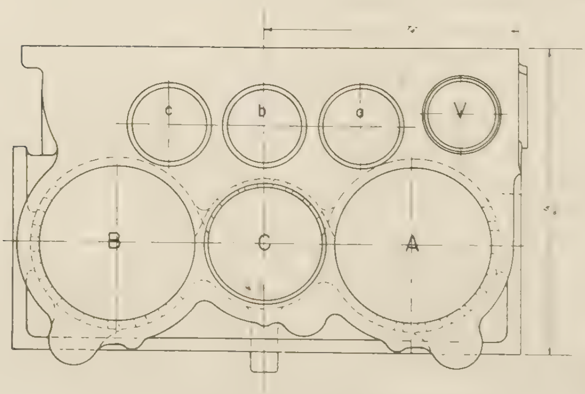

Fig. 3 Cross Section of Cylinder Block

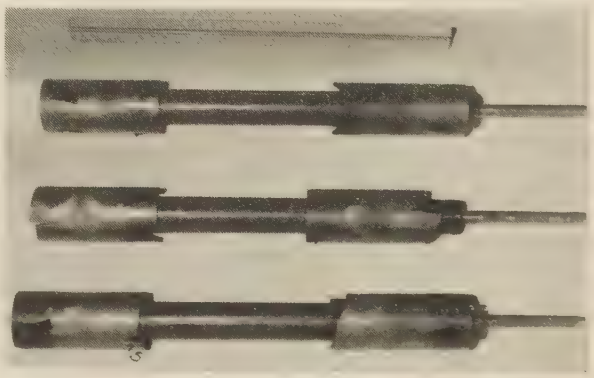

Fig. 4 Piston Valves

edges of a valve admit steam to a cylinder, the inner edges of the same valve will open during the exhaust stroke of the other cylinder, thus providing auxiliary exhaust for the second cylinder without any additional parts being required.

Running forward, steam from the receiver is directed by the outer edges of valve *b* to low-pressure cylinder *A* (Fig. 3) and by the inner edges of valve *c* to the low-pressure cylinder *B*. These two cylinders exhaust through central ports similar to those in the high-pressure cylinder, the steam being carried through a manifold to the exhaust line. Auxiliary exhaust is provided for cylinder *B* by the inner edges of valve *b* and for cylinder *C* by the outer edges of valve *c*.

The unique feature of this engine, as compared with other steam-reverse engines, lies in the fact that each valve supplies a particular cylinder when the engine is running forward, but a different cylinder when the engine is reversed. When the control valve is thrown to the reverse position, steam is no longer admitted to the inner edges of valve *a*, but rather to the outer edges of valve *c*. In the same manner, steam passes from the receiver to the outer edges of valve *a* and the inner edges of valve *b*, whence it passes to cylinders *A* and *B*, respectively. The opposite edges of the valves again act as auxiliary relief valves. It is by this means that a cutoff of 30 to 40 per cent (depending upon the slope of the serrated edges) can be obtained in all the cylinders of this engine. Earlier steam-reverse engines necessarily had 100 per cent cutoff and consequently their steam rate was several times as high as that of the present engine.

The usefulness of the serrated edges in combination with the unusual rolling motion, lies in the fact that the effective cutoff can be lengthened, thus increasing the torque of the engine. The rolling motion of the valve is imparted by a simple arrangement which combines a segmental bevel pinion on the valve stem with a segment of bevel gear on the eccentric strap. The vertical component of the motion of the strap, caused by the rotation of the eccentric, causes the valve stem to turn first in one direction and then in the other. Both stem and strap have the same reciprocal motion, and so the gear segments are always in mesh.

The serrated forward admission edges of the valves, shown in Fig. 4, exemplify one of the advantages which may be obtained as a result of the combined reciprocating-and-oscillating movement of the valves. In an engine where high power-to-weight ratio is desired, as was the case in the present engine, this construction permits lengthening the admission period for a given angle of advance of the eccentrics. This angle is limited to a rather narrow range, to obtain reversibility without serious reduction of efficiency. Because of the serrations, the valve, rolling over as it reaches its dead center, remains open for a longer period than would otherwise be possible with the proper angle of lead.

In engines where the power-to-weight ratio is not a prime consideration, these serrations can be omitted and the admission edges provided with piston rings, thus giving a shorter effective cutoff with resultant increase in total expansion ratio.

At *S*, Fig. 4, is shown a small slot or notch running longitudinally. The purpose of this slot is to lengthen the cutoff beyond that which would be possible with only the reciprocating motion of the valve, when the eccentrics are set at an angle of advance which gives an economical point of cutoff. If this slot were exposed to steam when the valve approaches the point of admission, steam would be admitted to the cylinder so far in advance of dead center that the engine would not operate. However, due to the rolling motion of the valve, the slot is covered by a bridge as it approaches the point of admission, and is rolled out into the steam space only after dead center has been passed. It remains exposed to steam until nearly the end of the stroke, however. Its width ordinarily would be from 2 to 3 per cent of the circumference of the valve. Thus, when the engine is starting or running at low speed, it is supplied with steam at approximately full pressure during practically the entire stroke. As it speeds up, the amount of steam which gets through per stroke becomes less important. At maximum speed the effect on efficiency is negligible. Thus, it is possible to obtain the effect of a long cutoff, and the resulting increased torque at starting and low speeds, with a fixed economic eccentric position. This eccentric position would normally give the same short cutoff at all speeds, except for this slot and the rolling motion of the valve. In this way is obtained the same effect of long cutoff and maximum torque at low speeds, with short cutoff and high efficiency at high speeds, which in other engines is only obtainable with a number of extra moving parts, requiring hand or governor manipulation.

Engine balance, particularly important because of the high operating speed, is accomplished by two sets of counterbalances. One set is mounted on the eccentric shaft which rotates in one direction in a plane above the cylinders. The other set is mounted on the oil-pump shaft which rotates in the opposite direction in a plane below the cylinders. The system is so effective that there is virtually no vibration when the engine is running either forward or in reverse.

Lubrication of the bearings, cranks, and eccentrics is accomplished by splash and by forced circulation from a small vane pump, through holes drilled in the crankshaft, to the surface of each crankpin. Lubrication of the valve and pistons is accomplished by a multiplunger lubricator which sends cylinder oil into the main line and to each end of the three valves.

The over-all dimensions of the engine are shown in Fig. 1. The weight of the engine is 450 lb and the maximum power of the engine is 90 hp, giving a weight of 5 lb per bhp.

The Engine Tests

A number of trial tests were made with the engine during its development period. The tests followed standard practice. Pressures and temperatures were measured with calibrated gages and thermometers. Condensate was weighed to determine the steam consumption of the engine. Back pressure was atmospheric in all tests.

The load on the engine during these tests was supplied by a water brake which was constructed especially for this purpose. The design established by Professor E. P. Culver was followed, and the brake operated in a very satisfactory manner. This type of brake was particularly desirable because it operated equally well in both directions of rotation.

The limitation on the output of the engine was imposed by the boiler which at the time of the engine tests was unable to deliver more than 1000 lb of steam per hr with smokeless combustion. Later the combustion was improved until the boiler developed more than 1500 lb per hr without smoke.

The results given inTable 1 are typical of those obtained during the tests of the engine.

TABLE 1 TYPICAL ENGINE TEST RESULTS

Length of run, min	30
Initial pressure, psi abs	595
Initial temperature, F	750
Back pressure, psi abs	15.2
Rotative speed, rpm	1020
Brake output, hp	69
Steam used, lb	503
Steam rate, lb per bhp-hr	14.6
Ideal steam rate, lb per bhp-hr	7.79
Engine efficiency, per cent	53.3
Engine heat rate, Btu per hp-hr	17500
Engine thermal efficiency, per cent	14.55

The present engine was the first of its type and was designed especially with a view to light weight per horsepower-hour, with the sacrifice of the maximum efficiency otherwise obtainable. In view of this, of its small size and the fact that no changes have been made in the design as the result of experience, although vari-

ous improvements have been indicated, it is reasonable to assume that, with large engines, running condensing, steam consumption much below the 14.6 lb per bhp-hr obtained with the present engine should be realized.

If, as may reasonably be expected, a water rate of 10 lb or better per bhp-hr should be attained, the fuel consumed per horsepower-hour, while considerably greater than that of a Diesel engine, would cost less because of the much lower price per gallon of fuel usable in the furnace of a boiler, as compared with that required in a Diesel.

The Knox Boiler

This boiler, which was developed to supply steam to the Knox engine, had to be compact, relatively light, and capable of pro-

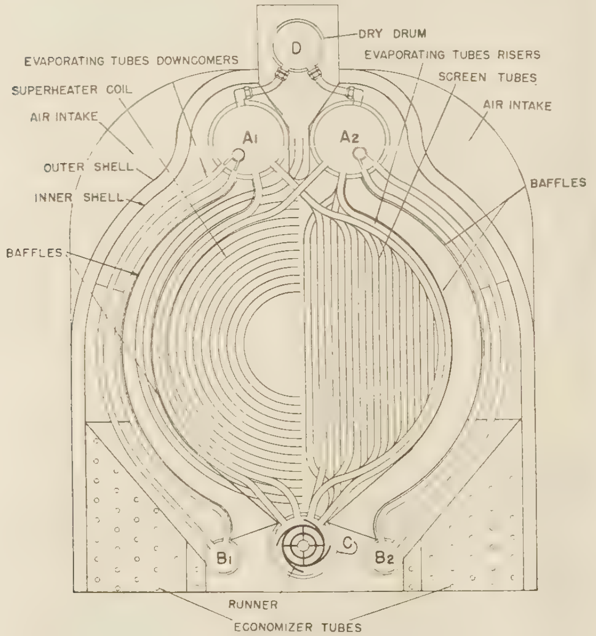

Fig. 5 Cross Section of Boiler

ducing 1500 lb per hr of steam at 700 psi 750 F with smokeless combustion and high efficiency. The necessity for forced circulation was immediately realized, but the difficulty of controlling the once-through or coil boiler made it desirable to retain the stability and reliability of the conventional multitubular drum boiler.

These requirements were met by using a new method of forced circulation, whereby a simple impeller in the middle lower drum, Fig. 5, causes the water to flow up through the riser tubes which surround the combustion chamber.

The feedwater is pumped into the two steam drums A_1 and A_2, after passing through the economizers which will be discussed later. Leading downward from these two drums are the downcomers, which are located in the third pass of the present boiler, and which connect to the two smaller lower drums B_1 and B_2. The center drum C at the bottom of the boiler is connected to the outer drums by headers at both ends, and within this center drum is located a hollow impeller. Figs. 5 and 6 show these points quite clearly. This impeller is driven by a shaft which emerges from the header through a simple stuffing box. When the impeller is rotated, the pumping action creates a pressure which forces the water in drum C to pass upward through the risers which form the walls of the combustion chamber. The suction caused by the displacement of this water causes the water in B_1 and B_2 to flow into the middle drum, and thus a vigorous forced circulation is established. This circulation is quite uniform along the length of the drum, as is shown in Fig. 7. This illustration is from a photograph taken when the boiler was not under pressure, but when the impeller was being operated. Glass plates pressed against the end of the drum prevented the water from spilling out of the boiler. Rough measurements indicate that the velocity is about 4 fps under the circumstances prevailing when the photograph, Fig. 7, was taken.

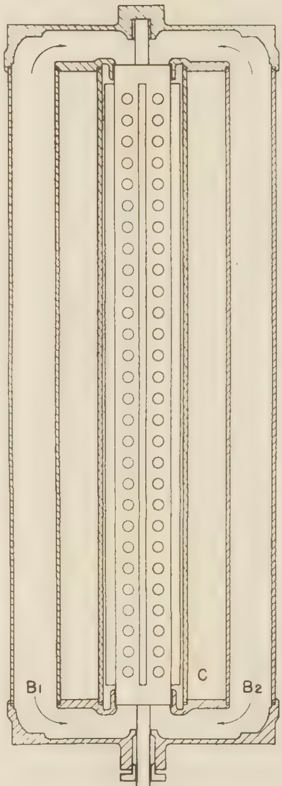

Fig. 6 Cross Section of Lower Drums

Since this pumping action is assisted by the natural circulation, it is obviously yet more powerful when the boiler is generating steam. In fact, the pump may be considered as a booster, when applied to a boiler having natural circulation, adding to the velocity which natural circulation will produce. It will be realized that this internally assisted circulation differs in several important respects from other well-known forced-circulation principles. By using the lower drum as the casing of the pump, the expense of a separate high-pressure pump is eliminated and, of far greater importance, uniform circulation is obtained. Balancing of the individual water circuits is automatically accomplished, since the added pressure comes from the centrifugal action of the impeller, which is uniform along the length of the drum. There is no tendency for the first tubes to rob the remaining risers, as long as the runner is adequate in size.

General Disposition of the Boiler Surface

The general arrangement of the heating surface is shown in

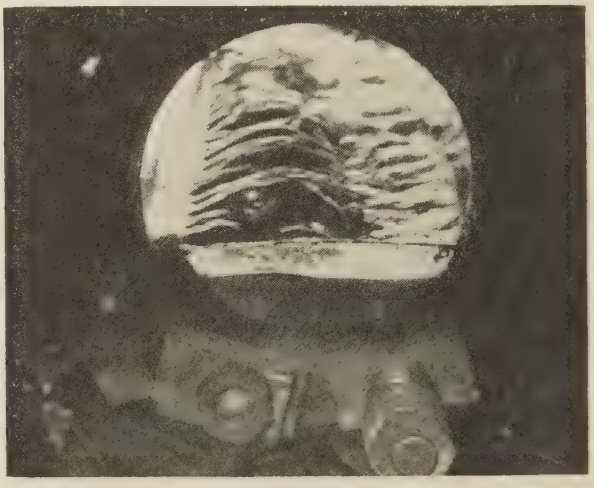

Fig. 7 Flow From Risers Into Upper Drum

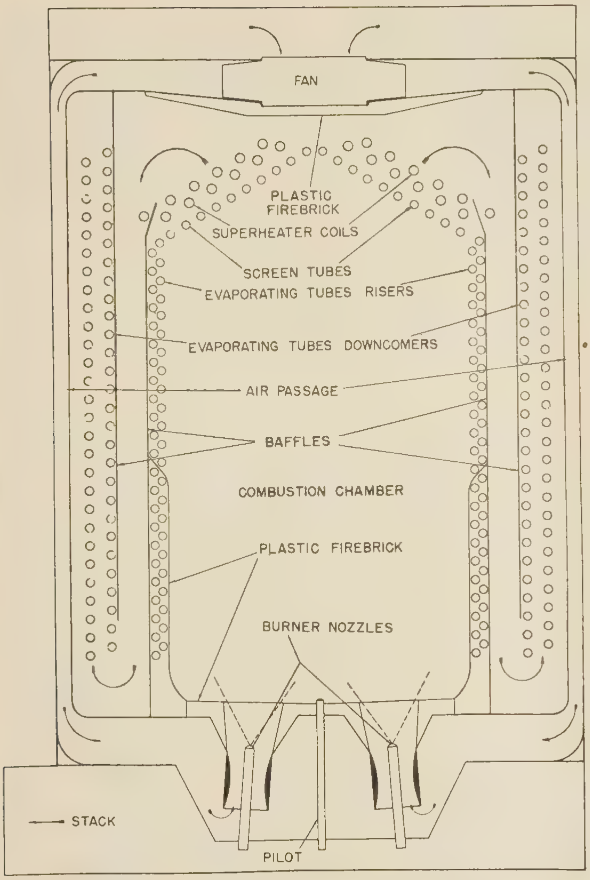

Fig. 8 Section Through Center of Boiler

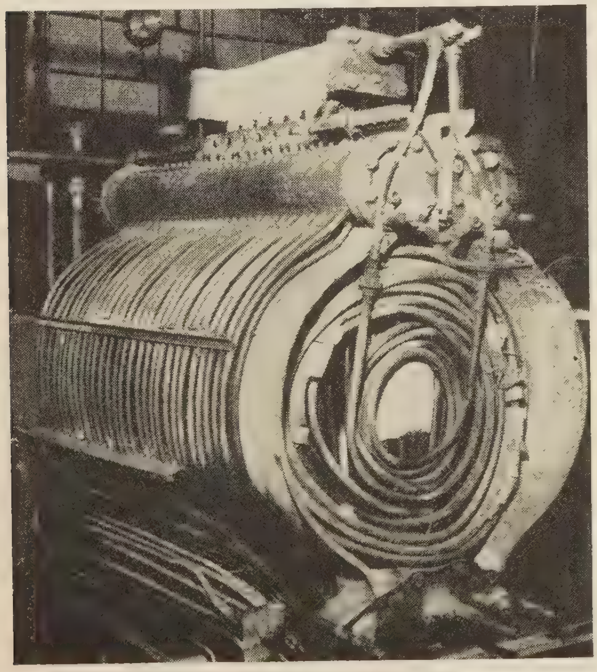

Fig. 9 View of Boiler With Outer Jackets Removed

Figs. 5, 8, and 9. The risers, through which the water is forced upward to the drum by the impeller, consist of four rows of stainless-steel tubes, each being $^1/_2$-in. outside diam $\times$ 0.43-in. inside diam, and approximately 3 ft long. The tubes are staggered, the distance between center lines being $1^3/_{16}$ in. The combustion space is thus entirely waterwalled, although it was found necessary to put refractory over the first few rows of tubes near the burner to provide radiant heat for vaporizing oil, Fig. 10, thus obtaining smokeless combustion and improved efficiency.

The downcomers are located in the third pass of the boiler. The heat transmission is much less intense in these tubes because they are not exposed to the radiation from the flame.

In order to supply dry steam to the superheater, a dry drum D is provided above A_1 and A_2 and is connected to them by a number of small tubes. This dry drum is only necessary because of the small diameter of the upper drums in this particular boiler. The boiler performed quite well for a long time without this drum but, under intensive operation, slugs of water would occasionally go through the superheater. In another boiler of this type having a single upper drum of a diameter suitably proportioned to the capacity of the boiler, no dry drum would be needed and a single upper drum would take the place of the three drums at the top of the present boiler. Perforated metal screens are also placed above the water line in A_1 and A_2 to prevent priming, or carry-over of large slugs of water.

The superheater is composed of a spiral coil of tubing and is located at the back of the combustion space, facing the burner, but separated from the combustion space by a screen of riser tubes. It was originally intended to install a second superheater around the burner, but this was found to be unnecessary. Control of the temperature of the superheated steam was accomplished by injecting feedwater into the inlet of the superheater. Steam temperatures above 800 F were obtained, although the temperature was usually kept around 750 F. The location of the superheater can be seen in Figs. 8 and 9.

Three economizers are used, with the result that the flue-gas temperature leaving the last economizer is always below 400 F. The first two economizers are located in the last pass, on each side of the unit, in space which is available because the general shape of the boiler proper is cylindrical. One of these economizers can be seen in Fig. 9 while Fig. 5 also shows them. The third economizer consists of coiled tubing, located in the duct leading to the stack.

Air Supply and Flue-Gas Disposal

Since the Knox boiler is intended to operate without the benefit of a stack to produce a natural draft, it is provided with a unique forced-draft system. The entire boiler is covered with an airtight metal casing, which is in turn partly enclosed within a removable aluminum outer jacket, Fig. 5. Between this outer shell and the jacket on either side of the boiler is a space through which the incoming air is drawn by the forced-draft fan. The fan consists of a cast-aluminum impeller, running on ball bearings, and driven by a V-belt and pulleys from the motor which also drives the water-circulating impeller. Major variations in fan capacity are accomplished by changing the driving pulleys and minor variations are made by restricting the air intake opening.

The air, which is drawn in through the fan after having passed over the outer shell of the boiler, is then forced through the space between the inner and outer shells, and is preheated to about 350 F before it enters the combustion chamber. The air pressure at the burner inlet is about 2 in. of water, under the usual operating conditions. As a result of this unusual arrangement of the air supply, the jacket around the boiler is at almost the same temperature as the surrounding air and, consequently, the losses due to radiation and convection are negligible.

The course of the flue gases can be seen in Fig. 8. After passing through the coiled superheater at the rear of the combustion chamber, the gases divide and turn back through the second pass, in which it was originally intended to locate more superheater tubes. Experience showed that the one coil was enough and, consequently, in future boilers this empty second pass would be omitted and the size and weight of the boiler thus reduced. This would also reduce the pressure required to force the gases of combustion through the boiler. Upon leaving the second pass, the gases go back through the third pass in which the downcomers are located. The gases then move down, as shown in Fig. 5, into the economizer space, and again come to the front of the boiler, where they enter the duct which leads to the last economizer and finally to the stack.

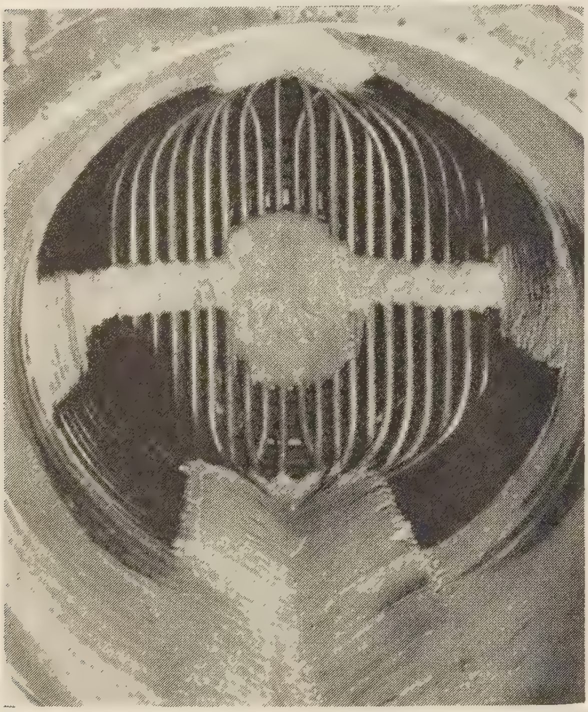

Fig. 10 Inside the Combustion Chamber

The usefulness of the last economizer is demonstrated by the test results, which show that the flue-gas temperature entering the duct was about 590 F at full load, while the gas temperature at the last economizer outlet was about 380 F.

Oil-Burning Equipment

The most serious problem encountered in the development of the boiler was the smokeless and efficient combustion of fuel oil at extremely high rates in a very small chamber. The combustion equipment which was in use when the tests were run consisted of four main burners, arranged around a small pilot burner, as shown in Fig. 8. The individual burners were located centrally in venturi-shaped openings through which the combustion air entered. The burners were standard 80-deg tips of the type used on domestic units, and the maximum capacity of the boiler could be varied by changing the size of the burners.

The boiler was started by electrical ignition of the pilot burner, which was located in the center of the refractory disk. After the main burners were ignited, the electrical ignition was shut off. The oil supply to the burners came directly from a small motor driven gear pump which kept the pressure at about 160 psi gage. The oil supply passed through a solenoid valve which was so connected into the control system that it could be shut off by the various safety devices which will be described.

The combustion chamber was partially lined with plastic refractory, as shown in Fig. 10, in order to give sufficient incandescent radiation surface to vaporize the oil. The amount of this refractory lining was determined by experience, and it was kept to a minimum so that the surface of the risers would be available for absorbing the radiant heat of the flame.

The furnace volume was about 5.67 cu ft, and this proved to be adequate during the official test for the release of 1,970,000 Btu per hr. The average heat release was thus about 350,000 Btu per cu ft per hr. Later the rate of smokeless combustion was increased to well over 400,000 Btu per cu ft per hr.

Controls and Safety Appliances

The problem of control in the boiler is greatly simplified because of the presence of the large upper and lower drums, which give a relatively large water capacity. Should the water level in the drums rise or fall as much as 2 in. above or below the usual line, no harm would be done. This is equivalent to more than 2 min output, about 15 times as long as it takes the controls to act, and many times as long as would be available for the much faster controls required for continuous-tube boilers. This feature of the boiler is particularly important, because it means that the fuel and water supplies do not have to be instantaneously adjusted.

Except for the safety valve, all controls are electrically operated. The safety valve is a double, spring-loaded poppet valve, which is connected to the two upper drums by pipes of ample size. The safety valve had no occasion to function during the tests.

The pressure control consisted of a spring-loaded bellows, equipped with contacts which were connected in series with the solenoid valve in the oil line. When the pressure rose above the set value, the contacts opened, the oil flow was shut off, the four main burners went out, and the pressure immediately fell. Since there is little refractory in the boiler to provide heat storage, there is consequently very little lag in the response of the pressure controller. The boiler can operate safely at pressures up to 700 psi, but during the test it was run at 500 psi gage.

The high- and low-water controls are of the thermal-expanding type, in which a brass tube alters its length in response to changes in the water level in the drum. Electrical contacts are opened when the water level falls too low, and the fuel supply is thus shut off. When the water level becomes too high, another set of contacts opens and these close a solenoid valve in the suction side of the feedwater line.

Manual control of steam temperature was in use during the tests. This control was accomplished by injecting water into the inlet of the superheater, the amount of injected water being regulated by a needle valve. The occasional wide swings of temperature which are to be seen in Fig. 11 are due to the fact that the water-level control affects the superheat control by altering the injection-water pressure. Subsequently, the superheat was controlled automatically.

It should be noted that a water-level glass is provided, the connections being taken from the right top drum A_2, in Fig. 5. This is an unusual feature for a forced-circulation boiler, since most of that type have such wide variations in water level that a glass would be impracticable.

During the test, the boiler was supplied with water by a multicylinder motor-driven reciprocating pump. The pump was kept running at constant speed, and the water delivery was regulated by the solenoid valve in the suction line. The water went from the pump through two separate circuits, each passing through two economizers, and then into one of the two upper drums.

Tabulation of Boiler Data and Report of Tests

The general over-all dimensions of the boiler proper are shown in Fig. 9. The height from the base of the boiler to the top of the cover over the dry drum is 38 in., the over-all length is 60 in., and the width is 36 in. The floor space occupied by the boiler is thus 15 sq ft. The secondary economizer is outside of the boiler and occupies about 3 sq ft if placed horizontally, or about 1 sq ft if placed vertically. If the space under the boiler now occupied by the fan motor is included, the over-all height is 57 in. A motor for driving the fan and runner could be placed within the present rectangular space 36 in. wide × 38 in. high, at the end of the boiler, or elsewhere if minimum height is required for the installation. If the unused second pass were eliminated, the boiler would be only 30 in. wide. Table 2 Gives general boiler data.

TABLE 2 GENERAL BOILER DATA

Heating surface		
Primary economizer, sq ft	26	
Secondary economizer, sq ft	32	
Total surface in economizers, sq ft		58
Downcomer surface, sq ft	43	
Riser and screen, sq ft	42	
Total evaporating surface, sq ft		85
Superheater surface, sq ft		11
Total heating surface in boiler, sq ft		154
Furnace volume		
Combustion chamber, cu ft	5.67	
Total volume within outer shell, cu ft	40.3	
Weight data		
Total weight of boiler (including refractory), lb	1235	
Weight of water in boiler, hot, lb	85	

Combustion data

Fuel burned—Essoheat medium, No. 2 domestic fuel oil; specific gravity 0.85 at 78 F; heating value 19,400 Btu per lb; 14.38 lb of air required per lb of oil
Air supply—forced draft, preheated to 360 F, 2-in. water pressure at burner intakes
Burners, number and type—four 80-deg tip, 3 gph capacity
Oil pressure (during test) 150 to 160 psi gage

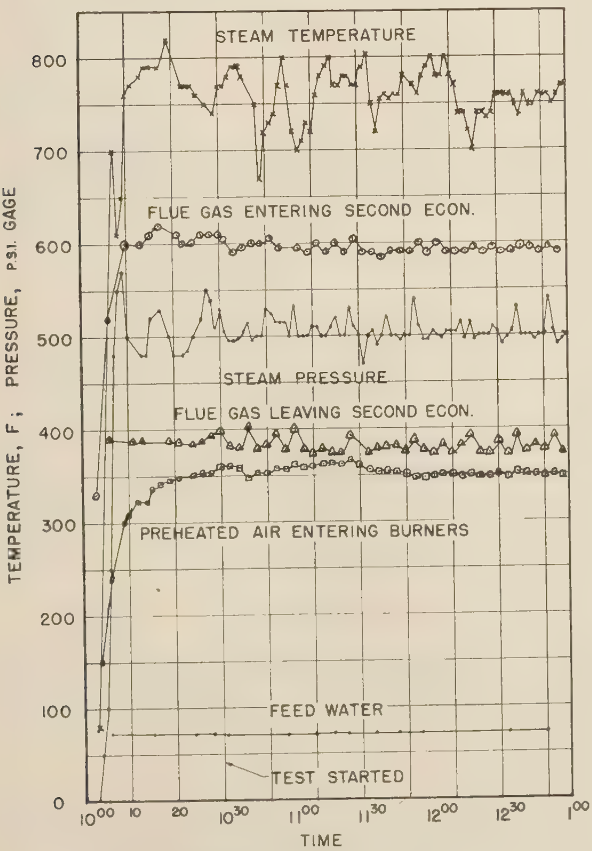

Fig. 11 Air, Flue-Gas, Steam, and Oil Data; Full-Load Test of Knox Boiler

Tests were run on the boiler on June 13, 1939, to determine the steam capacity, efficiency, and operating characteristics of the boiler at maximum smokeless output with four burners, each rated at 3 gph. In brief, the tests were conducted as nearly as possible in accordance with the A.S.M.E. Test Code for Oil-Fired Steam Generators, although the duration of the tests was less than that specified in the code. Table 3 shows the results.

TABLE 3 FULL-CAPACITY TEST

Test no	1
Date of test	6/13/39
Duration, hr	$2^1/_2$
Steam generated, lb	2966
Rate of generation, lb per hr	1186
Oil burned, lb	254
Evaporation, lb steam per lb oil	11.7
Pressure, average, psi abs	515
Temperature of steam, average, F	755
Enthalpy of steam, Btu per lb	1387
Temperature of feedwater, F	71
Heat added per lb, Btu	1348
Heating value of oil, Btu per lb	19400
Efficiency, per cent	81.3
Average CO_2 in flue gas, per cent	12
Air temperature, F	80
Flue-gas temperature, F	380
Loss to flue gas (approx), per cent	13.6
Loss to incomplete combustion, unburned carbon, etc. (by difference), per cent	5.1

The data for the tests are given in Fig. 11. It will be seen that the boiler was started from a cold condition at 10:03 a.m. The steam pressure began to rise almost at once and reached 500 psi within 4 min. The steam temperature rose with equal rapidity and reached 700 F within 4 min. The beginning of the test was deferred until 10:30 a.m. when equilibrium had been reached and all readings were steady.

The steam pressure was maintained at about 500 psi gage and the temperature at nearly 750 F. The variations in the steam temperature are due in part to difficulties with the water-level control, which altered the pressure of the water which was injected to control the steam temperature.

The air and gas temperatures varied but little during the test. The air entering the burners remained at 350 F, the flue gas entering the economizer at 590 F, and the flue gas leaving the second economizer remained at approximately 380 F. The feedwater temperature was constant at 71 F. The percentage of carbon dioxide in the flue gas ranged between 11 and 12.5 per cent.

The averaged results have been presented in Table 3. Some allocation of the losses can be made from the percentage of CO_2 and the flue-gas temperature. With an average CO_2 percentage of 12, the percentage of excess air was about 27 and the weight of dry flue gas produced per pound of oil burned was about 18 lb; 1.13 lb of water were produced by burning the hydrogen in the fuel. With an average flue-gas temperature of 380 F, losses were

	Per cent
Loss to water vapor	6.93
Unavoidable loss to dry gas	5.25
Avoidable loss to excess air	1.44
Total loss to flue gases	13.62
Percentage absorbed by water and steam (efficiency)	81.3
Unaccounted for (difference)	5.08

There was no smoke whatsoever during this test, and the boiler functioned smoothly. The test was terminated at 1:00 p.m. because it was felt that sufficient data had been obtained to give a reliable picture of the boiler performance.

The unit was quiet in operation, the two outer jackets serving to muffle the noise of the fan and the flame. There were no difficulties with the pressure control, nor any puffs when the burners went back into action after being shut off for a time. Ignition was positive and immediate.

Measurements for carbon monoxide in the flue gas were not

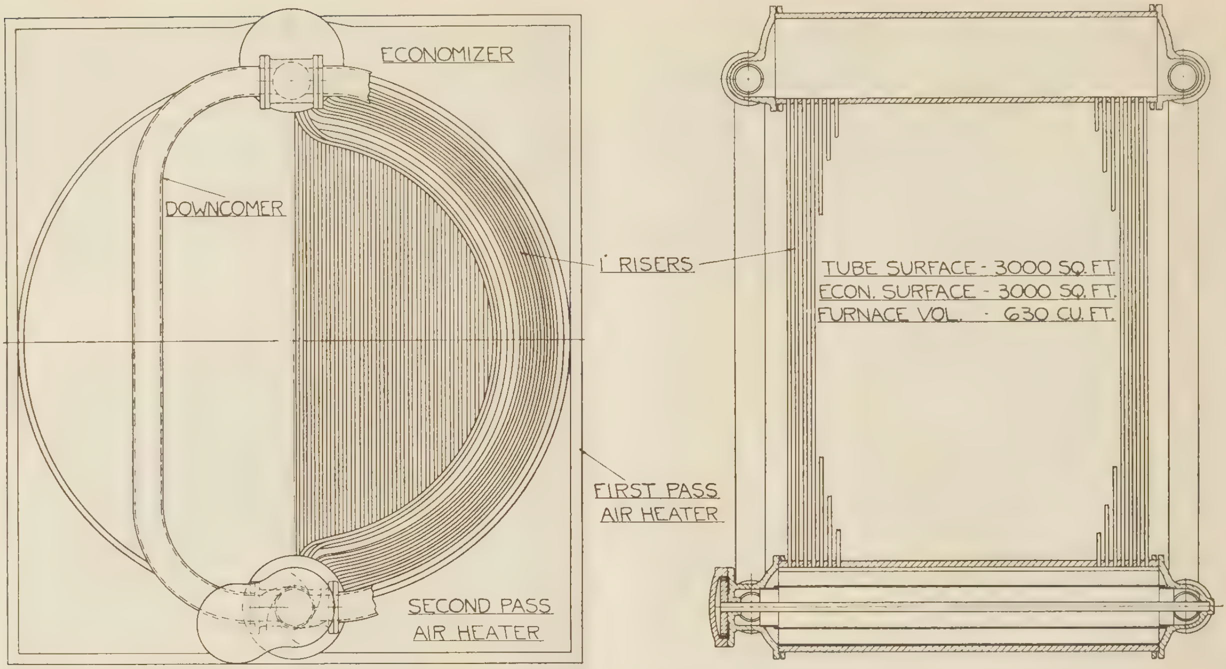

Fig. 12 Front and Side Views of Proposed Two-Drum Boiler

made during the test, since none had been found during the many earlier tests of the boiler at which the authors were present. In view of the large excess-air percentage, it is unlikely that any appreciable carbon monoxide existed.

There were no deposits of carbon in the combustion chamber, as Fig. 10 indicates. The refractory lining and the screen tubes were entirely free of carbon, and only a thin layer of soot adhered to the riser tubes. Furnace temperature was not measured but, judging from the color of the flame and from the heat release, it must have been very high.

No trouble was experienced with the packing on the impeller shaft, despite the fact that the pressure was between 500 and 600 psi, and the water must have been nearly saturated. Both fan and impeller were driven by the same 1.5-hp a-c motor.

After completion of the reported test, further development and tests of the boiler were conducted. During these tests, slightly over 1500 lb of water per hr were evaporated with smokeless combustion. While the boiler weighed 1235 lb, 300 lb of this were due to portions of the boiler installed for sound protection only, bringing the net weight of the boiler down to 935 lb. If correction is made for the amount of steam which would have been generated with the same fuel consumption, and at the same temperature, had the water been condensed and recirculated to the boiler at 200 F, it will be found that the weight of the boiler per pound of steam evaporated reduces to 0.56 lb.

The extremely high heat release, 430,000 Btu per hr per cu ft of combustion space, without harmful effects upon the tubes, refractory, or the general structure of the boiler, testifies to the effectiveness of the unique circulatory system used in the boiler.

The boiler as described was built with the special view of minimum over-all dimensions for a given output. Two water-level drums were used to obtain a maximum area of steam-release surface without the increase of height and weight which would have resulted had a single drum of equal steam-releasing surface been used. Where these limitations which controlled the design of this particular boiler are not essential, the boiler may be built with two drums instead of six, the downcomers consisting of large tubes connecting the upper and lower drums at their ends, all the small tubes being risers. Such a construction is shown in Fig. 12.

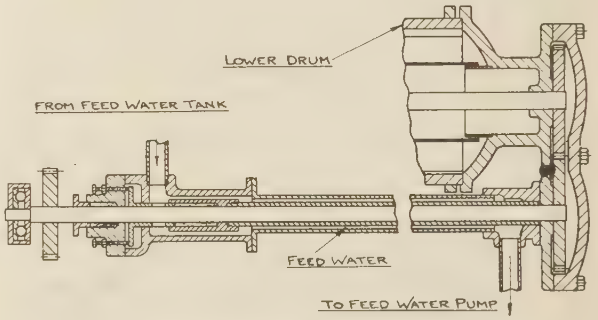

Fig. 13 Proposed Method of Driving the Impeller

Fig. 13 shows an improved method of driving the runner. The runner shaft carries a gear which connects with another below the driving shaft, running beneath the boiler and throughout its full length. This driving shaft is surrounded by a pipe carrying the feedwater. Thus, the packing is not subjected to high temperature, and could, in fact, be omitted, as the amount of water which would leak back along the driving shaft would be insignificant. Since it would go directly into the feedwater, there would be no loss of heat.

Conclusion

This paper has been devoted to a presentation in some detail of the unique features of an engine and a boiler, each of which is thought by the authors to embody certain advances over current practice. The test results are representative of the performance of these units.

Acknowledgment

The authors wish to express their appreciation of the assistance rendered by Kenneth Comey, of the Knox Engineering Company, and by Aaron Levine, of Stevens Institute of Technology.

Discussion

F. O. Ellenwood.[4] This paper is interesting and stimulating because it contains considerable information about a new engine and boiler, concerning which future progress reports will be welcome. The present performance data suggest interesting possibilities and it is sincerely hoped that future developments may place both the new engine and the new boiler on a successful production basis.

The test data show an engine efficiency of about 53 per cent, which is an excellent performance for an engine of this size. In this connection, will the authors indicate where the chief losses occur in this engine? In other words, what portion of the available energy is not utilized by reason of cylinder condensation, mechanical losses, thermal losses from the outside of the cylinder, and fluid-friction losses?

The new system of forced circulation in the boiler seems to be excellent and it is hoped that periods of long runs will still find it functioning properly. The improved method of driving the impeller, as indicated in Fig. 13 of the paper, seems to the writer to be a step in the right direction.

For a steam generator of this capacity to give an efficiency of 81 per cent, as has been shown by the test, means that great care has been taken in its design, and the writer desires to congratulate those who have been responsible for the development of this unit.

[4] Professor of Heat-Power Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

W. F. Ryan.[5] It is regrettable that the authors did not include in the published data the power consumption of the auxiliaries during the tests of this very interesting steam-generating unit.

In view of the relatively high draft loss and the power input to the circulating pump, it would ordinarily be assumed that the auxiliary-power consumption would be an excessive proportion of the limited output of the combined engine and boiler unit. If this is not the case it would naturally increase the value of the paper to indicate that fact by actual test results.

Authors' Closure

In reply to Professor Ellenwood's comments it is probable that a large portion of the losses in the engine are due to fluid friction. The unusual valve arrangement of this engine is possible only because of the somewhat involved paths through which the steam is led to the appropriate cylinder, and it is necessary to sacrifice some efficiency in order to attain the ends which were sought in the design.

In reply to Mr. Ryan, the power consumed by the a-c motor which drove both the fan and the impeller was about 0.75 hp. During a typical run, the current to the motor was 9 amp, at 220 v. The boiler feed pump which was used during the tests was driven by a large d-c motor for speed control, and the power which it required was far greater than that which would have been needed by a more efficient combination. In any event, the feed-pump power with this boiler should not be different from that with any other boiler of similar pressure and steam generation.

[5] Mechanical Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Fellow A.S.M.E.

A Study of Circulation in High-Pressure Boilers and Water-Cooled Furnaces

By JOHN VAN BRUNT,[1] NEW YORK, N. Y.

This paper is devoted to a discussion of the elements of boiler and furnace design dealing with the problem of obtaining adequate circulation for the higher pressures now demanded by current practice. After explaining the action within a simple evaporating circuit, the author indicates the procedure in analyzing waterwall circuits and calculating the circulation in boiler tubes.

Nomenclature

THE following nomenclature is used in the paper:

D_s = density of saturated water, lb per cu ft
D_d = density of downcomer mixture, lb per cu ft
D_r = density of riser mixture, lb per cu ft
d = tube diameter, ft
f = dimensionless friction factor
g = acceleration due to gravity, fps per sec
H = total heat absorbed by tube surfaces of furnace and boiler, Btu per hr
L = height from water level in drum to bottom of the circuit, ft
L_1 = distance from furnace floor, per cent of the furnace height
l = length of tube, ft
l_1, l_2 = see Q_1 and Q_2
Q = heat absorbed in distance L_1 as per cent of total heat absorbed by furnace
Q_1 = heat absorption by lowest portion l_1, of furnace-wall tube
Q_2 = heat absorption by greater portion l_2, of furnace-wall tube including l_1
S = summation of all losses due to flow velocity
V = linear velocity, fps
V_1 = linear velocity at entrance, fps
V_2 = linear velocity at exit, fps

Boilers in General

High-pressure steam-generating units of capacities and types installed and operated during the last few years are, with few exceptions, fired by pulverized coal in more or less completely water-cooled furnaces.

The problem of designers of such units is to provide a furnace of correct design to burn the specified fuel satisfactorily and, in addition to the furnace walls, such convection surface as is necessary to generate the required amount of steam. Superheating surface, economizer and air-heater surface must be provided and proportioned to give the desired over-all operating characteristics and efficiency.

It is not the purpose of this paper to go into the details of such proportioning of surfaces, but rather to bring out the elements of the circulating system and to show that the design discussed is both simple and adequate.

Natural circulation in a boiler may be defined as the movement of water, and steam and water, through the boiler tubes in conformity with the available head or difference in density of the circulating fluid in the downcomer and riser circuits.

Analysis of Simple Circuit

The simplest evaporating circuit is a U-tube Fig. 1(a), of uniform diameter and without restrictions in either the riser or downcomer leg. Such a circuit will, when heated on the riser leg, produce the maximum circulation, that is, it will pass through the circuit a maximum weight of water and steam for given conditions of tube diameter and head.

Any departure from this simple circuit which introduces resistance to flow, such as headers or junction boxes at X, or reduced area of tubes Y Fig. 1(b), will reduce the weight of water circulated. Heating the downcomer Fig. 1(d), or reducing the height, Fig. 1(c), will decrease the available head and, therefore, the flow.

Consider for example a simple U-tube circuit as in Fig. 1(a), having 30 ft of vertical heated length with the downcomer leg not heated, and passing water at the saturation temperature. Assume that the mean density of the mixture in this riser is 0.8 that of saturated water in the downcomer; then 24 ft of water in the downcomer will balance 30 ft of the mixture. Since there is 30 ft of solid water in the downcomer the static head is 6 ft. This 6 ft of head is available to overcome all friction and other resistances in the circuit. In the circuit considered, there are five losses; namely, entrance resistance, friction in the downcomer and in the riser, loss due to accelerating the mixture in the riser, and the exit loss from the riser. If these losses, expressed in feet head of saturated water, total 6 ft, the circuit is in equilibrium with 0.8 density in the riser circuit. If at an assumed velocity the resistances total but 4 ft, more water will flow through the circuit, thereby increasing the density in the riser and reducing the static head, and increasing the velocity and magnitude of the losses until equilibrium is established. Should the resistances at an assumed velocity total 8 ft, less water will flow through the circuit, thereby decreasing the density in the risers, increasing the static head, and reducing the velocity and resistances until equilibrium is reached.

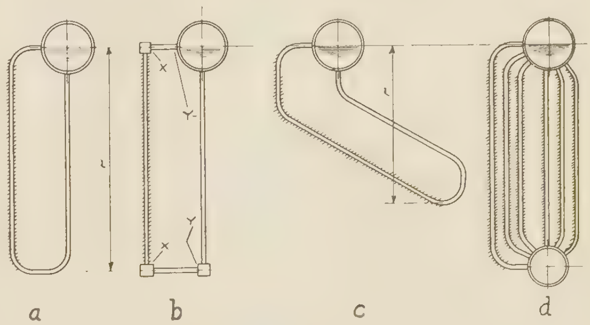

Fig. 1 Evaporating Circuits of Various Forms

If, in the same circuit, we assume that the downcomer is

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Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

heated and that the steam is generated in the downcomer in an amount sufficient to reduce the mean density of the downcomer mixture to 0.9 that of saturated water, this mixture will enter the riser in which additional steam is generated to give a mean density of 0.7 of saturated water. The equivalent static head is $(0.9 \times 30) - (0.7 \times 30) = 6$ ft, which as already noted is available to overcome resistances.

From the foregoing rough illustration, it will be seen that circulation will be stabilized at whatever density and velocity exists when all losses equal the static head; i.e., head in downcomer equals head in riser plus all losses expressed in head.

The term "dryness fraction" is the percentage of steam by weight in the mixture. "Top dryness" is the percentage by weight of steam at the top of the tube or at the point where heat input ceases.

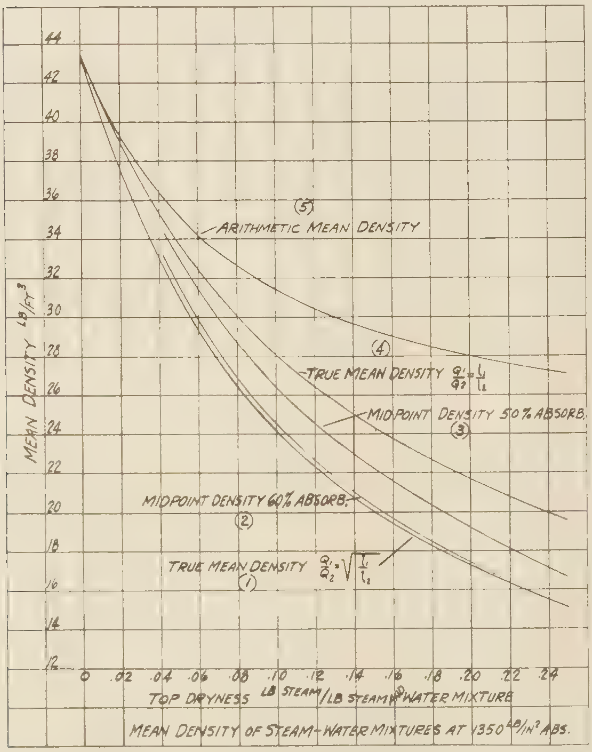


Fig. 2 Mean Density of Steam-Water Mixtures at 1350 Psi Abs

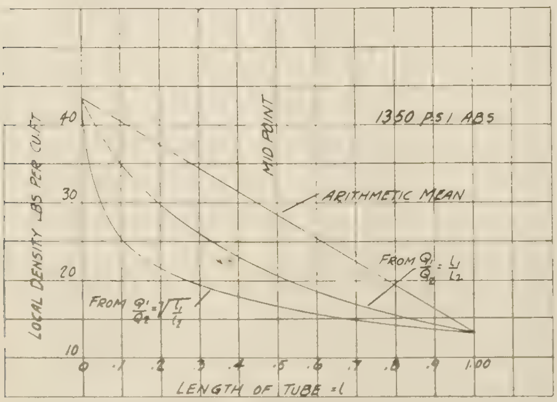


Fig. 3 Changes in Density Along Tube

Inasmuch as the static head available to overcome all resistances depends upon the density of fluid in the risers and downcomers, any analysis of circulation must begin with the determination of the density. This must be based on the correct rate of heat absorption by the exposed surface in the circuit being analyzed.

Density Versus Heat Absorption

Fig. 2, curve 1, gives the true mean density in a heated riser for heat absorption based on $\frac{Q_1}{Q_2} = \sqrt{\frac{l_1}{l_2}}$, or expressed in the percentage absorbed in any portion of the tube, Q (per cent) $= 10\sqrt{L_1}$.[2] Curve 2 shows the mid-point density based on 60 per cent of the heat absorbed in the bottom half of the tube. Curve 3 shows the mid-point density for 50 per cent heat absorbed up to mid-point. Curve 4 is the true mean density for uniform absorption, and curve 5 is the arithmetic-mean density. All of these curves are based on saturated water entering the bottom of the tube.

Fig. 3 illustrates the changes in density along the tube, the lowest curve is based on $\frac{Q_1}{Q_2} = \sqrt{\frac{l_1}{l_2}}$. The next curve corresponds to a heat input of $\frac{Q_1}{Q_2} = \frac{l_1}{l_2}$ and the top curve is the arithmetic mean of entrance and exit densities. The curves in Fig. 3 are plotted for saturated water at the bottom and a top-dryness fraction of 0.18, corresponding to a top density of 13.3 lb per cu ft, and a relative top density of 0.305 that of water at saturation temperature.

To visualize further the effect which these methods of calculating density have on the available static head, the available head for 100 ft of height of mixture when balanced against 100 ft of saturated water is shown in Fig. 4.

In analyzing the circulation in a furnace-wall circuit, one must start with the total heat absorbed at maximum load and, as may be seen from the curves in Figs. 2, 3, and 4, select the absorption

[2] "Factors Affecting Metal Temperatures of Furnace and Boiler Tubes," by W. S. Patterson, *Combustion*, August, 1940, pp. 24–29

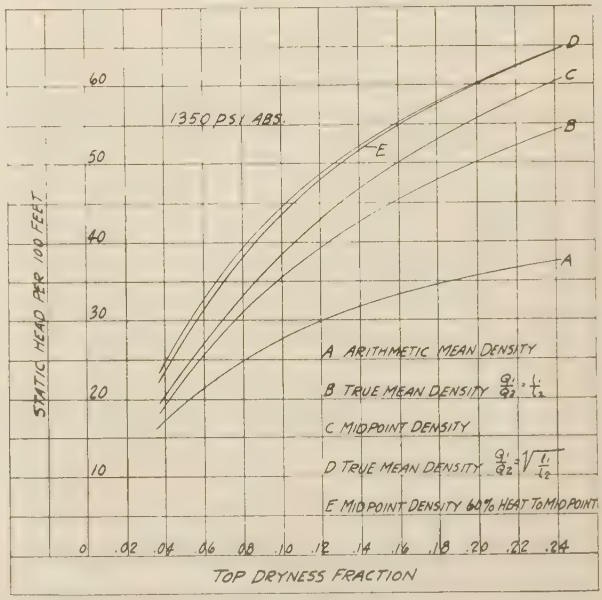


Fig. 4 Effect of Density-Calculating Methods on Available Static Head

rates along the tube which will give the correct density. Density calculated for conditions of curve 1, Fig. 2, will give the maximum available head. However, it is quite probable that ash or slag accumulations on the lower part of furnace walls will at times lower the absorption rate in the lower part of the furnace. Design must provide for these unfavorable operating conditions, such as producing minimum available head. These conditions may result in uniform absorption for each increment of tube length and, under some unusual furnace condition, perhaps localized on a few tubes, it is possible that the absorption in the upper half of the tube may exceed that in the lower half. Such a condition will result in a higher mean density than that shown by curve 4, Fig. 2.

Calculations for Furnace Tubes of Typical Unit

For purposes of analysis of waterwall circuits, a unit of 650,000 lb per hr capacity at 1350 psi pressure and 925 F total steam temperature is selected. Fig. 5 is a diagram of such a unit.

From Fig. 5, it will be noted that the rear waterwall risers are connected through the lower drum to the fourth and fifth rows of tubes. An alternate connection to the top drum is indicated by the dotted line joining the rear waterwall with the front row of boiler tubes. Either arrangement is satisfactory, the selection depending upon the spacing of front boiler tubes, which in turn is fixed by the number of superheater elements.

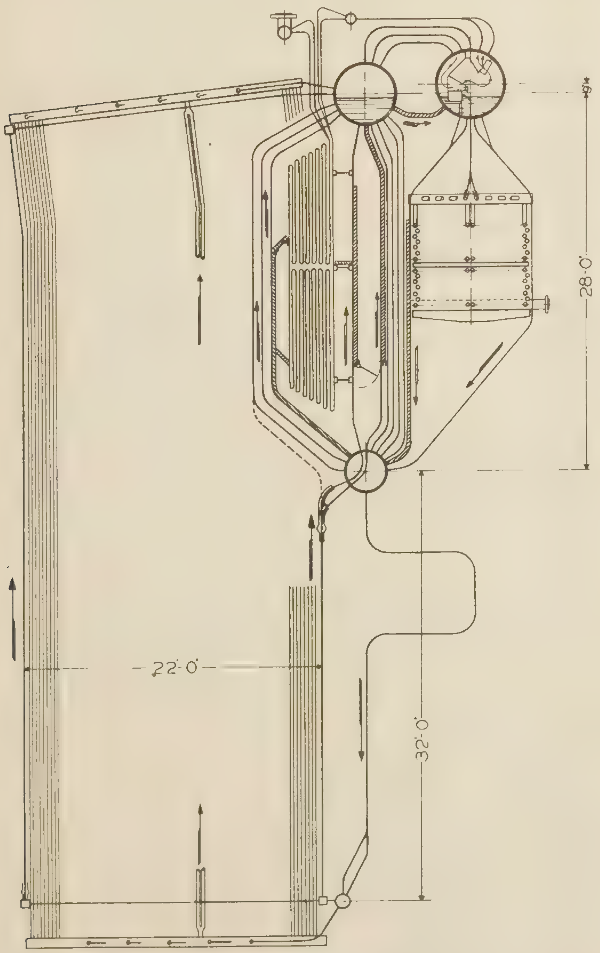


Fig. 5 Steam-Generating Unit of 650,000 Lb per Hr Capacity at 1350 Psi and 925 F Total Steam Temperature

Steam generated in the boiler downcomers returns to the upper drum through some of the tubes in the fifth row.

The feedwater temperature leaving the economizer is assumed to be 470 F. This feedwater will be raised to saturation temperature in a steam washer; to do this, 170,000 lb of steam per hr will be condensed in the washer. All of the water entering into circulation beyond the drum containing the washer will be at saturation temperature and, therefore, all heat absorbed by evaporating tube surface will produce steam. $H \div 584 =$ lb of steam (where H is the total heat absorbed by the unit and 584 is the heat of evaporation), and the total steam to be generated will be 650,000 + 170,000 = 820,000 lb per hr.

The furnace is 30 ft wide and the boiler 35 tubes wide. There are three rows of tubes ahead of the superheater and eight rows behind it, of which the fourth and fifth rows are risers, the sixth, seventh, eighth, and ninth downcomers, and the tenth and eleventh downcomers from the rear or offtake drum. Also, there are 35 horizontal water circulators between the two upper drums.

The furnace is so proportioned that 730,000 lb of steam per hr will be generated in the walls, 50,000 lb in the front three rows of boiler tubes in addition to that absorbed in those tubes by direct radiation from the furnace and, in the remaining eight rows 40,000 lb, of which approximately 25,000 lb will be evaporated in the four rows of downcomers, Nos. 6, 7, 8, and 9 and 12,000 lb in risers Nos. 4 and 5.

The wall-tube units are two 3-in. bifurcated tubes, each pair of 3-in. tubes being forged into a short $3^1/_4$-in. end at top and bottom. The bottom of the furnace is made up of 70 fin tubes 3 in. diam, the front and back walls of 110 tubes 3 in. diam or 55 pairs of 3-in. bifurcated units, and the side walls of 80 tubes 3 in. diam or 40 bifurcated units. The four furnace walls are supplied by 72 downcomers $3^1/_2$ in. diam between the lower drum and the round distributing header.

From the distributing header 70 nipples 3 in. diam lead to the rear waterwall header and 20 tubes $3^1/_2$ in. diam to each of the lower side-wall headers. The upper front-wall header is connected to the front drum by 70 fin tubes 3 in. diam and the upper side-wall headers by 30 tubes 3 in. diam.

Using an average absorption rate of 70,000 Btu per sq ft projected area of furnace wall per hr, the evaporation in each 60 ft of 3-in. front and side-wall tube will be $\dfrac{0.25 \times 60 \times 70{,}000}{584} = 1800$ lb of steam per hr. Using this figure, the steam generated in the furnace walls will be as follows:

Front wall, 110 × 1800 = 198,000 lb; rear wall 110 × 1800 = 198,000 lb; side walls 80 × 2 × 1800 = 288,000 lb. The roof of the furnace with an estimated absorption rate of 40,000 Btu per sq ft per hr will evaporate 45,500 lb per hr. This makes a total evaporation of 729,500 lb per hr. The remaining 90,000 lb of steam will be generated by convection in the eleven rows of boiler tubes as previously described.

An analysis will be made of the circulation in the front waterwall, side wall, and front row of boiler tubes only. These will serve as examples of the method which can be applied to any circuit in the boiler.

Briefly stated, the problem is to determine the equilibrium point at which the available static head is equal to the sum of all losses in head due to velocity

$$\left(\frac{D_d}{D_s} \quad \frac{D_r}{D_s}\right) \times L = S$$

in which

D_s = density of water at saturated temperature, lb per cu ft
D_d = density of mixture in downcomer, lb per cu ft

D_r = density of mixture in riser, lb per cu ft
L = length (height) from water level in drum to bottom of circuit, ft
S = sum of all resistances

To determine the total amount of water entering the bottom of all evaporating tubes divide the total weight of steam generated by the assumed top-dryness fraction. From the weight of water thus found subtract the weight of steam to find the amount of water discharged into the front drum. Similarly, to find the amount of water entering the furnace-wall tubes, divide the weight of steam generated in the walls by the assumed top-dryness fraction.

From the weight of water discharged by the front drum, subtract the amount flowing through the horizontal water circulators to the rear drum to obtain the weight flowing through the boiler downcomers.

From these flows, the velocities of water in the boiler and waterwall downcomers is obtained for any top dryness. The water to the furnace walls is assumed to be distributed to the four waterwalls in proportion to the calculated requirements.

These flows and velocities for front and side-wall circuits are given in Table 1, as well as other necessary velocities, also mean densities, top densities, and available head. Mean densities are taken from curve 4, Fig. 2.

TABLE 1 FLOWS AND VELOCITIES, ETC., FOR FRONT AND SIDE-WALL CIRCUITS

Top-dryness fraction (assumed)...	0.135	0.15	0.165	0.18
Steam generated, lb per hr........	820000	820000	820000	820000
Water to evaporating surface, lb per hr........................	6080000	5460000	4970000	4550000
Steam to rear drum, lb per hr......	820000	820000	820000	820000
Water discharged to front drum, lb per hr....................	5260000	4640000	4150000	3730000
Water flow to rear drum, lb per hr...	1330000	1330000	1330000	1330000
Water to boiler downcomers, lb per hr........................	3930000	3310000	2820000	2400000
Water to waterwall downcomers, lb per hr.....................	5400000	4875000	4450000	4050000
Velocity in boiler downcomers, fps.	5.95	5.05	4.35	3.65
Velocity in waterwall downcomers, fps..........................	11.6	10.5	9.55	8.75
Velocity in 18-in. connecting nipples, fps.......................	8.9	8	7.3	6.7
Velocity in floor tubes, fps.........	4.45	4	3.65	3.35
Velocity in side-wall downcomers, fps..........................	8.1	7.35	6.65	6.1
Velocity in $3^1/_4$-in. bifurcated ends, fps..........................	4.72	4.25	3.83	3.54
Velocity in 3-in. tubes at bottom, fps..........................	2.83	2.55	2.32	2.12
Mean density in 3-in. wall tubes, lb per cu ft..................	25.25	24.37	23.5	22.7
Velocity at point of mean density, fps..........................	4.87	4.55	4.28	4.15
Density at top of 3-in. wall tubes, lb per cu ft..................	17.03	15.05	14.1	13.3
Velocity at top of 3-in. wall tubes, fps..........................	7.7	7.4	7.15	6.9
Velocity entering roof tubes, fps...	12.1	11.7	11.5	10.85
Density leaving roof tubes, lb per cu ft.........................	14.3	13.3	12.45	11.8
Velocity leaving roof tubes, fps.....	13.5	13.1	12.7	12.4
Available head, assuming saturated water in downcomer, ft.........	25.2	26.3	27.5	28.7
Velocity in side-wall risers, fps.....	15.4	14.8	14.3	13.8

From these velocities and densities, the various entrance, exit, friction, and acceleration losses are determined. Considering a circuit from the front drum through downcomers and up through the front wall, there are twenty-one distinct losses, including entrance, exit, and friction in boiler and waterwall downcomers, in 18-in. nipples, in floor tubes, in furnace-wall risers, and in roof tubes; in addition there are acceleration losses in boiler downcomers, furnace-wall risers, and in roof tubes. Entrance resistance cannot exceed $\frac{1}{2}\frac{V^2}{2g} \times$ relative density. Exit resistance cannot exceed $\frac{V^2}{2g}$ times relative density. For friction loss the Fanning equation is used. Friction loss $\frac{4f\,l\,V^2}{d \times 2g} \times$ relative density. Relative density is evaluated at the point where velocity is calculated.

V = velocity at point of mean density, fps
g = 32.2 fpsps
f = 0.006
l = length of tube, ft
d = diameter, ft

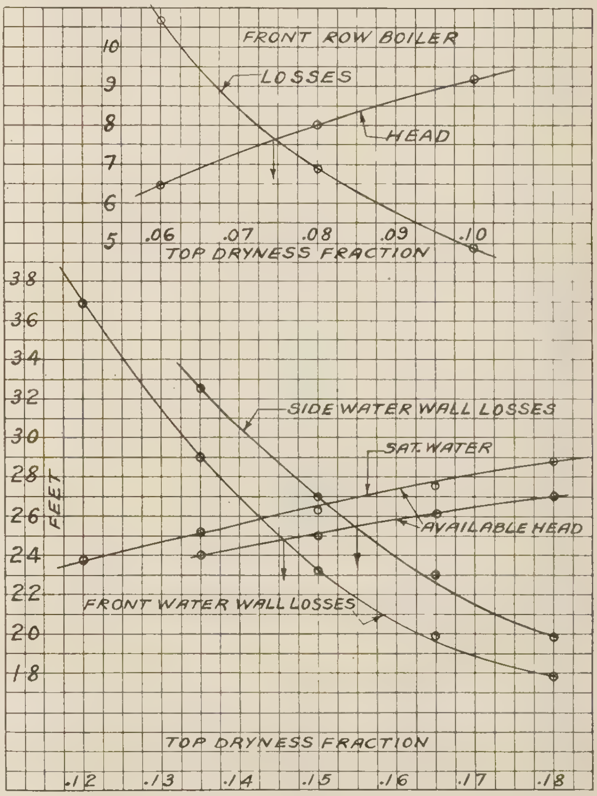


Fig. 6. Loss in Head for Various Top-Dryness Fractions

TABLE 2 RESISTANCES DUE TO VELOCITY

	Entrance	Exit	Friction	Acceleration	Total front	Total side
Boiler downcomers, ft....	0.147	0.294	0.97	..	1.41	....
Waterwall downcomers, ft	0.707	1.415	7.85	..	9.92	11.33
18-In. nipples, ft.........	0.415	0.830	0.167	..	1.41	....
Floor tubes, ft..........	0.104	0.208	0.228	..	0.544	....
60-In. wall tubes, ft......	0.228	0.730	1.15	0.35	2.458	....
Roof tubes, ft...........	0.665	0.72	2.04	0.44	3.86	....
Side-wall downcomers, ft.	0.345	0.69	2.7	..	....	3.735
60-Ft side-wall tubes, ft..	0.228	0.730	1.15	0.35	....	2.458
Side-wall risers, ft.......	0.955	0.955	3.6	..	....	5.5
Total front-wall circuit, ft..................					19.602	
Total side-wall circuit, ft..................						23.023

The acceleration loss is $\frac{V_1}{32.2}(V_2 - V_1)$, where V_1 = velocity at entrance and V_2 = velocity at exit.

There may be some differences of opinion as to the correct or most logical method of figuring friction losses in heated tubes. In this paper, the friction factor of 0.006 is used, although 0.005 is generally accepted. It can be seen, however, that even if the friction losses in the riser tubes are doubled, the top-dryness fraction of both side and front walls will still be less than 0.17.

As an example, an analysis of flow through the front and side-wall circuits with a top-dryness fraction of 0.165 will be taken. For this condition, the velocities, head, densities, etc., will be as given in column 3, Table 1.

Because of the change in area, where the $3^1/_4$-in. bifurcated end joins the wall tubes and the change in direction and turbulence, the entrance resistance in the bottom and top wall header will be figured as $\frac{V^2}{2g}$ instead of $\frac{1}{2}\frac{V^2}{2g}$

From the flow and velocities in column 3, Table 1, the resistances in the various elements of the circuits in feet of water at saturation temperature are calculated and listed in Table 2.

Similar calculations for top-dryness fractions of 0.135, 0.15, and 0.18 may then be plotted, as in Fig. 6. At the intersection of the available head and resistance curves, the circuit is in equilibrium and the corresponding top-dryness fractions that exist may be read from the curve.

Calculations for Boiler Tubes

The same method is applied to the circulation in the front bank of boiler tubes. Taking one of the front rows of boiler tubes, the evaporation is first determined by adding to the heat input by radiation, the convection transfer, and dividing by the latent heat. The evaporation per tube is approximately 2400 lb per hr. Again, as with the furnace tubes, this weight is divided by the assumed top-dryness fraction to find the weight of water entering the bottom of the tube.

As 25,000 lb of steam is generated in the boiler downcomers, the mean density in the downcomers is determined from the amount of steam generated and the flow through downcomers corresponding to the assumed top-dryness fraction. Velocities through the entire circuit and other necessary data are given in Table 3.

At 0.18 top-dryness fraction 2,400,000 lb of water flow through the boiler downcomers. The mean density in the downcomers is approximately the same as the mid-point density, thus 2,400,000 —12,500 = 2,387,500 lb of water per hr. The volume of steam and water = 58,875 cu ft; density = 2,400,000 ÷ 58,875 = 41 lb per cu ft. Relative density = 41 + 43.5 = 0.94. For other dryness fraction the densities are 41.2, 41.5, and 41.8, and the relative densities 0.945, 0.955, and 0.962.

The tubes in the front row of the boiler are 3 in. diam and approximately 34 ft long. There are only four resistances; namely, entrance, exit, friction, and acceleration. From inspection of the losses calculated for the waterwall circuits, it is obvious that with top-dryness fractions of 0.15 or more the available head will be greatly in excess of that required. To plot the equilibrium curves for the circuit, assumptions of 0.10, 0.08, and 0.06 top-dryness fractions will be used. No attempt will be made to correct the flow of water through the boiler downcomers to correspond to these assumed dryness fractions as the change in relative density would be small. The essential data are given in Table 3.

The available heads and losses are plotted at the top of Fig 6 and the head and loss curves drawn through the points. The intersection will be found at 0.075 top-dryness fraction.

The upper available head curve Fig. 6, is based on saturated water in the downcomers. Correcting this curve for downcomer densities of 0.94, 0.945, 0.955, and 0.962, respectively, at

TABLE 3 FLOW DATA, FRONT ROW OF BOILER TUBES

Top-dryness fraction (assumed)	0.06	0.08	0.10
Lb of steam per hr per tube	2400	2400	2400
Amount of water entering bottom of tube, lb	40000	30000	24000
Volume of water, cfs	0.256	0.192	0.1535
Velocity entering, fps	8.45	6.4	5.12
Mean density from curve 4, Fig. 2, lb per cu ft	32.4	30.0	28.0
Velocity at mean density point, fps	11.35	9 3	7 96
Velocity at exit, fps	15.0	12.9	11.7
Density at exit, lb per cu ft	24.5	21.8	19.0
Head in downcomer, saturated water, ft	27.3	27.2	27.1
Head of saturated water in riser, ft	20.85	19.2	17.9
Available head, saturated water, ft	6.45	8.0	9.2
Entrance loss, saturated water, ft	0.55	0.32	0.203
Exit loss, saturated water, ft	2.18	1.30	0.930
Friction loss, saturated water, ft	6 20	3.95	2.650
Acceleration loss, saturated water, ft	1 71	1.30	1.040
Total losses, ft	10.64	6.87	4.823

0.135, 0.15, 0.165, and 0.18 top dryness, the head will be as in the lower curve. The top-dryness fractions at the equilibrium points are 0.145 for the front wall and 0.155 for the side wall.

It is possible to equalize the flow velocity in side and front walls either by increasing the resistance in the front-wall circuit, or decreasing the resistance in the side-wall circuit.

Conclusions

It appears from the foregoing analyses that the circulation in high-pressure units of the type illustrated is adequate for the maximum load conditions and with a comfortable margin for substantial overloads.

Similar analyses of furnace-wall tubes $2^1/_2$ in. outside diam show equally adequate circulation.

Should it be necessary to reduce the losses below the figures submitted herein, downcomer areas should be increased.

If the height of boiler furnace is decreased, the available head and friction losses in the downcomers and risers also decrease. All inlet and outlet losses and friction losses in the horizontal members are unchanged; therefore, there is a lower limit of height below which the circulation would become progressively less until the top-dryness fraction approaches unity, under which conditions the evaporating tubes would fail from overheating. It is obvious, therefore, that for high-capacity units of low height, natural circulation may become inadequate.

For those who are interested, a more complete analysis of circulation flow, reference is made to a recent paper,[3] by W. Yorath Lewis and Struan A. Robertson.

Discussion

E. G. Bailey.[4] The author has shown the application of the well-known fundamental laws of boiler circulation to a large high-pressure steam-generating unit of recent design. This undoubtedly will give many engineers a clearer picture of the problems involved in calculating circulation than they have had from the simpler illustrations available in the engineering literature.

While the author has made clear his assumptions and his general procedure in carrying out calculations, the writer believes it should be brought out even more clearly that assumptions should be correct, if the final calculations are to be in keeping with the actual results from the unit in operation.

It should be further emphasized that experimental results from actual tests should be used to confirm or modify some of the more important assumed factors. The author amplifies the different methods of calculating mean density of steam-

[3] "Circulation of Water and Steam in Water Tube Boilers and the Rational Simplification of Boiler Design," by W. Yorath Lewis and Struan A. Robertson, Proceedings of The Institution of Mechanical Engineers, London, England, vol. 143, June, 1940, pp. 147–175.

[4] Vice-President, The Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

water mixtures, which is the fountainhead of boiler circulation. From actual field data, it is believed that the effective circulating head is likely to be less than any of the assumptions he has made, and certainly less than the "true mean density" used in the final calculations. This difference is believed to be partly due to the relative velocity of the steam and the water in the mixture. The mean density of any given circuit depends a great deal upon the relative heat absorption in different parts of that circuit. That of itself is a problem which requires considerably more data from field tests, because there are a great many diverse conditions in different types of units, different types of firing, and different ash and slag conditions.

The author has introduced an expression "dryness fraction" which will be new to many engineers. It is in reality "per cent of steam by weight" or the reciprocal of one often used, which is the "mixture-to-steam ratio." Another term "pounds of water per pound of steam" is often used. Still another term "per cent of steam by volume" has much greater significance. It might be well for some consideration to be given to standardizing terms, so that we will all talk the same language.

Units of the type described in the text, having both boiler and waterwalls supplied from a common source, must be designed so that the proper circulation will obtain in each circuit when all circuits are in equilibrium. This cannot be assured without solving all circuits simultaneously.

In his conclusions, the author states that losses are reduced by increasing downcomer areas. The fact should also be mentioned that, in some types of circuits, this can be brought about to better advantage by increasing the riser areas.

The fourth paragraph of the conclusions seems somewhat confusing because the friction losses in the horizontal members are actually reduced when the total circulating flow is reduced by the lower height of the heat-absorbing circuits, although the resistance of the horizontal members may be unchanged.

Fred Dornbrook.[5] In his conclusions the author indicates that circulation is adequate "until the top-dryness fraction approaches unity." He probably does not wish to convey the impression that high-pressure high-capacity designs can employ, for instance, 99 per cent steam and 1 per cent water in the risers under peak conditions.

It is submitted that, for the high reliability requirements of modern high-pressure high-capacity boiler units, circulation requires more involved study than simply a calculation of average circulation velocities and dryness factors. Perhaps the author will tell more of his experience in regard to limiting velocities, dryness fractions, and heat-transfer rates.

M. H. Kuhner.[6] The author's treatment of the theory of available force to produce natural circulation for higher-pressure steam-generating units is a valuable addition to boiler-design information. Such analyses are necessary to prevent the operating difficulties of higher-pressure installations, resulting from insufficient circulation, as reported for some installations placed in service during the last few years.[7]

The author may wish to amplify his conclusions by mentioning that entrance and discharge losses of tubes connected to boiler drums and waterwall headers are difficult to determine. These losses depend largely upon the density of the fluid and direction of flow in reference to entrance and discharge openings and the possible turbulence existing in drums and headers. From the standpoint of practical design, it may be suggested that waterwall headers be made large in internal areas and that the water supply to bottom headers and the offtakes from the upper headers be liberal in area and uniformly distributed over the length of these headers. This for the purpose of obtaining minimum disturbance of flow in the vicinity of entrance and discharge of the steaming tubes.

In applying the author's theories to practice, it should further be pointed out that the distribution of heat to individual groups of steaming tubes, such as the group of side waterwall tubes or rear waterwall tubes, is not uniform for each single tube. Those tubes placed closest to the source of heat, such as the tubes in the middle of the side waterwalls or those directly opposite the burners in the rear wall of the furnace, may contain a steam-water mixture of considerably lower mean density than other tubes of the same group placed in the corners of the furnace, or which may be otherwise shielded from direct radiation. The mean top-dryness fraction of the individual group of tubes may show to be safe while at the same time the top-dryness fraction in a few of the tubes may be unity, so that these tubes are overheating. The same problem applies to the steaming tubes of the boiler. It is known that those tubes placed over the center of the furnace are subjected to higher radiation and gas temperatures than the tubes near the side walls.

It would be interesting to apply the author's theoretical investigation of circulation to one or more of the installations discussed in E. P. Partridge and R. F. Hall's paper.[7]

The further conclusion, drawn from the author's paper and applied to practice, shows the importance of proportioning the flow areas in circulatory systems of waterwalls and boilers of higher-pressure installations as liberally as practical and with a minimum of obstructions or changes in cross-sectional areas, so that the losses of flow in tubes, headers, and drums be kept at a minimum.

Another important factor influencing the available force producing natural circulation is the density of the fluid in the downcomer tubes. The buoyancy of steam formed in downcomers tends to oppose downflow of water. Therefore, downtake tubes must not be exposed to high gas temperatures.

It would appear, that, if a given boiler unit should be operated at various pressures, a particular dryness fraction would be realized for each particular pressure. A definition by the author of the force governing this relation would be interesting.

E. F. Leib.[8] If the heat absorption in parallel circuits is not equal for each circuit, the possibility exists that the flow rate through certain tubes varies widely from the flow through the others; under certain conditions, the flow can reverse and a riser may operate as a downcomer. This is due to the circumstance that various flow rates may result in the same pressure difference. This condition is referred to as instability. In the following, two methods are outlined to examine whether, for a given case, the same pressure difference which is maintained by the flow rate through the bulk of the tubes may also correspond to other flow rates.[9] Uniform heat input over the entire tube length is assumed.

Forced-Flow Tubes

This system has no recirculation and no economizer. The tubes are very long. Therefore, pressure losses other than due to friction can be neglected. The direction of flow is assured, but the rate may be undetermined. The feedwater enters

[5] Chief Engineer of Power Plants, Wisconsin Electric Power Company, Milwaukee, Wis. Mem. A.S.M.E.

[6] Chief Mechanical Engineer, Riley Stoker Corporation, Worcester, Mass. Mem. A.S.M.E.

[7] "Attack on Steel in High-Capacity Boilers as a Result of Overheating Due to Steam Blanketing," by E. P. Partridge and R. F. Hall, Trans. A.S.M.E., vol. 61, 1939, pp. 597–621.

[8] Combustion Engineering Company, Inc., New York, N. Y.

[9] "Unstabilität der Strömung bei natürlichem und Zwangsumlauf," by M. Ledinegg, *Die Wärme*, 1938, pp. 891–898.

cold, is heated to saturation temperature in the lower section of the tube, and is completely evaporated in the upper section. The relative length of both sections depends upon the specific flow, i.e., the flow rate divided by the heat input. Since the friction loss is less in the section containing the liquid than in the section containing the steam-water mixture, it is possible that the pressure loss decreases while the saturation point moves upward due to an increasing flow rate. The relation between flow and friction is obtained as follows:

The force of friction is assumed proportional to the kinetic energy of the flowing fluid and to the area of friction between fluid and tube, hence

$$dF = f \times \frac{\rho w^2}{2} \times 2\pi R dl$$

where f = friction factor, ρ = density of fluid, w = flow velocity, $R = \frac{D}{2}$ = tube radius, l = tube length.

Then, the pressure due to this force is obtained by division by the cross area

$$dP = \frac{dF}{\pi R^2} = \frac{f}{R}\,\rho w^2 dl \ldots\ldots\ldots\ldots [1]$$

Substituting the specific volume $v = \frac{1}{g\rho}$ and introducing the flow rate

$$G = \pi R^2 \frac{w}{v} \text{ lb per sec}$$

Equation [1] can be written

$$dP = \frac{fv}{g\pi^2 R^5}\,G^2 dl \ldots\ldots\ldots\ldots [2]$$

From this equation the pressure drop can be calculated for both tube sections. Over the entire tube length L the heat Q is supplied in unit time. If the heat necessary to bring the water to saturation temperature is H Btu per lb, then the length of the lower tube section L' is

$$L' = L\frac{GH}{Q}$$

The specific volume of the fluid nearly equals the volume of saturated water v'. Then, the pressure drop in the lower section is

$$\Delta P' = \frac{fv'L}{g\pi^2 R^5}\,\frac{H}{Q}\,G^3 \ldots\ldots\ldots\ldots [3]$$

The volume of the steam-water mixture in the upper tube section is

$$v = x(v'' - v') + v' \ldots\ldots\ldots\ldots [4]$$

where v'' = specific volume of saturated steam.

For uniform heat absorption, the ratio of the dryness x at the point l to the dryness x_0 at the end equals the ratio of the pertaining lengths of the upper tube section

$$\frac{x}{x_0} = \frac{l - L'}{L - L'}, \ldots\ldots\ldots\ldots [5]$$

All heat, added in the upper section, serves to supply the heat of evaporation r. Then, the total amount of steam generated is given by

$$Grx_0 = Q\,\frac{L - L'}{L} \ldots\ldots\ldots\ldots [6]$$

From Equations [5] and [6], the local dryness fraction is obtained as

$$x = \frac{Q}{Gr}\,\frac{l - L'}{L} \ldots\ldots\ldots\ldots [7]$$

and the local volume of the steam-water mixture from Equation [4] is

$$v = \frac{Q}{Gr}\,\frac{l - L'}{L}\,(v'' - v') + v' \ldots\ldots\ldots\ldots [8]$$

If this value is substituted into Equation [2], integration of this equation between the limits L' and L then gives the pressure drop in the upper section as

$$\Delta P'' = \frac{fLv'G^2}{2g\pi^2 R^5}\left(1 - \frac{GH}{Q}\right)\left[\frac{v'' - v'}{v'}\left(\frac{Q}{Gr} - \frac{H}{r}\right) + 2\right] \ldots [9]$$

and the pressure drop through the entire tube is

$$\Delta P = \Delta P' + \Delta P'' = \frac{fLv'Q^2}{2g\pi^2 R^5} \times \left[\frac{BH^2}{r}\left(\frac{G}{Q}\right)^3 - 2\left(\frac{BH}{r} - 1\right)\left(\frac{G}{Q}\right)^2 + \frac{B}{r}\,\frac{G}{Q}\right] \ldots [10]$$

where $B = \frac{v'' - v'}{v'}$.

From this relation the pressure drop, for constant values H, can be represented by curves of the type shown in Fig. 7 of this

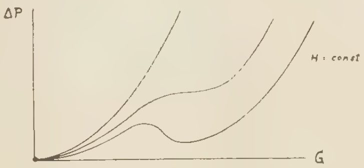


FIG. 7 CURVES REPRESENTING PRESSURE DROP FOR CONSTANT VALUES OF H

discussion. For small values H, there is a monotonous rise of pressure drop with the flow rate. At a certain higher value H, the curve shows a horizontal point of inflection, which indicates the transition from the stable region to the unstable region. For still higher values H, the curve has a maximum and a minimum; between these, the pressure drop decreases while the flow increases. In this region, three different values of the flow rate result in the same pressure drop; any one of these flow rates may exist in such a tube, when the pressure drop is given by the operating conditions of the other tubes. The location of the maximum, minimum, and point of inflection depends only on the specific-flow rate, $\frac{G}{Q}$ (lb per Btu). The value H, above which the region of instability begins, is that for which the first and second derivatives of Equation [10] are zero. It is found by differentiation that

$$H = (4 + \sqrt{12})\,\frac{r}{B} = 7.464\,\frac{r}{B} \ldots\ldots\ldots\ldots [11]$$

while the pertaining value of the specific flow is

$$\frac{G}{Q} = \frac{\sqrt{12} - 3}{6} \times \frac{B}{r} = 0.07735\,\frac{B}{r} \ldots\ldots\ldots\ldots [12]$$

Further improvement may be applied to these calculations by assigning to the friction factor (which is here assumed as con-

stant) a certain function of the Reynolds number and thus handling f as dependent upon G also.[10]

It must be determined whether the range of instability can be reached under usual operating conditions. Since complete evaporation of the water was assumed, the flow rate which satisfies the instability condition must not be higher than the amount of water which can be evaporated by the heat quantity Q, namely

$$\frac{G}{Q} = \frac{1}{H + r} \qquad [13]$$

The curves for $\frac{G}{Q}$ from Equations [12] and [13] are plotted in Fig. 8 of this discussion, against the saturation pressure. It

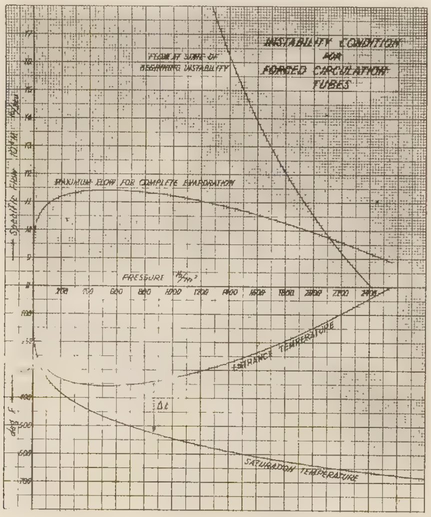


FIG. 8 INSTABILITY CONDITION FOR FORCED-CIRCULATION TUBES

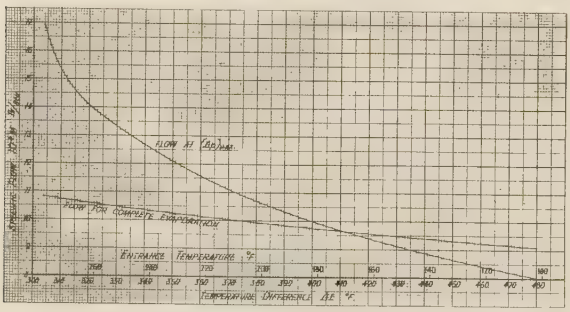


FIG. 9 INSTABILITY IN FORCED-CIRCULATION TUBES AT 1350 PSI

follows from the diagram that only for pressures higher than 2120 psi the range of beginning instability is reached. In the lower part of Fig. 8 are plotted the entrance temperatures of the water belonging to the values H, as calculated from Equation [11] for the same pressures. It is shown that, for those pressures where the range of instability could be reached, the entrance temperatures are below 120 F, a condition which is quite unlikely to be encountered. Since, for still higher values H the maximum of the ΔP curves moves to smaller values G, there is still a possibility that the range of instability may be reached. This check is made in Fig. 9 of this discussion for 1350 psi pressure. The specific flow $\frac{G}{Q}$, when ΔP has a maximum, has been calculated for various values H by differentiation of Equation [10] and plotted against the corresponding entrance temperatures, together with the curve of the amount of water which can be evaporated according to Equation [13]. It follows from Fig. 9 that, only for entrance temperatures below 170 F, the range of instability can be reached at this pressure. It has thus been shown to be quite unlikely in forced-flow tubes that the range of instability may be reached under normal operating conditions.

[10] "Stabilität der Strömungsverteilung in Heizflächen mit Zwangsdurchlauf," by A. Kleinhans, *Archiv, für Wärmewirtschaft*, 1939, pp. 135–138.

NATURAL-CIRCULATION TUBES

The problem has a different aspect in a natural-circulation system. Only a fraction of the circulating water is being evaporated. The entering water has been heated in an economizer to a temperature approaching saturation. Therefore, the saturation point can be assumed to be at the tube entrance, and a steam-water mixture of varying dryness fills the entire tube. The direction of the flow is not assured and reversion of flow may occur, but only one value of the flow rate is possible as long as the fluid flows upward. The pressure difference between the lower and upper header consists now of the static head, the entrance loss, the acceleration loss, and the friction loss. We consider a circuit through a system of waterwall (vertical) tubes. For upward flow, the three losses diminish the pressure with the height; they act in the same direction as the static head, and the pressure drop due to the losses must be added to that due to the head. For downward flow, the losses act in the opposite direction to that of the static head, and the pressure drop due to them must be subtracted from that due to the head. The pressure difference due to the static head is

$$\Delta P_s = \int_0^L \frac{dl}{v} \qquad [14]$$

where v has to be substituted from Equation [4]. In this case ($L' = 0$) the ratio of the local dryness to the top dryness is

$$\frac{x}{x_0} = \frac{l}{L}$$

and the amount of steam generated is

$$Gx_0 = \frac{Q}{r} \qquad [15]$$

Then the dryness at the length l is

$$x = \frac{Q}{Gr}\frac{l}{L}$$

If this value is substituted into Equation [14], integration gives the static head as

$$\Delta P_s = \frac{GLr}{BQv'} \ln\left(1 + \frac{BQ}{Gr}\right) \qquad [16]$$

The entrance loss is

$$\Delta P_e = \frac{v'G^2}{2g\pi^2R^4} \qquad [17]$$

The acceleration loss is

$$\Delta P_a = \frac{x_0(v'' - v')}{g}\left(\frac{G}{\pi R^2}\right)^2$$

or with the use of Equation [15]

$$\Delta P_a = \frac{v'BQG}{g\pi^2 R^4 r} \quad \text{[18]}$$

The friction loss is obtained from Equation [9] if we take $H = 0$

$$\Delta P_f = \frac{fLv'G^2}{2g\pi^2 R^5}\left(\frac{BQ}{Gr} + 2\right) \quad \text{[19]}$$

The total pressure difference is

$$\Delta P = \Delta P_s \pm (\Delta P_e + \Delta P_a + \Delta P_f)$$

where the plus sign is for riser tubes and the minus sign for downcomer tubes. From Equations [16], [17], [18], and [19] follows the total pressure difference

$$\Delta P = \frac{LrG}{BQv'} \ln\left(1 + \frac{BQ}{Gr}\right) \pm \frac{v'}{g\pi^2 R^4}\left[\left(\frac{1}{2} + f\frac{L}{R}\right) G^2 + \left(1 + \frac{1}{2} f \frac{L}{R}\right) \frac{BQ}{r} G\right] \quad \text{[20]}$$

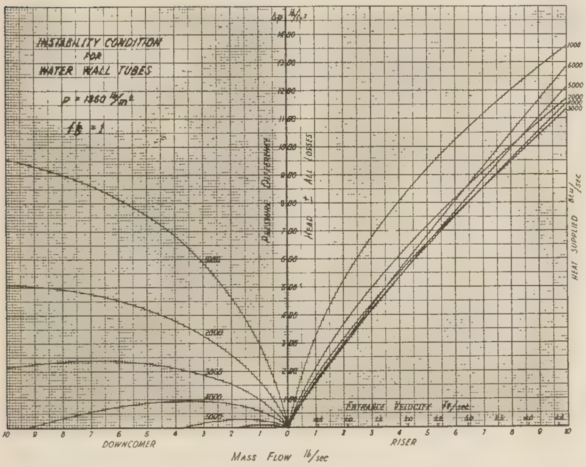


Fig. 10 Instability Condition for Waterwall Tubes $\left(p = 1350 \text{ psi};\ f\frac{L}{D} = 1.\right)$

This relation between pressure difference, flow rate G, and heat absorption Q is shown in Fig. 10 of this discussion for the following example:

The pressure is 1350 psi, tube length 50 ft, tube diameter 3 in., friction factor $f = 0.005$. The steadily rising curves on the right-hand side refer to riser tubes, while the curves on the left-hand side which all go through a maximum refer to downcomer tubes. To each heating rate belongs one curve, consisting of two branches. When the mass flow in a riser is large enough, such that a horizontal line traced through the pertaining pressure difference does not intersect with the corresponding branch for the downcomer, then the flow is stable. There will be one flow rate at each heating rate, where the horizontal line through the pressure difference just touches the curve branch for the downcomer. This flow rate then represents the limit between the stable and unstable region for the given heating rate. For lower mass-flow rates, the horizontal line will intersect twice with the corresponding branch for downward directed flow; each of the three flow conditions may then exist in the tube, and the flow is unstable. Therefore, waterwalls must be laid out for such flow rates as assure stability of flow at the desired heat-absorption rate.

Author's Closure

Mr. Bailey is right in stating that the assumptions made in this paper should be correct. It would be better to state that all assumptions should be as nearly correct as it is possible to make them under the known conditions. It should also be understood that such assumptions cannot and need not be exact.

The desirability of actual tests to confirm circulation calculations cannot be questioned. In the absence of such tests, the operating records of a large number of similar high-pressure boilers may serve as a substitute.

The criterion of adequate circulation is the complete protection of the tubes. Just what is necessary to obtain such complete protection is not positively known. Each case is a problem in itself. In general, the degree of protection is dependent upon the percentage of water by volume, the velocity of the mixture, the inside diameter of the tube, and the position of the tube, i.e., whether it is vertical or at some angle from the vertical.

The mean-density determinations must start with the total heat input to the circuit and also the distribution of this input along the tube. If heat inputs are as assumed in Figs. 1 and 4 of the paper, the densities will be as calculated for these conditions, unless the steam moves faster than the water, in which case the mean density will, of course, be higher than calculated. However, for tubes of $2^1/_2$ in. inside diam and less, it is believed that there is little difference between the steam and water velocities, except for a relatively short distance from the bottom end of the tube.

If the arithmetic-mean density is used in calculating head and the same top-dryness fractions are used, in the assumed case the front and side walls will be in equilibrium at top-dryness fractions of 0.17 and 0.18, corresponding to approximately 75 per cent steam by volume.

Designers recognize the fact that absorption in different parts of a circuit, or in different parallel circuits, will vary due to conditions of firing, burner location, slag accumulation, etc. Here, the designer must rely on judgment based on his experience.

The term "dryness fraction" was used by Lewis and Robertson in the paper[3] referred to in the author's paper. The term seems to be fully explanatory and certainly less clumsy than "per cent of steam by weight." The reciprocal of top dryness, that is, the weight of water entering the circuit, divided by the weight of steam generated is the circulation ratio, which the author believes is also fully explanatory. The term "dryness fraction" can be used for any amount of steam by weight at any part of the circuit for which this information is desired or known, while the top-dryness fraction is the percentage of steam by weight at the point where the heat input ceases.

If different circuits are nearly identical, they may be solved simultaneously, otherwise they are solved separately and afterward corrected for any deficiency. In the paper, for example, the side wall and front wall are in equilibrium at different top-dryness fractions. Although not necessary, these circuits can be brought into equilibrium with the same top-dryness fraction by decreasing the resistance in the side-wall circuit.

Losses may be decreased by increasing either riser or downcomer areas as Mr. Bailey suggests. In the assumed boiler, there are as many risers as can be conveniently used, therefore, the best method for reducing the side-wall losses is by decreasing the downcomer area.

There might readily be some confusion in following the conclusion that decreasing the height of the furnace progressively decreases the circulation. However, if the total heat input remains the same, the available head will decrease and the top-dryness fraction will increase. With the increase in dryness, resistance will also decrease, but not as rapidly as the loss in head.

Mr. Dornbrook has brought up a point which is well worth

discussion. It is the author's opinion that the safe "dryness fraction" at the top of a steam-generating tube depends upon a number of conditions, e.g., tube diameter, velocity, position of tube, whether horizontal or inclined, pressure, and heat input at the top end of the tube. It is believed that at any pressure a mixture of 75 to 80 per cent steam by volume will provide ample protection to tubes at the top end, provided the velocity of the mixture is sufficient to cause turbulent flow.

It is hoped that, as experience with large high-pressure units becomes more extensive, data on limiting velocities, dryness fraction, and transfer rates will become available.

With reference to Mr. Kuhner's comments on entrance and discharge losses in headers, these losses, expressed in head of fluid flowing, may be assumed as correct, provided the flow of fluid is across the header and not lengthwise. If the flow through the header is axial, naturally, headers should be of ample cross section to insure low axial velocity. In the paper, entrance losses were figured at twice the maximum theoretical figure of $^1/_2$ $V^2/2g$ to compensate for possible turbulence losses in headers.

If top-dryness fractions are figured for the maximum heat input, tubes having less than maximum heat absorption will have a lower top-dryness fraction. If, however, adjacent tubes absorb heat in greatly differing amounts, circulation may become unstable unless each tube connects directly with the drum without the interpositioning of an intermediate header.

The influence of heated downcomer tubes in a boiler bank is pointed out in the paper and, in such units as described, the lowered density due to the small amount of heat absorbed in the downcomer bank is not sufficient to impair the circulation. A downcomer bank, following a large furnace and a high-temperature superheater, absorbs but a small fraction of the total absorbed by the boiler and furnace.

As to the effect of change in pressure on the top-dryness fraction, it is obvious that, due to the increase in specific volume of steam, the calculated velocity of the mixture will increase at lower pressure for the same top dryness. The increased resistance will then increase the top-dryness fraction. Whether this decrease in percentage of water will necessitate a lowered rating will depend upon the liberality of design. For the relation between top-dryness fraction and relative volumes of steam and water refer to a previous paper[11] by the author.

As to Mr. Kuhner's suggestion that this method be applied to a study of the circulation of the boilers described in the paper[7] by Messrs. Hall and Partridge, it is the author's opinion that circuits such as shown in Fig. 3 of that paper cannot be analyzed by any known method.

In circuits such as that given in Fig. 14, showing the furnace at Rivesville, and Fig. 27, the Logan Station, also from that paper, analysis discloses entirely adequate circulation for the protection of the upper ends of the circuits. In such tubes, the velocity of the water entering is low, probably between $1^1/_2$ and $3^1/_2$ fps. At these velocities steam will segregate along the top of the tube and will remain so segregated until the velocity becomes high enough to cause turbulent flow. Such steam segregation or blanketing is the cause of the corrosion encountered in these installations.

Mr. Leib calls attention to the study of stability conditions of flow. He has improved the method developed in Germany by eliminating some simplifications and introducing the correct terms instead. It certainly is desirable to enable the designer to check flow conditions in a given tube system by means of an exact method. It must, however, be borne in mind that the method devised requires the knowledge of the amount of heat absorbed by the individual tubes in a parallel circuit. In general, this amount of heat can only be estimated. Therefore, the success of the method hinges entirely upon the accuracy of this estimation. As the knowledge of temperature distribution in furnaces progresses, this accuracy will undoubtedly improve, and, at the present state of our knowledge, a check of the stability of circulation in all cases where there is any doubt of stability is very desirable.

[11] "Design of High Capacity Boilers," by J. Van Brunt, Trans. A.S.M.E., vol. 60, 1938, Fig. 10, p. 488.

Recent Developments of the Pease-Anthony Gas Scrubber

By R. V. KLEINSCHMIDT[1] and A. W. ANTHONY, JR.,[2] CAMBRIDGE, MASS.

This paper brings up to date recent developments in an improved cyclonic-spray scrubber. Theory of design is discussed briefly, but more attention is devoted to actual installations and their performances, troubles, and possibilities. A table of test data is given covering units from 200 to 50,000 cfm. Scrubbing of boiler flue gases for fly-ash removal is covered with particular emphasis on possibilities for improved efficiencies and lower costs; possible benefits and economies through recovery and sale of SO_2 are indicated. Two installations for treatment of tar fog are described. Application to the cleaning and cooling of blast-furnace gas is reported in a preliminary way. A modification, retaining the principles of fine atomization but with the gas and liquid in countercurrent contact, is described, and compared favorably with bubble-cap columns and packed towers. Miscellaneous special applications are listed briefly.

IT IS four years since M. D. Engle (3)[3] reported to the A.S.M.E. on the first full plant installation of Pease-Anthony cyclonic-spray scrubbers. Since that time, development has proceeded actively so that, although these scrubbers are still in regular operation, they must, in the light of present standards of efficiency and economy, be regarded as obsolete. It would not be difficult to modernize this installation, for the physical changes involved are minor, involving only nozzles and baffling, as will appear later.

Theory of Scrubbing

The cyclonic-spray scrubber consists of a cylindrical chamber with a tangential gas inlet of suitable cross section near one end and a central gas exit at the other end, Fig. 1. A suitable spray of finely atomized particles of the scrubbing fluid is formed near the axis of the cylinder in the region directly above the inlet. Rotation of the gas in the chamber, due to the tangential entrance of the gas at a controlled velocity, causes the spray particles to travel outward through the gas to the walls of the chamber. The radial motion of the water particles across the gas stream causes them to collide with the dust particles and carry them to the walls from which they are washed down and discharged from the scrubber.

At the time these flue-gas scrubbers were designed, we had only hazy notions of the theoretical basis of design. The patent[4] indicated the necessity for using finely atomized spray, but it was believed that this was mainly because of the larger frontal area and better distribution of water particles. This view did not account for the rapid decrease in scrubbing efficiency, which is noted in the case of fine dusts, fumes, and smoke in all spray types of gas washers. Various reasons have been assigned for this phenomenon, among which may be mentioned inability to wet the dust particles, adsorbed gas layers, electric charges, and surface coating of the water droplets by the large number of fine dust particles which each water drop must remove. In most cases these appear to be decidedly minor effects, the major effect being simply derived by considering the aerodynamics of the collision of two particles surrounded by a gas. If the two particles are of comparable size the gas has little effect on the

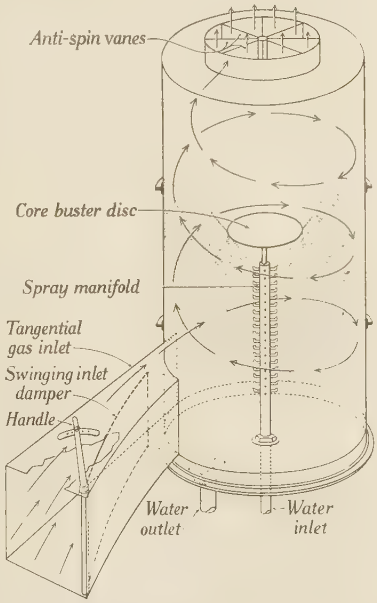


Fig. 1 Schematic View Showing Elements of Cyclonic-Spray Scrubber

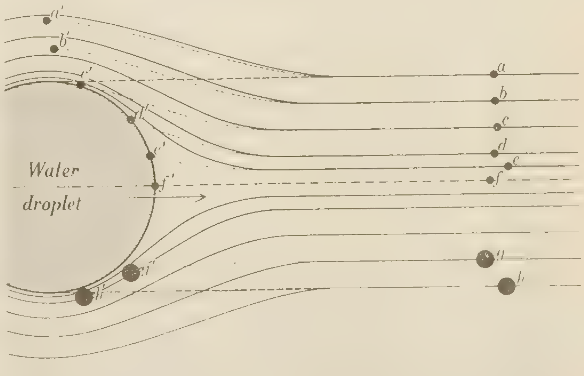


Fig. 2 Collision of Moving Water Droplet With Dust Particles of Various Sizes

[1] Physicist, Arthur D. Little, Inc., Consulting Engineer, Pease-Anthony Equipment Company. Mem. A.S.M.E.

[2] President, Pease-Anthony Equipment Company.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

[4] "Method of Washing Gases," by F. F. Pease, U. S. Patent No. 1,992,762, Feb. 26, 1935.

Contributed by the Process Industries Division, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

collision but, if the particle which is in motion relative to the gas is large, it will be surrounded by a streamline pattern in the gas which will carry small air-borne particles around it without collision, Fig. 2. This effect has been demonstrated by H. G. Houghton (1)[4] in his work on sampling fog droplets in the air. Houghton has found that the sampling efficiency of a flat plate held in an air stream falls off rapidly for fog particles of a diameter smaller than about 0.001 of the width of the exposed plate. In the case of water particles hitting gas-borne dust particles, the effective diameter ratio is of the order of 200 to 1. This permits a direct quantitative computation of the effectiveness of any type of spray on any dust, fog, or fume of known particle size.

As to the type of spray, we have obtained with Houghton's cooperation considerable information on the numbers and sizes of spray particles formed by certain types and sizes of nozzles at various pressures, Table 1 (table for one nozzle). As to the sizes and properties of dusts, a great deal has been written in a general way, but it is difficult to obtain adequate data for design purposes on most of the finer dusts and fumes, Tables 2 and 3. In fact, in the case of extremely fine fumes and fogs, it may be simpler to determine particle sizes from operation of a pilot plant than to measure them directly, especially in the case of liquids or tarry fogs which agglomerate readily. Once the characteristics of the dust are known, it is a simple matter to apply the collision theory to determine the diameter and height of scrubber chamber, the number and size of nozzles, pressure of water, and pressure drop of gas. Several of these factors may be selected to meet the particular conditions, the others being then determined by the design theory. The details of this design method have been given by Kleinschmidt (2), but a résumé will be given here.

TABLE 1 SIZE DISTRIBUTION OF WATER DROPLETS FROM CENTRIFUGAL NOZZLE, 0.188 IN. AT 65 PSI, 60 F[a]

1	2	3	4	5	6
Diameter of drop[1] microns	Number of drops measured	Number per cc. of spray[2]	% of volume	Area per cc. of spray sq. cm.	% of area
25	878	97,250	0.072	1.9	1.48
50	460	51,000	.334	4.0	3.12
100	190	21,000	1.11	6.6	5.15
150	89	9,850	1.75	6.9	5.38
200	53	5,870	2.46	7.4	5.78
250	33	3,650	3.00	7.2	5.61
300	22	2,440	3.46	6.9	5.38
350	16	1,770	3.97	6.8	5.30
400	13	1,440	4.83	7.2	5.61
450	11	1,220	5.84	7.8	6.09
500	10	1,107	7.25	8.7	6.79
550	8	886	7.74	8.4	6.55
600	7	776	8.79	8.8	6.87
650	6	664	9.59	8.8	6.87
700	5	554	9.68	8.5	6.64
750	4	443	9.84	7.8	6.08
800	3	332	8.94	6.7	5.22
850	2	221	7.13	5.0	3.90
900	1	110	4.22	2.8	2.18
950	0	0	0	0	0
1200	0	0	0	0	0
	1,811	200,583	100.00	128.2	100.00

[a] Courtesy of American Institute of Chemical Engineers.

The collision theory assumes that dust removal is due solely to mechanical collision of moving water droplets with dust particles which happen to be in their path. It is also assumed, as previously explained, that only water particles less than 200 times the diameter of the dust particles are effective. It is assumed that the effective length of path of the water particles through the gas is the radius of the scrubber. These assumptions lead to conservative estimates of scrubbing efficiency in most cases. Taking into account the fact that the probability of hitting a dust particle becomes less as the concentration of dust in the gases decreases, we obtain as an expression for the efficiency of scrubbing:

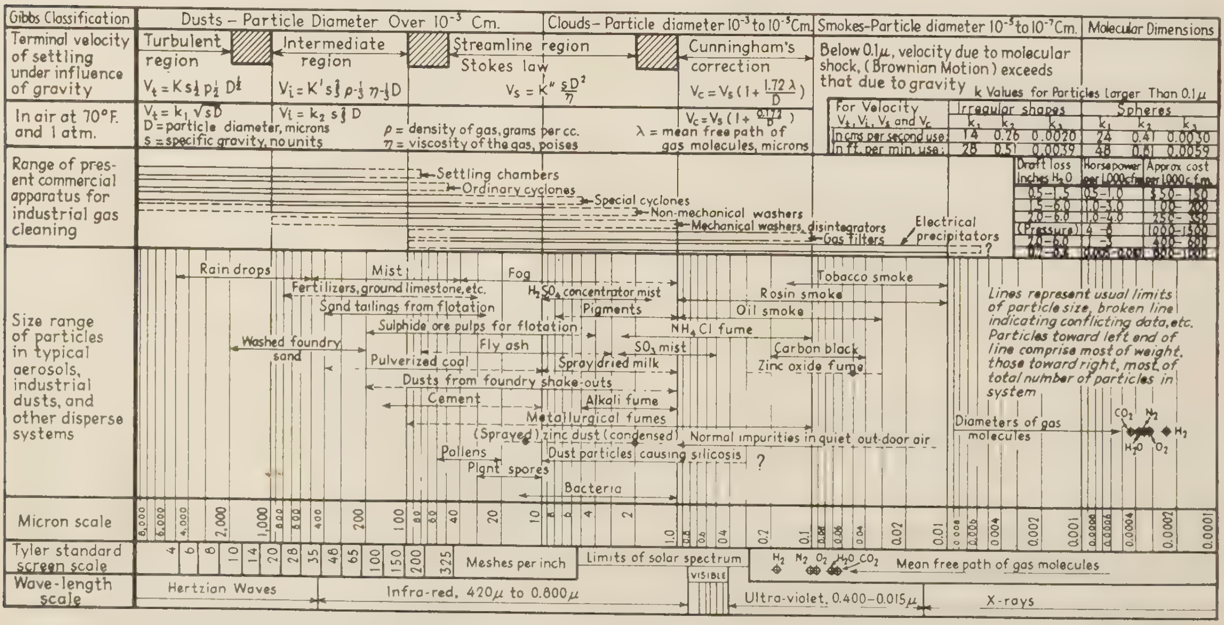
$$\text{Efficiency} = 1 - e^{(3DW/4dG)}$$

where D = diameter of scrubber, in.
d = diameter of water particles, in.
W = effective volume of water sprayed, cfm
G = volume of gas scrubbed, cfm

Values computed from this equation are given in Table 4.

Since the fineness of atomization which will be effective depends upon the size of the dust particles, it is necessary, in ap-

TABLE 2 SIZES AND PROPERTIES OF DUST PARTICLES AND FUMES[a]



[a] Courtesy of *Chemical and Metallurgical Engineering*.

plying this formula, to work out the efficiency of removal for each size range of the dust to be caught. The height of the scrubber and the inlet velocity required are computed from the centrifugal force necessary to drive the finest spray particles across the gas stream while the gas is within the scrubber.

It is interesting to note that entrainment of spray particles in the exit gases may occur at low gas loadings, since the centrifugal force decreases as the square of the gas inlet velocity while the axial velocity decreases only in proportion to the inlet velocity. Also, the gases rotate more or less in the manner of a solid cylinder, so that the centrifugal force at the axis is zero. Two recent improvements now incorporated in scrubber designs take care of these conditions. For low gas loadings, a swinging damper in the inlet permits control of that area with variation of gas flow, so as to maintain the gas inlet velocity at a suitable value. The height of the scrubber can be materially decreased by putting a circular plate baffle just above the spray manifold to act as a core buster to force all gas and spray away from the axis out into regions of higher centrifugal forces. See Fig. 1.

Pilot Plants

It has already been noted that pilot plants are frequently desirable when handling new types of dusts or unusual conditions. Several recent installations have been made for this purpose, and it is important, in considering the results obtained, to bear in mind their nature and significance.

The purpose of a pilot plant is primarily to obtain certain specific engineering information, which cannot be obtained in the laboratory, for use in design of a larger plant. It is not intended to demonstrate the efficiency either of a unit of its own size or of a larger unit. In fact, it is important that it should not be designed for high efficiency since, if overdesigned, it becomes more difficult to obtain useful data. For this reason efficiencies of 50 to 80 per cent are better than higher ones. Another fact to be remembered is that, for a given efficiency, the ratio of spray to gas scrubbed is, inversely, as the diameter of the scrubber, so that a pilot-plant scrubber of 3 ft diam would require 5 times as much water per 1000 cu ft of gas as would a 15-ft unit of the same efficiency. At the same time, since the gas must accelerate the spray droplets introduced, excessive amounts of water reduce the rotation of the gas unduly and necessitate high pressure drop in the pilot-plant unit. For these reasons and others of similar nature, it is not possible to demonstrate either the efficiency or the economy of a large installation in a pilot plant. It is possible, however, to obtain data from which efficiency and economy may be computed with considerable accuracy.

Engineering Problems

The scrubber shell proper presents no serious problems. It may be made of steel and lined with acidproof tile if corrosive conditions are severe, or it may be a self-supporting bonded-tile structure. Large-size glazed sewer pipe has been suggested for small scrubber shells, Fig. 3. In any case, corrosion and erosion must be carefully considered. Gas flows of from 1000 to 60,000 cfm (60 F basis) are easily handled in single shells, at axial velocities of 100 to 500 fpm.

Nozzles present the most serious problems in materials. For fine atomization, large numbers of relatively small nozzles are required, and high pressures are desirable. Available materials and cost limit the pressures used to approximately 200 psi even with clean water; and waters containing silt or recirculated dust limit the pressure to around 60 psi. Even at these pressures, mechanical strength, and corrosion and erosion resistance limit the available materials. Some of the so-called "lava" materials have been useful in resisting acid corrosion; they are cheap, abrasion-resistant and resistant to most acids, but must be protected from mechanical and hydraulic shock. Brass is good only for clean water at low pressures. Hard rubber has been used to resist dilute fluorine compounds at low pressure. A new type of nozzle now under development gives promise of eliminating most of the erosion troubles experienced with spin-chamber

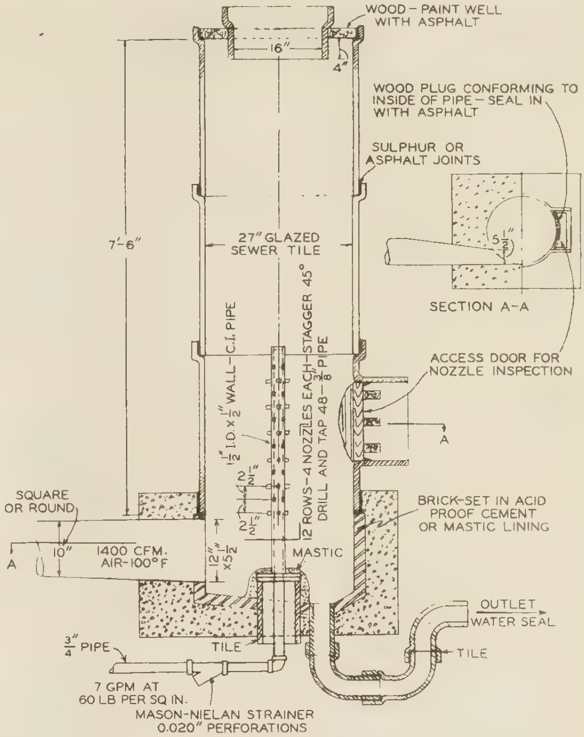


Fig. 3 Acid-Fume Scrubber

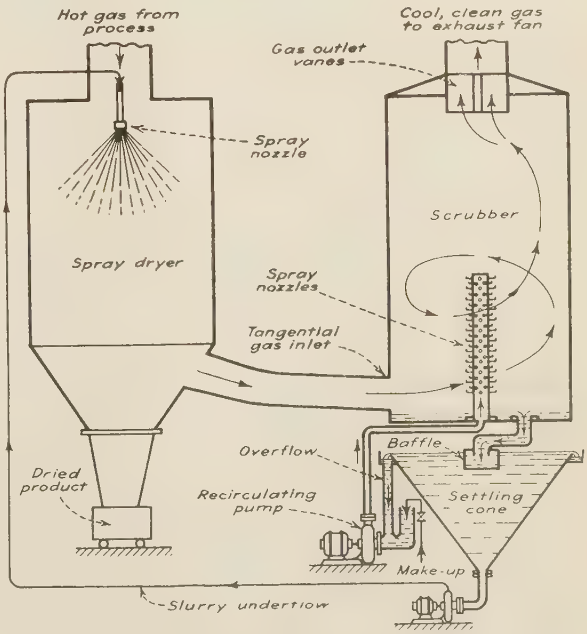


Fig. 4 Diagram of System for Recovery of a Dry Dusty Product

pressure nozzles. Every problem, however, requires a special study of the nozzle, and pilot-plant experience will often be valuable, until a vast background of experience has been accumulated.

Auxiliary Equipment

Handling Solids. Most scrubbing jobs require not only getting the solids into the liquid, but also getting them out of the liquid, either for recovery of the solids or for recirculation of the liquid. The solids may have value, or may be a costly nuisance. Local conditions usually dictate which of the numerous methods available should be selected. Fly ash from boilers has been settled out in Callow-type inverted cones, and the thickened underflow run to ash tanks for further settling. Tar from a certain tar fog formed a sticky mass which would not flow; as there was plenty of fresh water available, the effluent water drops its tar in an outdoor sump which is cleaned by a clam-shell-bucket crane several times a year, and the water then goes to waste. Soluble, valuable chemicals can be concentrated by recirculation in many cases, with a bleed-off going to processes already in operation. When hot gases containing chemicals are to be cleaned, the scrubber may be tied in with a spray drier, as shown in Fig. 4, the scrubber recovering dust from the drying operation while acting as a concentrator of the dilute solution.

Pumps. Centrifugal pumps with rubber-lined casings and rubber-coated impellers have given satisfactory service for 60 psi 115 F, weakly acid water carrying a small amount of fly ash in suspension. Numerous alloys were tried, without success. The cost and corrosion problems in high-pressure pumps limit the use of high nozzle pressures for corrosive conditions.

Piping. Rubber-lined pipe, with different degrees of hardness, has been useful. Tile pipe is frequently cheaper but must be supported. Lead pipe or lead-lined steel pipe and monel pipe have been used for certain severe conditions.

Although much of the development work on auxiliary equipment has been related to corrosion resistance, this problem is no more severe than with other types of wet scrubbers. The simplicity of the cyclonic-spray scrubber facilitates the attainment of resistance to corrosion and has therefore directed our attention to problems which no other equipment could handle.

Commercial Developments—Flue Gases

Flue-gas scrubbing is one of the most promising fields for this equipment, but widespread application has been held up pending full development of SO_2 recovery and sale, as a logical attack on the corrosion problem.

The only major installation of cyclonic scrubbers has been described by Engle (3). Two boilers were equipped in 1930, and an additional boiler was equipped in 1931. Engle covered the subject comprehensively, so only three points of universal interest will be considered here: efficiency, capital costs, and operating costs.

1 Efficiencies (on weight basis) of the installation were reported by Engle, and substantially confirmed by one of the authors. Little has been done since 1932, to improve efficiencies; in fact, owing to lack of fundamental knowledge, many things have been done tending in the opposite direction. In any event, by the use of more nozzles, better suited to the requirements, high efficiencies for these scrubbers can be maintained and yet keep within present costs of power for draft losses and for pumping.

Two scrubbers per boiler were installed in 1930, each rated at 72,500 cfm at 515 F. At the time of the Engle tests, these units had about 90 centrifugal-type nozzles of 0.1875 in. orifice, operating at about 55 psi, and showed full-load efficiencies of 80 to 82 per cent. Disregarding requirements for saturation of the gases, 180 efficient nozzles of 0.063-in. orifice, at the same pressure, would pass about 25 per cent of the amount of water, but the efficiency would rise to about 88 per cent. If 270 similar nozzles were used, with 37.5 per cent of the water, the efficiency would be about 93 per cent.

An improved type of nozzle will be ready for service in the near future, which will yield the same quantity of fine atomization with few of the coarse ineffective droplets, so that about 50 per cent of the gallonage of water need be pumped. Of course, to scrub with fine atomization, the gases must first be saturated but, assuming saturation of the gas, the following efficiencies can be realized commercially:

Nozzles: Size, in.	Nozzles: No.	Gpm, per cent	Removal efficiency, per cent
0.1875	90	100	80 to 82
0.063	180	25	87 to 88
0.063	270	38	92 to 93
0.063	New type	19	93 to 95
0.063	New type	38	96 to 98

Power requirements for pumping would decrease proportionately. While the foregoing figures may seem to be overoptimistic, similar methods in design indicated 98 per cent efficiencies on the scrubbing of blast-furnace gas, with realization of better than 99 per cent, as will be shown.

If entrainment in the scrubbed gases be eliminated by the addition of disks and swinging inlet dampers, less hot air need be added before the induced-draft fans, and a small increase in efficiency of the station can be realized. The swinging inlet dampers should be adjusted to maintain substantially constant inlet velocities at all gas loads and temperatures; draft loss will be the same at all loads, involving slight increases in fan power only at the lower loads.

2 Regarding the item of first cost, several years ago, when a prominent manufacturer was licensed to make and sell the scrubber, complete installations were quoted at prices which worked out at 25 cents to 35 cents per rated cfm (hot) for boilers of 100,000 lb per hr and larger. Corrosion was to be met by use of nonmetallic materials in contact with the gases and solutions.

If, however, corrosion can be eliminated so that plain, unprotected steel can be used, the cost of the scrubber, pump, piping, and nozzles can be cut at least 50 per cent. Apparently, the best chemical method is the use of an alkaline washing solution; however, the scrubber is such an excellent gas-liquid contactor that the alkaline solution will soon become acid, unless continuously replenished or regenerated to maintain the alkalinity. In England, lime or chalk slurries are continuously replenished, with a minimum of corrosion, but a considerable mass of fly ash and gypsum must be disposed of at considerable expense. H. F. Johnstone at the University of Illinois has studied extensively the regeneration of alkaline solutions and very recently he has reported important conclusions (4). Although he reports the absence of corrosion in his pilot plant, nevertheless he recommends some protection of steel surfaces, but not to the extent described by Engle. Acidproof units installed now for solids removal will, of course, be able to absorb SO_2 simultaneously, if and when the public convenience and necessity require.

3 Maintenance of the scrubber proper has been low, but in the case of the protective coatings of the induced-draft fan and uptakes, it has been high because no entirely satisfactory coating has yet been developed. On the other hand, erosion of induced-draft-fan rotors has been negligible; life expectancies of rotors under normal acid conditions are now certainly 10 years, and possibly equal to those of forced-draft-fan rotors. Future costs should be lower, because of advances in the art of synthetic resinous coatings, which are now available having substantially zero

moisture absorption; these are needed for the interiors of the induced-draft-fan casings, the uptakes and breechings. Heretofore, all the coatings tried have absorbed moisture and transmitted it to the metal beneath, with resultant corrosion, so that the coatings have required annual replacement. Such coatings probably will not be needed when alkaline scrubbing solutions are used; the indications are that plain steel can then be used without protection, or with a coating which normally should last for several years.

Entrainment at low gas loadings has increased the corrosion troubles in the past. Entrainment is unnecessary, and can be completely eliminated, as previously pointed out.

Miscellaneous—Flue Gases

Disposal of Fly Ash. Building blocks utilizing considerable fly ash are reported from Detroit (5). The cement industry reports beneficial results from the admixture of certain percentages of fly ash (6). The mixed-fertilizer industry uses it as a diluent in some cases (5). Its disposal after removal from the flue gas is still a problem, although the firing of pulverized coal in wet-bottom furnaces apparently retains more of the ash, and proportionately less goes up the stacks.

Mist Plume. Since the gases at the top of the stack contain considerable water vapor, during cold weather this becomes visible as a white plume or cloud, having a good deal of the appearance of steam although not quite as dense. The action of this plume has been studied under all weather conditions, and in general it behaves as would any other body of heated gas. It rises when all stack gases rise, and it strings out horizontally or even downward when wind conditions compel. In cool weather, as the gas emerges from a large stack, only the water vapor in the outer shell of gas, in immediate contact with the cool atmosphere, condenses, but turbulence and diffusion soon cause further condensation in the interior of the plume. A little later, evaporation of the fog droplets on a large scale sets in, and soon the white plume is gone. On days of high humidity, as would be expected, the plume is more persistent, and it has been observed stringing out as far as 1/2 mile.

Acid Rain. Fears have been expressed that an acid rain would fall on those beneath. We have been at some pains to find evidences of acid attack chargeable to the scrubber, but have found none. Qualified chemical engineers, who have studied this question of acid effects from the effluent scrubbed gases, have concluded that the over-all results are beneficial as to SO_3, which is largely scrubbed out by the wash water and ultimately run to waste, and with but slight effect as to SO_2, some of which probably is absorbed by the fog droplets as formed at the top of the stack, but is soon liberated when the fog evaporates.

Sulphur Dioxide Recovery

Some of the benefits of SO_2 recovery as a by-product have already been suggested. When flue gases are thoroughly scrubbed, the SO_2 present is a problem from either the standpoint of corrosion or of disposal of the recoverable by-product. However, if the gases are dry-cleaned, there is no problem at least to the boiler-plant operator, since the SO_2 is turned loose to the prevailing winds. Papers have been published to prove that SO_2 harms neither plants nor human beings, but nothing has yet appeared showing beneficial effects on life, or on steel, concrete, paint, etc. In short, SO_2 is on the defensive, and is generally considered as a nuisance which must be tolerated for the present, at least, in this country.

The ideal solution of the combustion problem would be to attain stack discharges containing neither solids nor acid gases, but consisting only of nitrogen and oxygen, with CO_2 and water vapor. Probably this ideal is not possible of achievement, but 90 to 95 per cent elimination is not only possible, but may be accomplished at no great increase in over-all cost, if and when due credit is allowed for the advantages, which include:

(*a*) Credit for sale of SO_2.

(*b*) Virtually complete elimination of need for maintenance of protection against corrosion.

(*c*) Minimum boiler outage chargeable to maintenance of scrubbers.

(*d*) Smaller induced-draft fans and less power to drive them, because gases are handled at lower temperatures.

(*e*) No erosion of induced-draft-fan wheels.

(*f*) Improved public relations, if the public is advised of the new methods being installed.

There are of course disadvantages:

(*a*) The price for SO_2 is not fixed and definite, and it will fluctuate with supply and demand, but it probably will not depart far from the price of the contained sulphur. The quantities available may be enormous, relative to present supplies. New outlets must be developed, and fortunately at least one is in sight, i.e., the new plastics containing nearly 50 per cent sulphur dioxide (4).

(*b*) Power-plant operators must know more chemistry.

(*c*) Duplication where solids eliminators are already installed.

(*d*) Fixed charges on increased plant.

If public interest is justified in requiring substantial elimination of cinders and fly ash from power-plant stacks, there is even more justification for requiring the elimination of SO_2, because probably the complete job can be done ultimately at small increase in annual cost of effective solids removal. If, however, SO_2 recovery is expected to show a profit after meeting annual costs not only on its own operations but also on those for solids removal as well, then the outlook for general acceptance is not favorable.

Tar-Fog Scrubbing

Fig. 5 shows a commercial cyclonic scrubber 8 ft diam × 20 ft high operating to remove tar fog from 28,000 cfm of the vent gases from ovens in which a pitch binder is decomposed. The resulting tar forms a viscous mass which adheres to the surfaces of ducts and fans, and which had caused serious trouble by clogging all other types of cleaning equipment, both wet and dry. This scrubber has been functioning satisfactorily for more than a year. Tar-fog droplets, as collected on a glass slide, are all below 10 mu and 89 per cent below 2 mu. Tests on this scrubber confirm the design theory as to effect of increasing water pressure. At 40 psi water pressure, the efficiency was 65 per cent and this was increased to 80 per cent by raising the pressure to 110 psi. This performance is entirely satisfactory to the users in that maintenance on the fan has been practically eliminated whereas, previously, the fan could not be kept in balance for more than a few days at a time. The water from this installation is not recirculated but is run to waste.

An important but secondary function of this scrubber is its action as a flame stop. The gases laden with tar fog are normally at moderately high temperatures, but occasionally fire breaks out in the tar deposited in the ducts, with possible damage to the induced-draft fan. The scrubber functioned as expected through one such fire, although all paint was blistered from the inlet duct.

The scrubber shell of this installation is of unprotected steel and its life has been rather short; it has been patched up and is still in operation but is scheduled to be replaced in the near future. Lava nozzles of smaller size are now being used but reports on efficiency with these nozzles have not been received.

Fig. 6 shows a pilot-plant scrubber installed to treat a tar fog

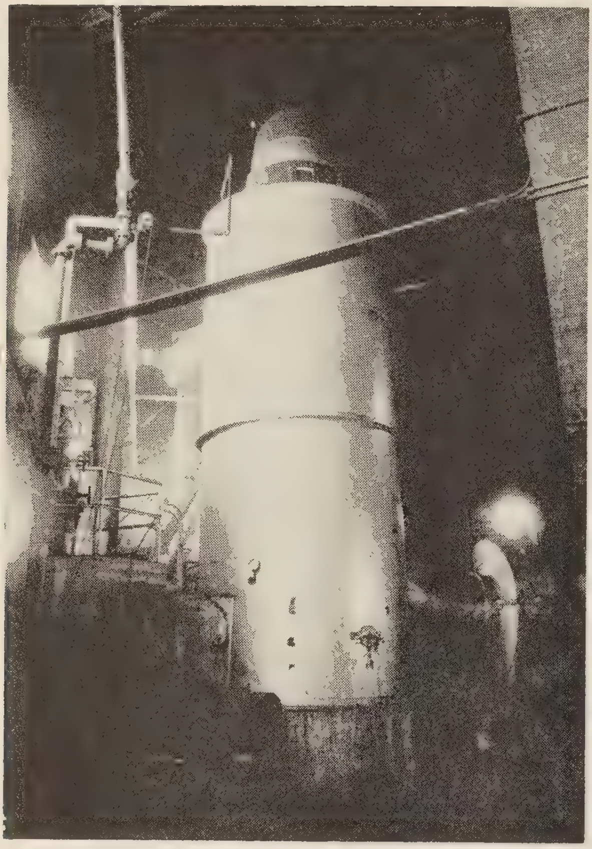


Fig. 5 Scrubber Handling 28,000 Cfm of Gas Containing Tar Fog

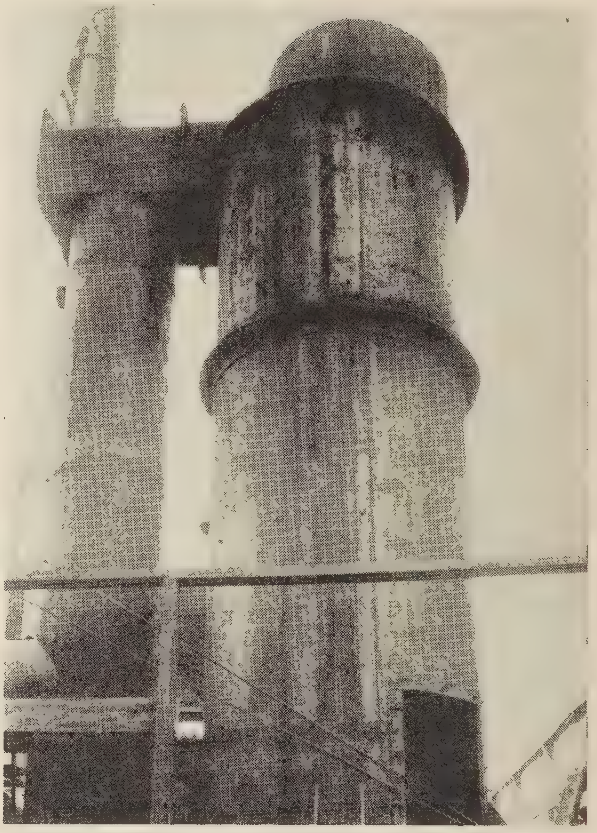


Fig. 6 General View of Tar-Fog Scrubber With Pressure-Regain Type of Outlet

from a sintering operation, the individual tar droplets ranging from 2 mu down to below 0.25 mu, the limit of resolvability by an ordinary microscope. From theoretical considerations, there is strong evidence that size of these tar droplets goes down to 0.1 mu and below. Efficiencies reported by this unit are lower than anticipated, chiefly because of the small size of the individual tar droplets. The most intensive scrubbing which has been provided, namely, 12 to 15 gal per 1000 cu ft through 335 nozzles (Fig. 7) of 0.037 in. diam at pressures as high as 320 psi made little effect on the appearance of the effluent cloud of tar fog, even though 50 to 75 per cent by weight of the material had been removed. Calculations in the light of the collision theory indicate that the individual tar droplets are present in overwhelming numbers. Of the 15 trillion tar-fog droplets in each cubic foot of dirty gas, about 3000 droplets were removed by each of the three billion water droplets formed by the nozzles per cubic foot of the gas. Doubling of this already intensive scrubbing should raise the efficiency to 85 or 90 per cent, which might begin to be perceptible to the naked eye.

Blast-Furnace Gas

The conditioning of blast-furnace gas has been considered one of the most difficult of gas-cleaning operations. Outlining the problem, approximately 25 per cent of the heat value of the coke charged into the top of a blast furnace comes out with the gases from the top as CO, in about 25 per cent concentration, or 90 to 105 Btu per cu ft. The heat quantities are enormous, and the gas is used in stoves for preheating the blast air, in large gas engines for blowing and power generation, for miscellaneous heating purposes; any residuum is burned under boilers for steam production. The requirements for cleanliness vary widely. Frequently raw gas is burned under boilers; however, Harmon (7) has recently pointed out the savings to be made by cleaning to 0.05 grain per cu ft. Gas for stoves should be cleaned to 0.02 grain per cu ft, and for engines to 0.015 grain per cu ft and better (8). Dehumidification also is necessary to obtain higher combustion temperatures and to minimize condensation and freezing in the gas mains in the winter.

To provide this cleaning and cooling, most blast furnaces are now equipped with three mechanisms handling the dirty gas in series, as follows:

1 Dry dust catchers, i.e., large dry cyclones, removing the brickbats and coarse dust.

2 Primary washers, of which there are numerous types, most of them operating to clean the gas down to 0.25 grain per cu ft. Large quantities of water are used in order to cool and dehumidify the gas while cleaning, 20 to 30 gal or more per 1000 cu ft of gas, to absorb the heat content of the dirty gas.

3 Secondary cleaners, which include rotary disintegrators and electrostatic precipitators. Preliminary data indicate that the cyclonic scrubber will perform simultaneously the duties of mechanisms Nos. 2 and 3 in one unit, with low power costs both for draft loss and for pumping, as well as much lower capital costs.

The operator of a number of blast furnaces, learning of the performance of this type scrubber for fly-ash removal from boiler flue gases with small quantities of water, recognized its possibilities for his purposes where much larger water quantities were

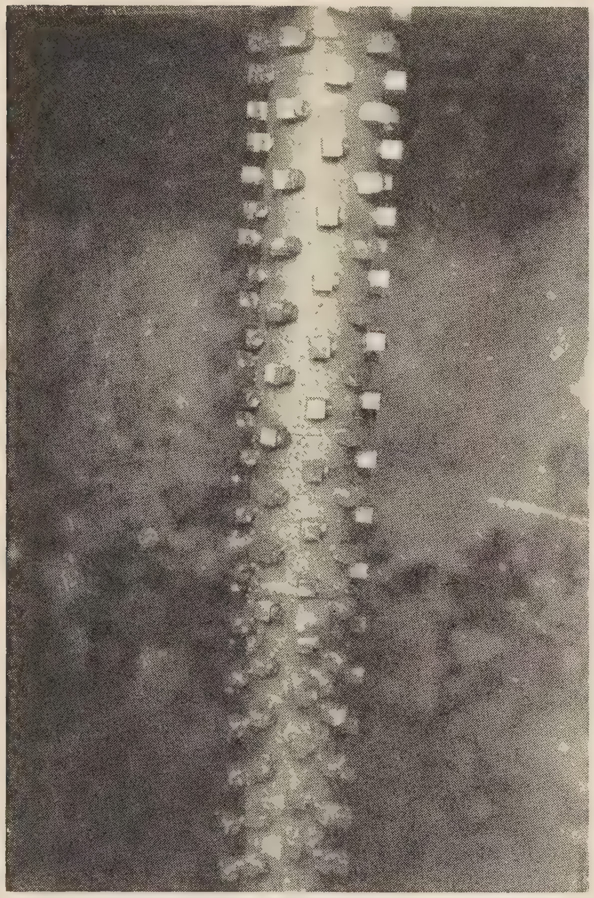


Fig. 7 Close-Up of Spray Manifold Showing Some of the 335 Nozzles With Orifice 0.037 In.

required. A full-scale preliminary tryout was arranged, involving the conversion of an existing primary washer (one of three in parallel) to the cyclonic type by changing the gas inlet from radial to tangential, and by installing an axial spray manifold of more than 500 brass nozzles. The gas piping, designed to handle one third of the gas from the furnace, was not changed as it was expected that this piping could handle all the gas during the short periods of test. Unfortunately, the draft loss through the overloaded gas piping was so high that it was not possible to obtain the complete test data originally planned for. Two test points were obtained and these showed efficiencies (weight basis) of 99.5 per cent. During normal operation, when dirty gas loadings run 2.5 to 3 grains per cu ft, the clean gas contained 0.012 grain per cu ft. During a period of severe slipping in the furnace, when dust loadings run 15 to 25 grains per cu ft, the clean-gas test showed 0.068 grain per cu ft. In addition, the operators considered the gas to be exceedingly dry, and were pleased with the degree of cooling obtained, in spite of the fact that only 17 gal of water per 1000 cu ft were used. The pressure for production of pig iron has rendered impossible the making of desired changes in the gas piping. This development has also been delayed with the problem of nozzle materials capable of withstanding erosion from the gritty water; the new nozzle referred to previously is on test, and gives promise of being satisfactory.

Chemical Applications

For solids, recovery primarily the scrubber has interesting possibilities in connection with the recovery of soda fume, along with considerable quantities of SO_2 and SO_3, from the discharge of the furnaces in which is burned the black liquor from paper mills cooking kraft, or sulphate pulp. The soda represents a recoverable value of considerable proportions. The particles, however, are very fine and not readily recovered. Pilot-plant tests indicate that the fume is almost entirely below 2 mu with a majority of the material in the neighborhood of 0.2 mu. These tests also indicate that the material can be caught in a carefully designed scrubber. The solution should be recirculated to provide a high concentration of soda in the effluent liquor, so that it can be economically treated for the recovery of the soda. The

TABLE 3 CHARACTERISTICS OF FINE DUSTS

(Dusts assumed to be spheres of density = 2)

Diameter, mu	Settling rate, fpm	No. particles per grain (millions)
20	4.8	15.6
10	1.2	125
5	0.30	1,000
2	0.048	15,600
1	0.012	125,000
0.5	0.003	1,000,000
0.2	0.0004	15,600,000
0.1	0.00007	125,000,000

TABLE 4 EFFECT OF QUANTITY OF SPRAY ON DUST REMOVAL

Number of times gas volume is swept by spray	Efficiency of dust removal per cent
1.0	63.2
1.5	77.7
2.0	86.5
3.0	95.0
4.0	98.2
5.0	99.34
6.0	99.76
7.0	99.91
8.0	99.97

TABLE 5 GAS-SCRUBBER TESTS

Rated Capacity c.f.m. Saturated	Gas and Dust	Conditions		Dimensions		Draft Loss in H_2O	Eff. %	Spray					Corrosive Conditions	Construction Materials			
		gr./c.f.	Size	Dia. ft.	Height ft.			Gal. per Mcf.	Nozzle No.	Orifice Dia.	Press. psi.	Recirculate		Shell Material	Shell Life	Nozzles Material	Nozzles Life
200	Chemical fume	17.0	0.5-3.5	1.0	6.0	9.	94	10	24	.037	60-360	Yes	Alkaline	Steel	?	Brass	Hours
"	Tar fog	2.-3.	0.1-2.0	"	"	"	91	10-30	"	"	75-360	No	SO_2, HF	"	?	"	"
750	Laboratory unit	Miscellaneous		1.5	5.0	1.5-2.3	-	3-12	max. 36	.046	40-600	No	-	"	Indef.	"	-
2,000	Soap dust	Var.	large	3.0	6.0	1.0	99+	1-5	rotary		-	Yes	No	"	"	"	Indef.
2,000	Chemical fume	2.25	0.2-2.0	3.0	30.0	4.0	65	6	48	.046	400	No	SO_2, etc.	"	"	Monel	?
5,000	Tar fog	2.0-3.0	0.1-2.0	4.0	12.0	5.0-12.0	50-75	5-15	335	.037	100-320	Yes/No	SO_2, HF	Lead lined	"	Monel/Brass	Days
15,000	Soap dust	Var.	large	7.0	12.0	2.5	99+	0.25-1.5	rotary		-	Yes	No	Steel	"	Brass	Indef.
25,000	Tar fog	.018-.027	0.1-2.0	8.0	20.0	3.0	65-80	2.8-4.6	164	.096	45-118	No	SO_2, HF	"	8 mo.	"	6 mo.
45,000	Boiler fly ash (one unit)	.25-2.5	2.0-5.0	10.3	20.0	0.8-1.5	82-95	2-3	80	.188	50-55	Yes	SO_2, etc.	Acid brick	10 yrs.	Lava	2 yrs.
50,000	Blast furnace	2.50	-	12.0	60.0	9.0	99+	17	534	.140	75-80	No	No	Steel	Indef.	Lava	Indef.

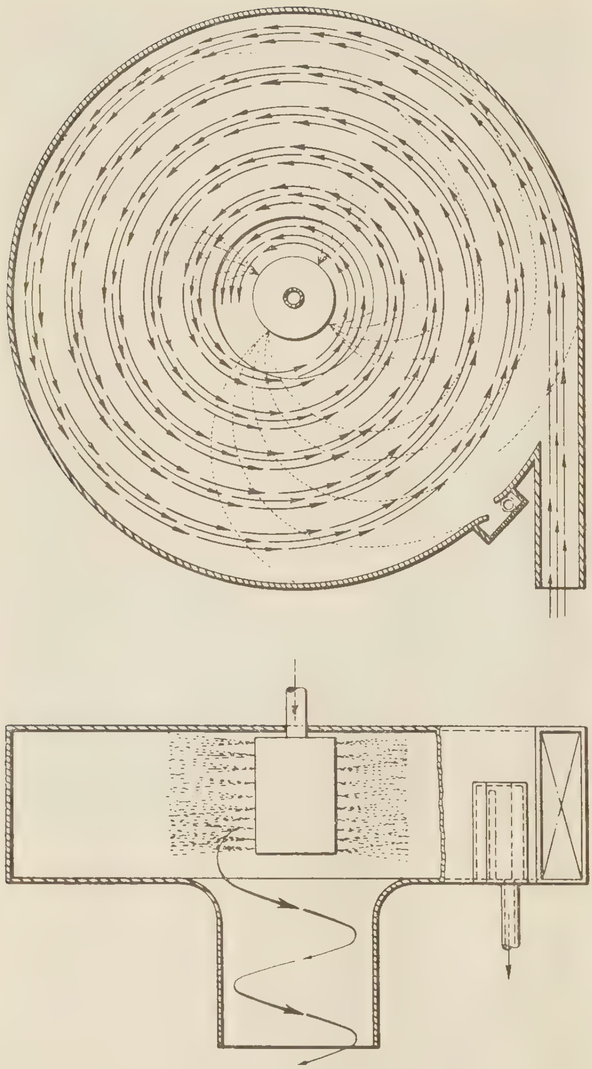


Fig. 8 Variation of Cyclonic-Spray-Scrubber Principle, Giving Countercurrent Contact Between Gas and Liquid

solution is strongly acid due to SO_2 and SO_3 in the gases, and the scrubber materials should be similar to those used in flue-gas scrubbing. Such an application offers an attractive economic picture as well as reduction of atmospheric pollution, both by solids and by "paper-mill odor."

The cross-current scrubber is able to provide such intimate gas-liquid contact that it is being considered for many chemical applications, among them gas absorption, removal of dilute acid gases from industrial process gases, elimination of odor nuisances from sewage and garbage-treatment plants, recovery of solvents, recovery of natural gasoline, vacuum fractionation of lubricating oils and of synthetic liquids of high molecular weights. It may be used as an evaporator or concentrator, as with a spray drier, Fig. 4.

Data on the efficiency of this scrubber for the absorption of soluble gases, i.e., SO_2 from flue gases, obtained in a large-scale installation, have been reported by Johnstone and Kleinschmidt (9). In this case 96 to 99 per cent of the SO_2 in the gases was absorbed in the scrubber under conditions which gave approximately 85 to 90 per cent dust removal. Thus the absorption efficiency is much higher than the efficiency of dust removal. This accords with theoretical predictions, since diffusion of the gas molecules increases the effective range of action of liquid particles.

Countercurrent Arrangement

Fig. 8 shows an important variation of the cyclonic-spray-scrubber principle, which yields true countercurrent contact between gas and liquid, with several obvious advantages. The scrubber shell is disk-shaped, large in diameter in relation to its height. The gas enters the full height of the periphery by means of a narrow slot, and pursues an inward spiral path, leaving axially at the center. The spray droplets are introduced substantially along the axis, and are taken up by the spinning gas body and pursue spiral paths across to the periphery. This arrangement has been tried out in a laboratory size 12 in. diam $\times$ 3 in. high, which passed 60 to 95 cfm and yielded contact equivalent to from 2 to 4 effective plates. This type of unit would be effective in absorption of ammonia in water, of H_2S in weak alkaline solution from which it is to be recovered by heating, of benzol vapors in wash oil, and numerous similar applications.

It is expected that, as more experience is gained, it will be possible for many applications to exceed with these units the performance of bubble-cap columns and packed towers with much lower space, weight, and costs.

Conclusions

The cyclonic-spray scrubber described has several advantages of importance; it is simple and versatile, has low draft loss, and has nothing to clog except spray nozzles. Efficiency is readily adjustable by change of pressure on the atomizing nozzles, and it can be very high for solids down to about 0.5 mu in size. Entrainment can be reduced to substantially zero, of importance in many chemical processes. This scrubber is especially effective for the economical cleaning of large volumes of gases, such as blast-furnace and boiler-flue gases. It can act as an absorber or gas-liquid contactor with simultaneous removal of dusts, as for instance SO_2 and fly ash from boiler flue gases.

Acknowledgments

So many different individuals and organizations have had a part in the development of this scrubber that a long list would be required to name them. Since, however, in many cases, identifications have not been possible, specific acknowledgments are omitted.

BIBLIOGRAPHY

1 "Measurement of Drop Size and Liquid Water Content in Fog and Clouds," by H. G. Houghton and W. H. Radford, Papers in Physical Oceanography and Meteorology, The Massachusetts Institute of Technology, vol. 6, no. 4, November, 1938, 31 pages.

2 "Factors in Spray Scrubber Design," by R. V. Kleinschmidt, *Chemical and Metallurgical Engineering*, vol. 46, August, 1939, pp. 487–490.

3 "Pease-Anthony Gas Scrubbers," by M. D. Engle, Trans. A.S.M.E., vol. 59, 1937, pp. 358–360.

4 "Recovery of Sulfur Dioxide From Waste Gases," by H. F. Johnstone and A. D. Singh, *Industrial and Engineering Chemistry*, vol. 32, August, 1940, pp. 1037–1049.

5 "Possibilities for Utilization of Pulverized-Coal Ash," by A. W. Thorson and J. S. Nelles, *Mechanical Engineering*, vol. 60, 1938, pp. 845–851.

6 "Find Favorable Results From Mixtures of Fly Ash With Cement," by R. E. Davis, *Concrete*, vol. 46, October, 1938, p. 26.

7 "Cleaning Blast-Furnace Gas for Boiler Plant Use," by R. R. Harmon, *Blast Furnace and Steel Plant*, July, 1940, pp. 648–685, 698–699, and 710–711.

8 "Blast-Furnace Practice," by R. H. Sweetser, McGraw-Hill Book Company, Inc., New York, N. Y., 1938, pp. 53 and 63.

9 "The Absorption of Gases in Wet Cyclone Scrubbers," by H. F. Johnstone and R. V. Kleinschmidt, Trans. American Institute of Chemical Engineers, vol. 34, 1938, pp. 181–198.

Discussion

H. F. Johnstone.[5] The authors' reference to the stack-gas investigation being conducted at the University of Illinois warrants discussion of two points mentioned in the paper (*a*) the economics of sulphur-dioxide removal and, (*b*) the application of cyclone-spray scrubbers as gas absorbers.

On several occasions, the writer has published statements regarding the cost of removing sulphur dioxide from dilute waste gases. The most recent of these was the result of a detailed study of what appears, at least from the chemical and mechanical standpoint, to be the best method of accomplishing this purpose.[6] Viewed from any position, it is recognized that the application of any process of sulphur-dioxide removal to large quantities of stack gases requires a large and costly installation. The item of greatest uncertainty and perhaps of greatest cost is the disposal of the recovered material, whether it be as a waste product or as a chemical raw material to which some value can be attached. In any case, much more information is required before it can be stated that simultaneous dust removal and sulphur-dioxide recovery is feasible and that definite savings can be made in the cost of the scrubber installation by removing the corrosive conditions encountered in the circulation of the acid solution.

The paper refers also to the tests made by Dr. Kleinschmidt and the writer on the absorption efficiency of a large dust-recovery unit in which an alkaline solution was circulated for the purpose of the tests. The high efficiency obtained in this case, compared with the known low absorption efficiencies of simple spray scrubbers, can be explained easily on theoretical grounds. For certain purposes, therefore, especially when saturation of the solvent is not desired, the wet cyclone would seem to fulfill the needs of many chemical-engineering absorption problems. Here again, it is unfortunte that more information is not available from other installations. It is particularly desirable to know the effect of the dimensions of the scrubber, of the location and size of the entrance duct, and of the number and type of nozzles on the absorption efficiency. Knowledge of the nature of the flow in the vortex and especially the tendency for the droplets to coalesce would be valuable. While the authors did not mention the latter point, it is obvious that it must be of great importance in dust removal also. Simple calculations will show that, unless the spray from the nozzles is quite uniform, the probability of coalescence of the small drops with larger drops is extremely great at radii above 3 or 4 ft. Consequently, the statement in regard to the relative water requirements of scrubbers of different sizes appears to be subject to some limitation.

[5] Professor of Chemical Engineering, University of Illinois, Urbana, Ill.

[6] "Recovery of Sulfur Dioxide From Waste Gases," by H. F. Johnstone and A. D. Singh, University of Illinois, Engineering Experiment Station, Bulletin No. 324, 1940. Abstract, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1037–1049.

Authors' Closure

Professor Johnstone has brought out several points which are of interest in connection with gas scrubbing but which the authors felt unable to treat adequately in the allotted time. As to the commercial recovery of sulphur dioxide, it seems that Dr. Johnstone's position is one which arises from his academic point of view. It is unquestionably true that we do not at the present time know all that we would like to know about the economic and industrial problems confronting the recovery of sulphur dioxide. At the same time it is also probable that such complete knowledge never can be and never has been acquired with respect to any commercial process. It is felt that Dr. Johnstone's publications on the subject indicate an adequate basis for a careful commercial study of a large installation which, like all first installations of radically new processes, must be regarded as experimental. The cost figures, which in Dr. Johnstone's opinion do not appear too favorable, include such unknown factors as the percentage of solution lost in carry-over in the gases and other similar losses from the cycle. Such items can only be definitely determined in large-scale operation, and if they become important, as they appear in Dr. Johnstone's figures, they can usually be reduced by proper design or operating procedure. From the authors' own experience, it is concluded that the loss of alkaline solution in the scrubbing step would be practically negligible.

As to the factors affecting the efficiency of absorption, their experience has been that it is a very simple matter to obtain such high percentages of absorption with the present type of scrubber that it is not necessary to go to any elaborate determination of the minimum required equipment except in special cases which might arise. Coalescence of the small drops into larger drops undoubtedly occurs to a certain extent since a spray which scrubs out small dust particles should also scrub out water droplets. Practically, coalescence appears to have little effect on the performance of scrubbers as is indicated by the fact that the actual performance follows rather closely the computed efficiencies.

In view of the many uncertainties involved in applying both the theory and the experimental results previously obtained on other installations to new and different problems, it is believed that further development of the scrubber will be along the lines of engineering development and experience in numerous applications, rather than in any extensive laboratory study of the well-known factors involved.

Relationship of Viscosity to Rate of Shear

By L. J. Bradford[1] and F. J. Villforth, Jr.,[2] State College, Pa.

This paper reports tests designed to check the validity of the assumption that lubricating oils belong to the class of a Newtonian fluid, which is defined as one in which the force required to shear it is directly proportional to the rate of shear. It is on this assumption, which in recent years has been questioned, that equations for the behavior of bearings have been based. Experimental evidence is produced in support of Petroff's equation which states that the torque required to rotate a journal, concentric with its bearing, is directly proportional to the product of the absolute viscosity and the rate of shear of the fluid separating the journal and the bearing. The agreement of the results with those predicted by the Petroff equation upon the assumption of the independence of viscosity from the rate of shear holds for all of the oils investigated.

A NEWTONIAN fluid is defined as one in which the force required to shear it is directly proportional to the rate of shear. Lubricating oils are generally supposed to belong to this class of fluid, and the equations applying to the behavior of bearings are based on this assumption.

Within the last few years the validity of this assumption has been questioned, and certain experimental evidence has been produced to show that the resistance to shear varies with the rate of shear. In other words, the viscosity of an oil is a function of the rate of shear as well as of temperature and pressure. The claim has been advanced that shearing of the laminas, which may be considered as forming the film separating two moving plates, causes the molecules making up the film to orient themselves. This orientation is said to cause a reduction in the resistance to motion of each lamina with respect to its neighbor. This is another way of saying that molecular orientation caused by flow results in a decrease in the viscosity of the fluid. The claim is also made that the higher the rate of shear, the greater the resulting orientation and, consequently, the greater the decrease in the viscosity of the fluid sheared.

Since the entire treatment of bearings operating in the fluid-film region has been built up on the supposed independence of viscosity from the rate of shear, a modification of the hydrodynamic theory of lubrication would be necessary if such independence really does not exist.

Apparatus Used

Two pieces of apparatus were used in this work. The first consisted of a steel ring suspended within a rotating steel bowl by means of a flexible steel rod. A fixed clearance was maintained between the ring and the bowl. Oil was introduced into the bowl and thrown by centrifugal force into the clearance space through which it passed. The assembly of this piece of apparatus is shown in Fig. 1. Details of the bowl and ring together with the more important dimensions are shown in Figs. 2 and 3. It will be seen that the mean diameter of the bowl was 5.2495 in. and that of the ring was 5.2429 in., giving a diametral clearance of 0.0066 in. when the ring and the bowl were at the same temperature. This clearance changed if the ring and bowl had different temperatures. It was also affected by the expansion of the bowl due to centrifugal force, and corrections for these changes had to be made to suit the conditions obtaining at the time of the observations.

The bowl was a loose fit on the driving spindle, which permitted it to center itself at running speeds.

The rod suspending the ring was 0.125 in. in diam and 19.5 in. long. It possessed but slight lateral stiffness and was therefore able to center itself with respect to the bowl when the latter was running. The attachment at the upper end was constructed so as to permit the ring to be approximately centered in the bowl while the latter was at rest. Thus, only a small amount of bending of the suspension rod was required to obtain the necessary degree of centering.

The temperature of the ring was determined by means of thermocouples placed in its wall, as shown in Fig. 2.

The temperature of the bowl was more difficult to determine, and two methods were tried. In the first, a hole was drilled in the edge of the bowl, as shown at A in Fig. 3. This hole was filled with oil and plugged. The apparatus was operated at a desired speed until conditions became constant and then stopped. The plug was removed and a thermocouple junction was placed in the oil. Several readings were taken, and the times at which the observations were made were noted. A plot was then made of temperature against time, with the time the bowl stopped taken as zero. The temperature of the bowl at the instant it

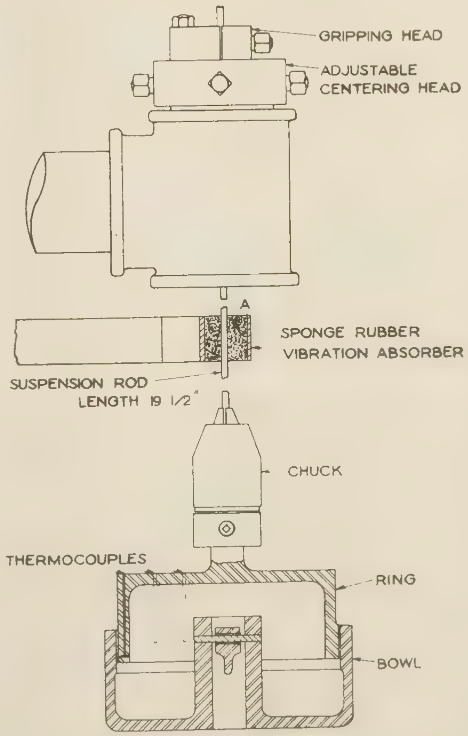


Fig. 1 Assembly View of Test Apparatus

[1] Professor of Machine Design, The Pennsylvania State College.

[2] Research Assistant, The Pennsylvania State College.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, N. Y., Dec. 2–6, 1940, of The American Society of Mechanical Engineers.

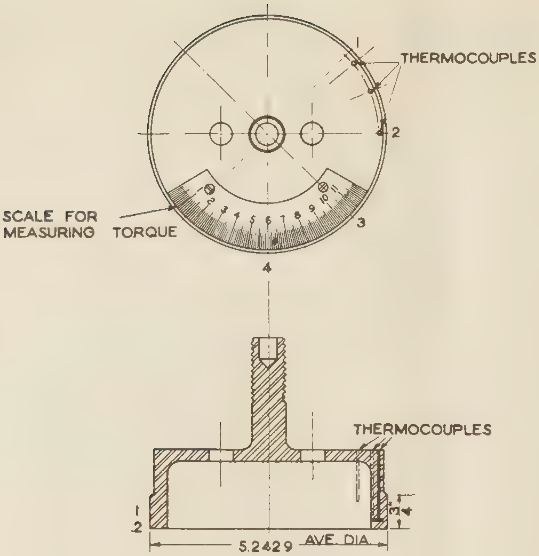


Fig. 2 Details of Ring

Radial position	Axial positions 1	2
1	5.2430	5.2429
2	5.2429	5.2429
3	5.2429	5.2429
4	5.2428	5.2429

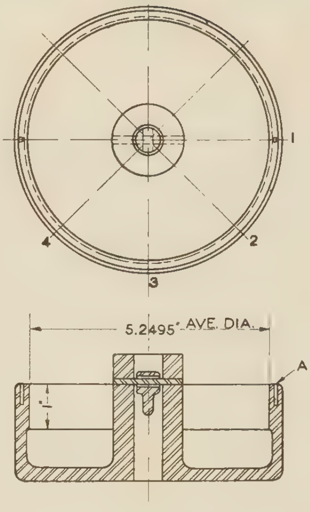


Fig. 3 Details of Bowl

Radial position	Axial positions 1	2
1	5.2494	5.2497
2	5.2494	5.2494
3	5.2495	5.2494
4	5.2494	5.2497

stopped was determined by extrapolating the curve to zero time.

This method was not found to be practicable because only 10 or 12 sec were required for the bowl to reach the temperature of the ring. This was usually too short a time to permit obtaining a sufficient number of readings to determine accurately the form of the time-temperature curve.

In the second method, a thermocouple junction was pressed lightly against the bowl and the indicated temperature observed. Since the rubbing of the junction against the bowl produced friction which resulted in temperature, precautions had to be taken to keep this low and uniform, and also to determine its amount. The thermocouples were mounted as shown in Fig. 4,

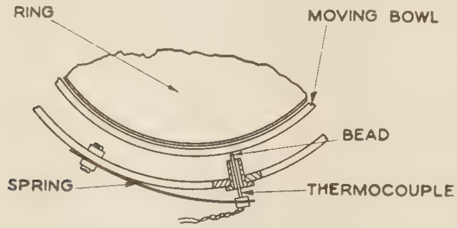


Fig. 4 Detail of Thermocouple on Bowl

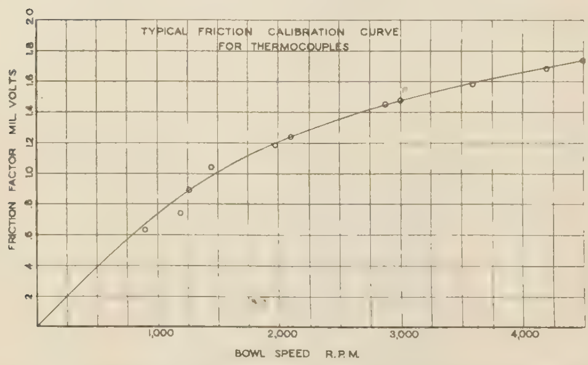


Fig. 5 Typical Friction Calibration Curve for Thermocouples

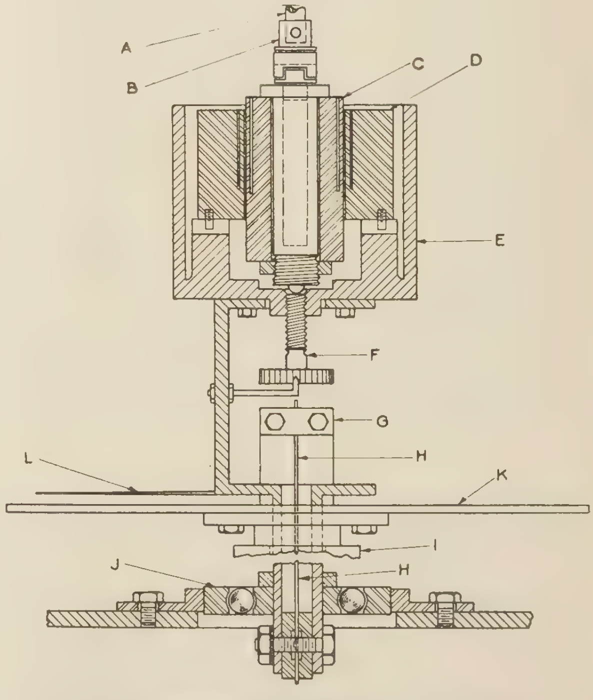


Fig. 6 Arrangement of Tapered-Plug Viscosimeter

A—Drive from drill press
B—Oldham coupling
C—Rotating plug
D—Stationary ring
E—Oil cup
F—Adjusting screw
G—Stationary grip
H—Torsion rod
I—Upper steady bearing
J—Lower steady bearing
K—Drill-press table
L—Pointer indicating angle of twist

and were pressed against the bowl by means of a light flat spring. They were so arranged as to be out of contact with the bowl except when readings were desired. This was done to prevent wear on both bowl and junction. Just enough pressure was used to keep the junction in contact with the bowl at all times.

In order to determine the amount of temperature rise due to friction the bowl was brought to room temperature. A run was made with the ring removed and the temperature indication at various speeds noted. A plot was then made of temperature against speed. Fig. 5 is typical of these plots. One of these calibration runs was made before each test run to determine the torque-speed relationship. The temperature rise due to friction, taken from this curve, was then subtracted from the temperature indicated by the thermocouple during the torque-speed run. The difference was taken to be the temperature of the outer surface of the bowl.

A rough check was made to determine the drop in temperature between the inner and outer surfaces of the bowl. As this indicated a drop of only about 1 deg, no attempt was made to correct for it.

Preliminary runs showed that severe lateral vibration of the rod suspending the ring occurred at each of several critical speeds. It was overcome by enclosing the rod in a pad of sponge rubber which damped out the vibrations but was sufficiently yielding to permit the ring to center itself without bending the rod unduly. This damping pad is shown at *A* in Fig. 1.

The second piece of apparatus consisted of the well-known tapered-plug viscosimeter, built from descriptions given by Albert Kingsbury.[3] Only slight departures were made from his specifications. The most important were (1) the introduction of an Oldham coupling in the drive in order to permit the plug to center itself with respect to the bearing, and (2) the introduction of a certain amount of freedom in the mounting of the bearing within the oil-container cup, in order to make possible the centering of the bearing over the suspension rod. Fig. 6 shows the arrangement. The dimensions of the plug and bearing are shown in Fig. 7.

Two thermocouple wells were drilled into the bearing, as shown, as close to the bearing surface as possible. The thickness of metal separating the bearing surface from the thermocouple well was about $^1/_{32}$ in. Two wells were similarly placed in the plug. These also are shown in Fig. 7.

[3] "Heat Effects in Lubricating Film," by Albert Kingsbury, *Mechanical Engineering*, vol. 55, Nov., 1933, pp. 685–688.

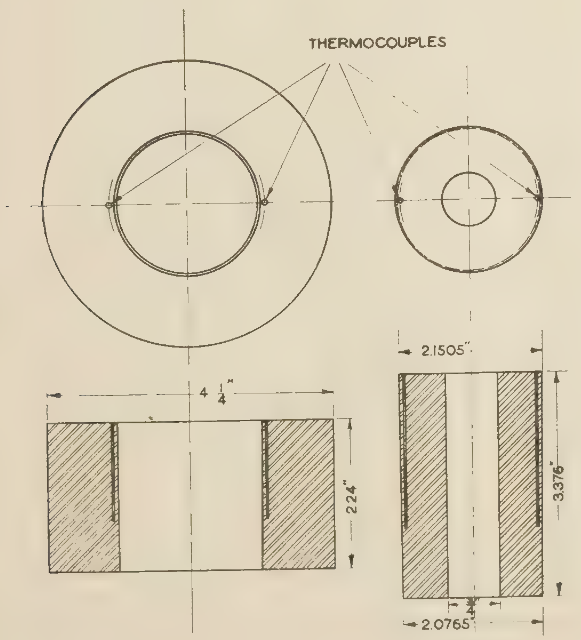


Fig. 7 Details of Plug and Bearing

Test Procedure

When using the rotating-bowl apparatus, the ring was first centered within the bowl while the latter was at rest. Oil, heated to about 180 F, was then supplied to the bowl by a gravity feed, and the bowl was brought up to the desired speed. The oil supply was then adjusted so as to be just sufficient to make good the quantity thrown out through the clearance space. The speed was held constant until the thermocouples indicated that constant temperature conditions had been reached. This required about 30 min. The angle of twist of the suspension rod and the temperature of each thermocouple were then read.

When using the tapered-plug viscosimeter, the oil was placed in the cup holding the bearing and raised to a level slightly above the top of the plug, care being taken to see that the clearance space between the plug and the bearing contained no air. The supporting micrometer screw at the bottom of the cup was then adjusted to give the clearance at which it was desired to operate. The plug was then rotated at constant speed until constant torque and temperature conditions were reached. The angle of twist of the suspension rod and the temperatures of the thermocouples in the bearing were observed, after which the plug was stopped and the temperatures of the thermocouples in it were noted.

The mean temperature of the oil film was taken as the mean between the temperatures of the ring and the bowl in the case of the first apparatus and the mean between those of the bearing and plug in the case of the tapered-plug viscosimeter.

The tapered-plug apparatus was used for rates of shear between 1330 and 239,000 reciprocal sec. The ring-and-bowl apparatus was used for rates of shear between 50,000 and 320,000 reciprocal sec.

Five kinds of oil were used. These are listed together with their viscosities in Table 1. It will be noted that this list includes mineral oils from several geographical fields and also one vegetable oil, namely, olive oil.

TABLE 1 VISCOSITIES OF OILS USED

Oil	Absolute viscosity—centipoises 130 F	210 F
Havoline 10 W	14.8	4.15
Olive	20.4	7.05
Quaker State 10 W	15.2	4.33
Sunoco 10 W	10.6	3.13
Texaco 10 W	15.1	4.14

Results of Tests

Petroff's equation states that the torque required to rotate a journal which is concentric with its bearing is directly proportional to the product of the absolute viscosity and the rate of shear of the fluid separating the journal and bearing. The plot of torque against rate of shear should, therefore, be a stright line passing through the origin, if the viscosity is constant. Otherwise, it will be a curve of some sort, depending upon the manner in which the viscosity varies. This relationship was used as a convenient means of detecting any variation in viscosity which might be caused by variations in the rate of shear.

The values of torque, observed on each of the machines described, were corrected for temperature and original viscosity differences to one arbitrarily chosen value. The resulting values were then plotted against the corresponding values of the rate of

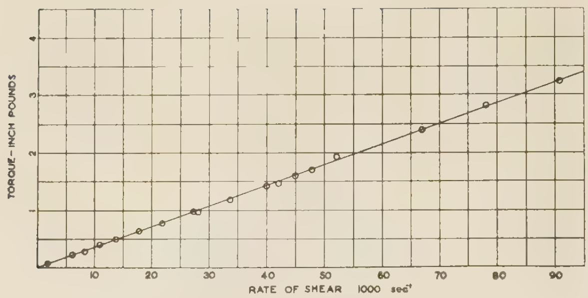


Fig. 8 Torque Versus Rate of Shear on Tapered Plug
(Oil used: Havoline 10 W.)

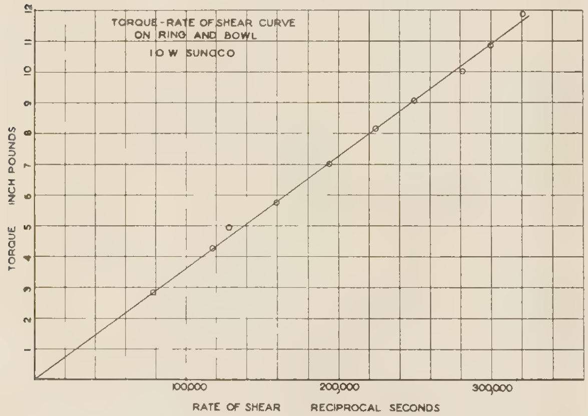


Fig. 9 Torque Versus Rate-of-Shear Curve on Ring and Bowl
(Oil used: Sunoco 10 W.)

shear. Any variation in viscosity, other than that due to temperature, would cause a departure of these points from the line representing Petroff's equation. Fig. 10 shows the results obtained. There is no consistent deviation from the Petroff line

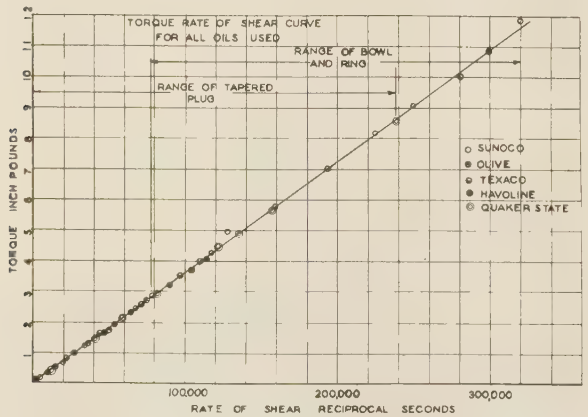


Fig. 10 Torque Versus Rate-of-Shear Curve for All Oils Used

at any rate of shear throughout the range investigated. Furthermore, there is no departure of any single observation to an extent greater than can be attributed to experimental errors normally present in work of this kind. This is true of the observations made with both machines, and the results obtained from each are in complete accord in the portion of the field covered jointly. The agreement of the results with those predicted by the Petroff equation, upon the assumption of the independence of viscosity from the rate of shear, holds for all of the oils investigated.

From these results it would appear that, if there is a change in viscosity due to molecular orientation, it occurs in a portion of the field not investigated by the authors. The most likely portion is the region of very low rates of shear and close proximity of the boundary surfaces. This portion should command our next attention.

Discussion

M. D. Hersey.[4] Can the authors supply information as to the clearances used with the tapered plug, and the range of film viscosities or film temperatures covered in their investigation?

A distinction may be made between Newton's law and Petroff's in that the former requires only proportionality, while the latter involves the calculation of a constant in terms of clearance and length. Should we consider that both laws have been verified and, if so, within what estimated limits of accuracy?

Kingsbury's investigation[3] indicated an approximately parabolic distribution of temperature over the cross section of the film. The temperature drop from the middle of the film to either metal surface in the ring and bowl may be calculated[5] on the assumption of radial conduction under steady-state conditions. For a viscosity of 4.5 cp (Table 1 of the paper) and conductivity 0.016 lb per sec deg F at 210 F, this temperature difference approximates 5.6 F at the maximum rate of shear, 320,000 reciprocal sec. Are such effects negligible within the limits of accuracy of the present work?

Authors' Closure

The tests reported in the paper were undertaken to determine whether the viscosity of the oils investigated varied with the rate of shear. Petroff's equation was made use of because it offered a ready means of detecting variations of viscosity with rate of shear. The authors accepted the statement, "Thus it appears for any given bearing, the friction torque is proportional to the viscosity of the lubricant and the speed," which appears on page 26 of Mr. Hersey's book "Theory of Lubrication."[5] Their claim is that the results secured support the conclusion that viscosity does not vary within the limits of the investigation.

It is felt that since the exact thermal conductivity of the oils treated was not available, calculation of the effect of temperature gradient across the thickness of the film would not add to the accuracy of the results.

[4] Research Director, Morgan Construction Company, Worcester, Mass. Fellow A.S.M.E.

[5] "Theory of Lubrication," by M. D. Hersey, John Wiley and Sons, Inc., New York, N. Y., 1936, pp. 116–117.

Effect of Temperature on Coiled Steel Springs Under Various Loadings

By F. P. ZIMMERLI,[1] DETROIT, MICH.

This paper consists of a presentation of the results of tests conducted in the laboratories of the author's company on the effect of various stress-temperature combinations on steel springs. These results are in the form of charts which, because of the immense number of tests involved, are believed to be accurate. For the carbon steels tested, it is shown that there is a temperature-stress equilibrium point about 400 F. Below this point there is a definite temperature-stress equilibrium. Above this point it is simply a time-temperature curve since, eventually, the springs will fail. The paper draws particular attention to the value of various strain-relief heatings after coiling.

Tests on springs hardened and tempered after coiling, as compared with those made of pretempered wire, give no evidence that the former method is to be preferred. This is contrary to the general understanding in the industry.

The paper shows that for each type of material there is an optimum Rockwell hardness for best heat resistance. Both S.A.E. 6150 and 9260 steels have greater resistance to load losses due to heating than carbon steels. They in turn are exceeded by 18-8 stainless and high-speed steels. A difference in time between 10 days and 3 days to reach true temperature-stress equilibrium exists between the various steels.

EVERY mechanical device, somewhere within itself or in its production, calls for the use of springs. During the last few years, in particular with the demand for greater operating speeds, the temperatures at which these springs function have continually increased. Data regarding the temperature-stress relationship of springs have as yet no general publication, and appreciation among engineers of the results which may be attained has lagged.

To remedy the condition, the laboratories of the author's company about 10 years ago commenced the task of testing all available spring materials in this respect. It soon became apparent that the amount of work involved would be immense and that too long a time would be required for completion of the original program.

To the end of speeding up the work, the International Nickel Company agreed to investigate its products, such as monel, inconel, Z nickel, and the like, in this regard. Preliminary tests on copper alloys, such as brass and bronze, proved that these materials were useless above 225 F and they were eliminated, leaving the scope of the project to cover coiled steel springs.

Before tests could be conducted for the purpose of obtaining the necessary design information, it was essential to know: (*a*) How long a time was necessary for a loaded spring to be held at a temperature in order to reach equilibrium;[2] (*b*) the effect of the degree of flexibility in the spring; (*c*) whether a spring should be pressed solid before or after the heating operation which removes coiling strains; and (*d*) whether or not the process of wire manufacture influenced the results to any great extent, i.e., would wire, built to the same specification by competing manufacturers, act the same?

Preliminary Tests

For these preliminary tests springs were coiled from pretempered material, ground, and heated to 800 F. They were then pressed solid with a load 100 lb in excess of their carrying capacity. The length, outside diameter, load, and wire size of each spring were noted. The springs in sets of 10 were placed over bolts which pushed steel collars against them and thus compressed the spring to predetermined lengths, checked with micrometers. The springs were then exposed to the desired temperature for different periods of time. They were then removed and checked again for load carried. The load testing was carefully performed, the springs being checked on the scales to within 0.001 in. in height, using a 0.001 gage, so that no error due to scale travel was possible.

Stresses were calculated before and after heating, using the Wahl formula

$$S = \frac{8PD}{\pi d^3}\left(\frac{4c - 1}{4c - 4} + \frac{0.615}{c}\right)$$

where $c = D/d$
P = load, lb
D = mean diameter, in.
d = wire diameter, in.
S = stress in outer fiber, psi

All stresses are those due to loads at room temperature and are not the stress on the wire at oven temperature. If it is desired, a correction could be made for this by obtaining the modulus at the various temperatures employed. This value was reported[3] by W. P. Wood, G. D. Wilson, and the author in 1930.

If G = modulus at room temperature, G_1 at oven temperature, S the stress at room temperature, then S_1, stress under oven conditions, has the relation

$$\frac{S}{S_1} = \frac{G}{G_1} \quad \text{or} \quad S_1 = \frac{SG_1}{G}$$

This can be proved from the formula of spring deflection

$$f = \frac{8PD^3N}{Gd^4}$$

where N = number of coils, and the other symbols are as previously stated.

The springs were not corrected for bolt expansion because the

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Contributed by the Special Research Committee on Mechanical Springs, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[2] By "equilibrium" is meant a stable condition such that further exposure to a given temperature will not cause any additional loss in load carried by the spring.

[3] "The Effect of Temperature Upon the Torsional Modulus of Spring Materials," by W. P. Wood, G. D. Wilson, and F. P. Zimmerli, Proceedings of the A.S.T.M., vol. 30, 1930, part 2, pp. 351–360.

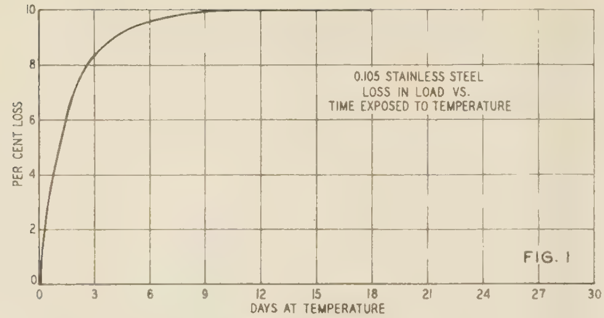


FIG. 1

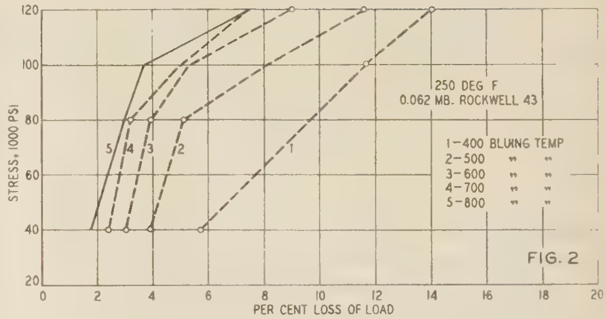


FIG. 2

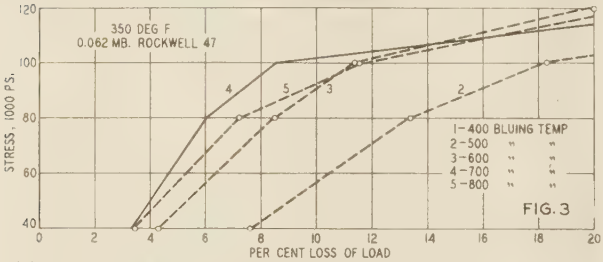


FIG. 3

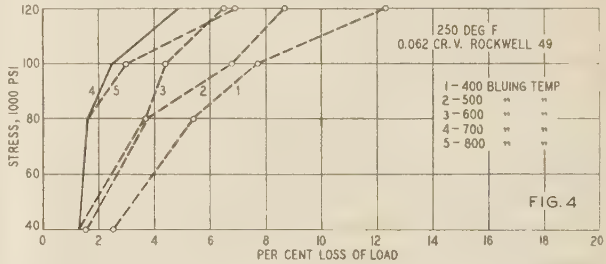


FIG. 4

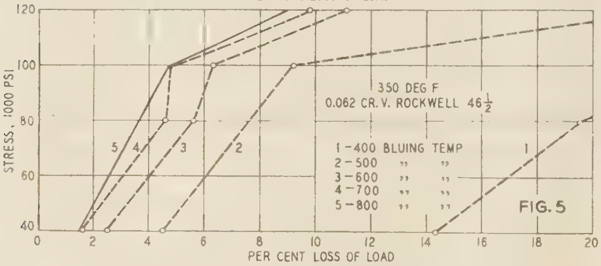


FIG. 5

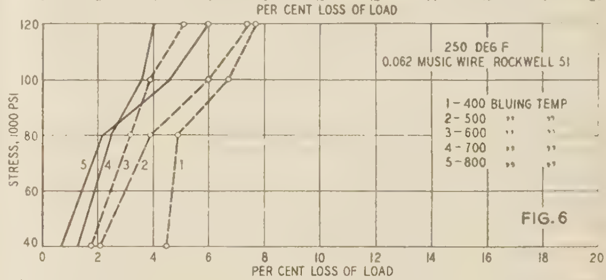


FIG. 6

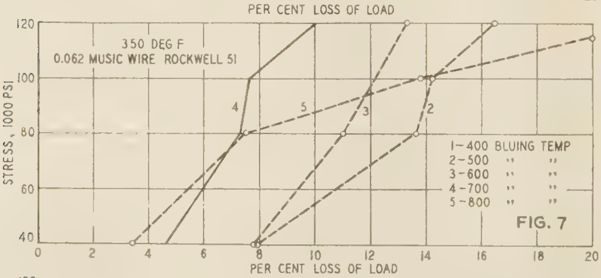


FIG. 7

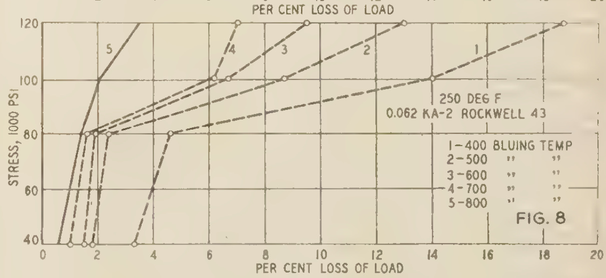


FIG. 8

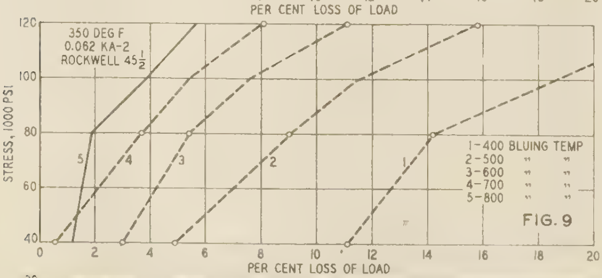


FIG. 9

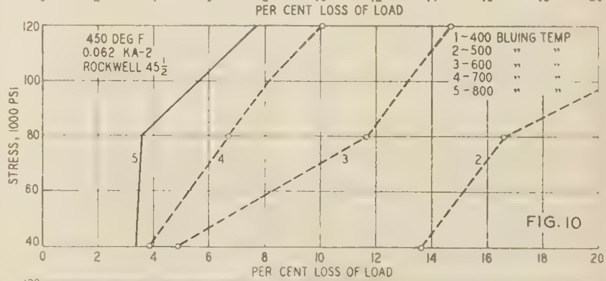


FIG. 10

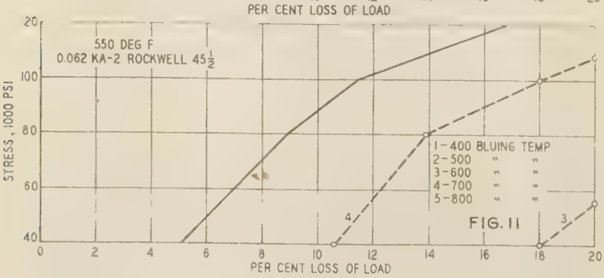


FIG. 11

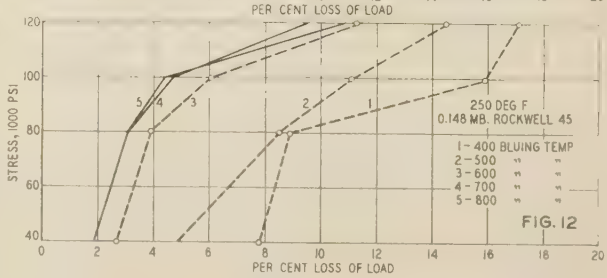


FIG. 12

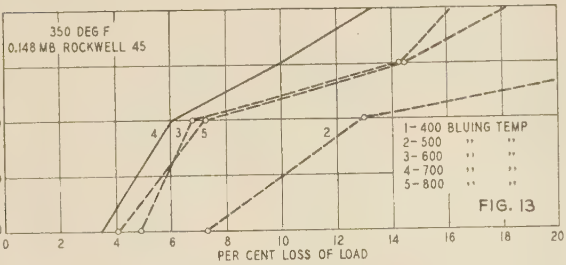


FIG. 13

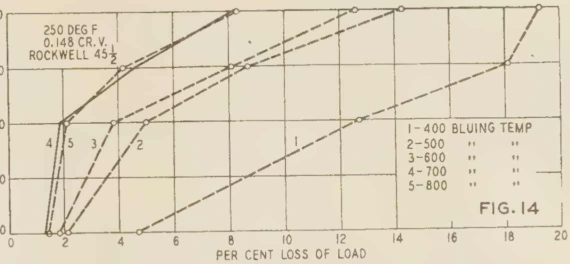


FIG. 14

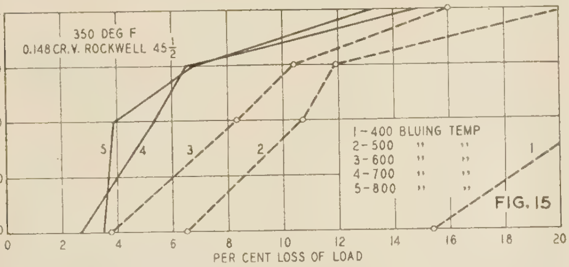


FIG. 15

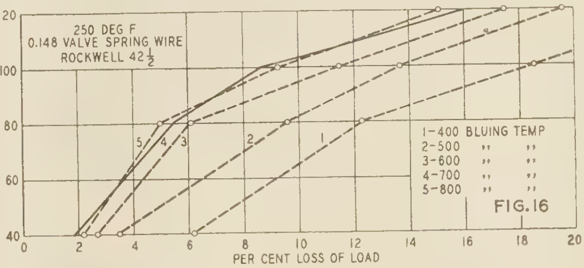


FIG. 16

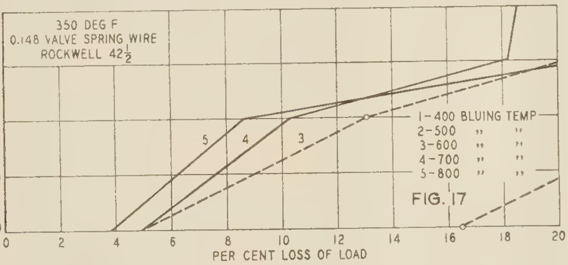


FIG. 17

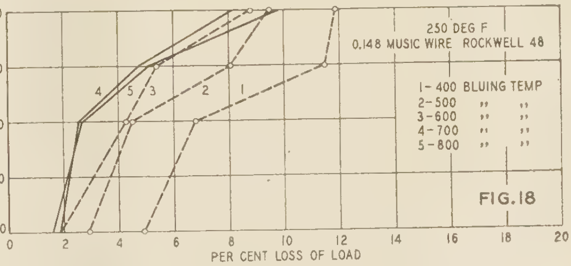


FIG. 18

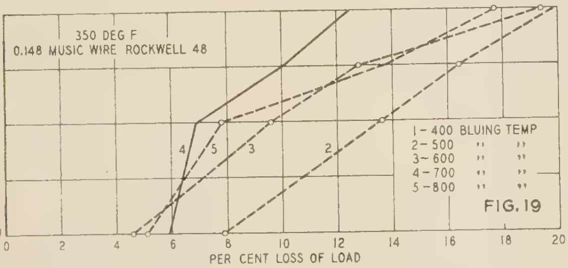


FIG. 19

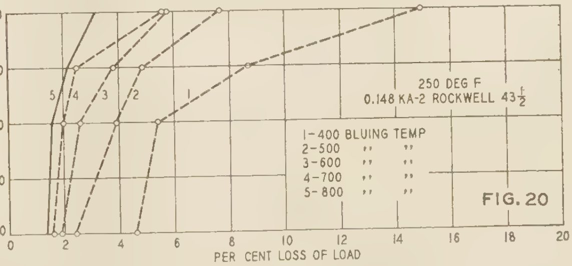


FIG. 20

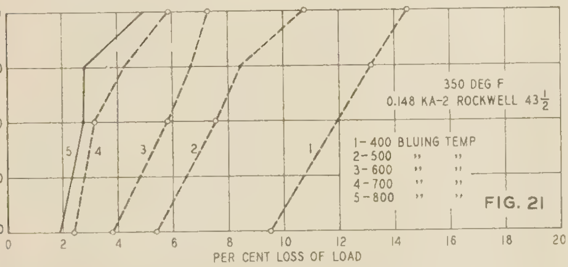


FIG. 21

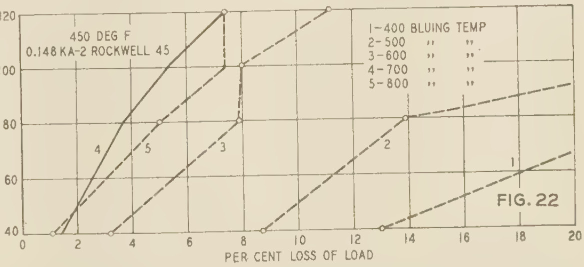


FIG. 22

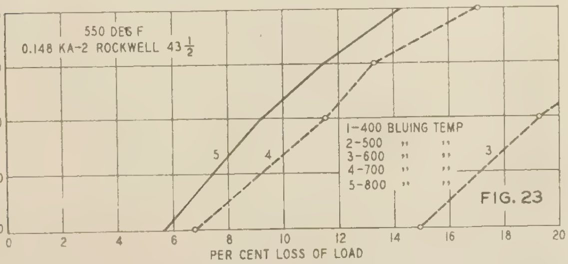


FIG. 23

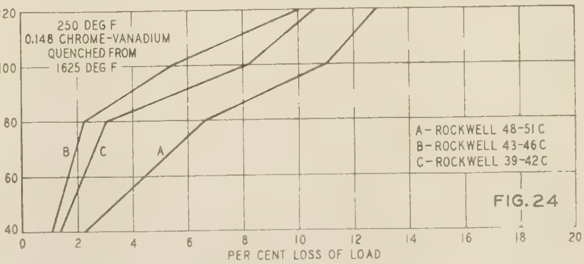


FIG. 24

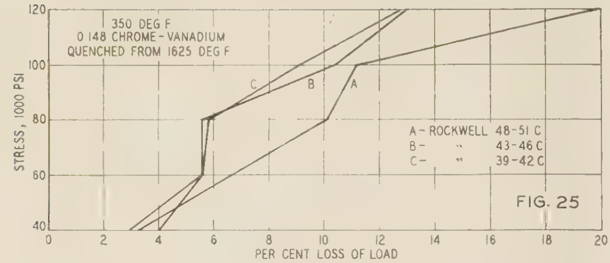


FIG. 25

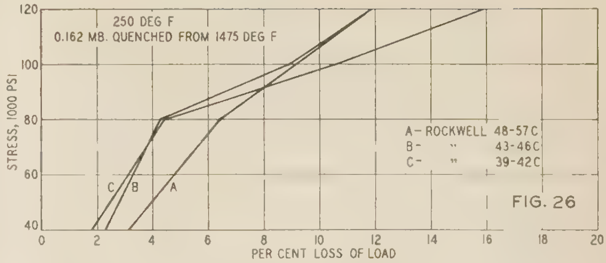


FIG. 26

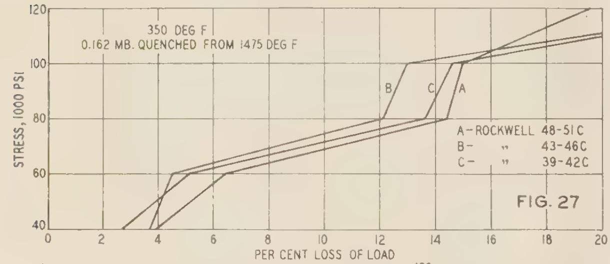


FIG. 27

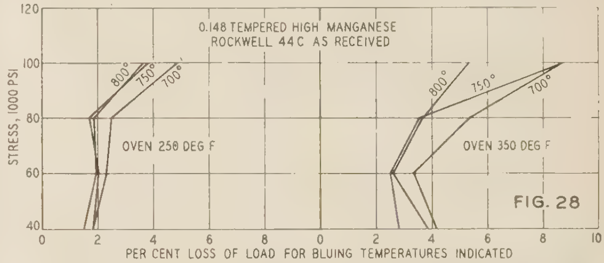


FIG. 28

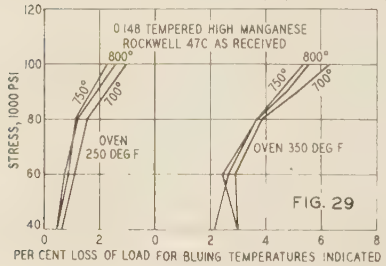


FIG. 29

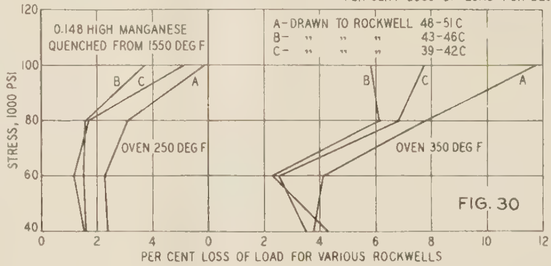
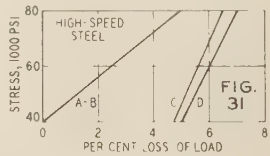


FIG. 30



CURVE	DRAWING TEMP	ROCKWELL "C"	OVEN TEMP
A	1150° F	57-59	600° F
B	1250	52-52.5	600
C	1150	57-59	700
D	1250	52-52.5	700

FIG. 31

wire in the spring actually expanded slightly to cover this error.

Lots of ten springs were set at a given stress and temperature and run for successive periods until no further loss of load was noted. Then another combination of temperature and stress was similarly tested until the entire field of possibility had been covered. These results indicated that the usual spring steels should be divided into two major groups, i.e. (1) straight carbon steel, low-alloy steels, and (2) high-alloy steels.

The first group consisted of S.A.E. 1065, X1065, 1080, 1095, 6150, and 9260. The second group was composed of 18-8 stainless steel and the usual 18-4 high-speed steel. Group 1 definitely reached equilibrium within 72 hr at heat, regardless of stress or temperature, provided the latter was less than 400 F. Tests up to 216 hr gave the same results as 72 hr, while 60 hr were extremely close, losses being but small fractions of 1 per cent of the total. Above 400 F, there was no time at which equilibrium was reached until the coils of the spring, when released, were still closed against each other. This is quite similar to the behavior of steel in tension at elevated temperatures, where it has been noted that, up to the equicohesive temperature, creep is observed for a definite time. It is postulated that the metal becomes strain-hardened at that time and resists further deformation.

The two samples in the second group were tested in a similar manner and equilibrium was reached not in 72 hr but in 10 days. A curve is appended which gives results of 18-8 stainless steel subject to 250 F at 120,000 psi for a period up to 18 days. This material will not reach equilibrium if the temperature greatly exceeds 550 F, but the set is slow so that for some uses it is satisfactory at slightly higher temperatures. High-speed steel is similar, except that higher temperatures are possible. The later discussion of commercial production tests will amplify these statements.

The next point to be considered was the rate of deflection of the spring itself. To test this, twenty springs were made from the same bundle of music wire with the same outside diameter, in the manner previously outlined. Ten had $5^1/_4$ coils and ten had $10^1/_4$ coils. When stressed at 40,000 psi, the following result was obtained after 72 hr:

$10^1/_4$ coils......................	loss 2 per cent
$5^1/_4$ coils......................	loss 1.6 per cent

A further test at 100,000 psi was made with the following results:

$10^1/_4$ coils......................	loss 9.3 per cent
$5^1/_4$ coils......................	loss 10.4 per cent

It was concluded that the degree of flexibility of the spring was not a factor governing loss of load due to temperature.

Two hundred springs were divided into two lots of 100 each. These springs were made of oil-tempered wire. One lot was pressed solid before the bluing or strain-relieving draw of 750 F, and the other lot was pressed after the heating. The springs were subject to various stresses at temperatures up to 400 F. Results showed that springs pressed after heating were more uniform and slightly better. Therefore, this procedure was adopted as standard.

The final preliminary tests were conducted on springs made from wire manufactured by four different mills. Slight differences were noted, the products of some mills showing a greater loss than others. However, in wires other than the hard-drawn quality this difference was not excessive. Hard-drawn wire varied greatly. Evidently, each manufacturer patents his wire differently (if at all). The results were so confused that it was decided not to run the production tests on this cheap type of material.

Procedure for Production Tests

The procedure in running the remaining tests, the results of which are shown in the accompanying curves, was as follows: A sufficient number of springs of each of the materials to be tested

TABLE 1 CHEMICAL COMPOSITION OF MATERIALS USED IN TESTS

Material	C	Mn	P	S	Cr	V	Ni
0.148 Music wire	0.91	0.31	0.018	0.022	....	....	....
0.148 M B	0.66	0.76	0.020	0.036	....	....	....
0.148 Cr V	0.52	0.75	0.007	0.020	0.87	0.18	....
0.148 Swedish valve spring	0.65	0.56	0.021	0.019	....	....	....
0.148 KA-2	0.24	0.42	.....	.....	18.2	....	9.21
0.062 Music wire	0.91	0.31	0.024	0.018	....	....	....
0.062 M B	0.59	0.75	0.020	0.025	....	....	....
0.062 Cr V	0.50	0.73	0.009	0.018	0.97	0.18	....
0.062 KA-2	0.12	0 41	.....	.....	19.2	....	9.14
0.148 High Mn (tempered)	0.70	1.34	0.022	0.024	.	....	....
0.148 Cr V (annealed)	0.54	0.69	0.011	0.026	0.89	0.17	....
0.162 M B (annealed)	0.62	0.73	0.018	0.055		.	....
0.148 High Mn (annealed)	0.70	1.34	0.022	0.024	.	...	....
0.152 High-speed steel	0.76	0.31	(tungsten 18.03)		3.83	1.10	....

TABLE 2 ROCKWELL C HARDNESS OF MATERIALS USED IN TESTS AFTER HEATING TO

Material	400 F	500 F	600 F	700 F	800 F
0.148 Music wire	52	51	52	48	45.5
0.148 M B	45	46	47.5	45	42
0.148 Cr V	47	46	46	45.5	44.5
0.148 Swedish valve spring	43	43	43	42.5	43
0.148 KA-2	44.5	45.5	45	45	43.5
0.105 Music wire	52.5	53	51	51	48
0.062 Music wire	52	52	52.5	51	47.5
0.062 M B	49	50	50	47	43
0.062 Cr V	49	49	49.5	49	46.5
0.062 KA-2	45	45	45	46	45.5

	As received	700 F	750 F	800 F
0.148 High Mn	46	46	45	44
0.148 High Mn	44	44	44	43
0.152 High-speed steel		(See data sheet)		

were obtained, about 120,000 pieces in all being required. These springs were heated to the various strain-relieving temperatures, indicated on the charts, for 30 min at heat, in an L & N Homo furnace. Ten springs were used for each stress-temperature test and the average loss plotted on the curves. All tests, except on stainless steel and high-speed steel were run 72 hr at heat. These high-alloy steels were given 10 days to be sure an equilibrium condition was obtained. All springs were pressed solid after heating for stress-relief. This was done with a load 100 lb greater than that necessary to close the coils. It was hoped that this procedure would aid in establishing uniformity of tests.

The springs were tested on accurately checked scales using a 0.001-in. gage, in order to obtain the desired load and height on each spring for a given stress. The springs were placed in a constant-temperature electric oven, being held on special bolts, with square seating collars, to the desired height and load. Upon removal from the oven, the springs were air-cooled and retested for load at room temperature to the nearest 0.001 in. and the results expressed as percentage loss of load corresponding to the given stress were plotted.

In working with these various materials, the data given are to the highest possible commercial application of the product. Thus, some curves indicate temperatures only as high as 350 F. Tests at 400 F on these same steels demonstrated that the steels were erratic, hence, not of commercial importance. Therefore, no data sufficiently accurate for any design problems could be presented. At 450 F, these selfsame steels simply collapsed when given sufficient time.

The effect of wire size in the limits tested is not great. Curves are presented in the interests of economy on only two sizes, i.e., $^{1}/_{16}$ and 0.148 in. Data are not available on any size larger than $^{3}/_{16}$ in. In general, the tendency is toward greater load loss with larger sizes at the same stress. The curves are as plotted from points taken at a minimum of four different stress figures. To reduce possible error, each of these points is the average value of ten springs.

Even with all these precautions it will be noted that the data give results which at some temperatures cause the curves of various bluing temperatures to cross each other. These points have been checked in some cases and the same result obtained. At present no good explanation has been developed for this experimental fact.

The materials used in these tests are listed in Table 1 and the hardness figures in Table 2.

Conclusions

In the author's opinion, a study of the work done justifies the following conclusions:

1 The usual spring steels are reliable when stressed 80,000 psi or less up to temperatures of 350 F. Between 350 F and 400 F and, at stresses up to 120,000 psi, the same continuity of results is lacking, but with proper forethought some commercial success might be expected.

2 The use of ordinary spring steels over 400 F is not possible.

3 Steels, hardened and tempered after coiling into springs, at the same hardness value, have no advantage over springs made of pretempered wire properly blued, under the conditions investigated.

4 Stainless steel of the 18-8 type resists temperature and stress better than other spring steels, except perhaps high-speed steel.

5 A middle hardness range in quenched-and-drawn springs is preferable to either high or low ranges.

6 An optimum temperature to heat springs after coiling for heat resistance is the highest one which will not render the hardness or other physical properties of the material objectionable.

7 The present Swedish valve-spring wire stands heat very poorly, in fact, is less satisfactory than many other steels.

8 Both high-manganese and silicomanganese steels equal the chrome-vanadium steel tested and may have commercial advantages.

Acknowledgments

The author wishes to acknowledge the assistance of the entire laboratory force in preparing this work; in particular A. C. Stenhouse, now with Vauxhall in England, G. D. Wilson, and Glen Brookes.

It can be readily appreciated that this is the work of many when it is realized that over 250,000 separate spring weighings were made in addition to other testing work.

It is to be hoped that other investigators will carry on the work into the field of larger wire sizes. The possibility of different heats and heat-treatments of high-speed steel and stainless steel should be completed. In particular, the investigation of the newer steels now in use and which we have not tested, such as the high-chromium series and chromium-molybdenum types, should be started if complete data are desirable.

Discussion

R. C. Zeidler.[4] As a user of great quantities of springs, the company with which the writer is connected has experienced its share of problems. In some small measure the experience gained during the last year and a half may contribute to the general knowledge of the subject.

Our larger springs are used as clutch-pressure springs. While they are subject to high temperatures, and design limitations impose high stresses, they are under only static loads. Consequently, we are not interested in fatigue life but only in springs which show a minimum of load loss under hard clutch-operating conditions. For practical and economical reasons, all of our

[4] Assistant Engineering Manager, Long Manufacturing Division, Borg-Warner Corporation, Detroit, Mich.

pressure springs are of S.A.E. 1065 oil-tempered or similar steels, covered by the author's MB specification. The following discussion pertains to this class of steel.

Our method of testing is similar to that used by the author, except that the springs are pressed twice to 300 lb, which is $1^1/_2$ to 2 times the normal load of the springs, before final selection for load is made. This is done to offset variations in load caused by failure of the vendor to remove most of the set and by poor packing and rough handling in shipping. Thus, springs supplied by various vendors are all placed on an equal test basis. Any further set therefore is due to the heat alone.

The data given in this discussion are for springs with stresses ranging from 75,000 to 106,000 psi, corrected according to Wahl formula and Rockwell hardness 43–46 C scale. Through a series of tests it was found that, after 15 hr in the oven at 350 F, the springs had lost approximately 90 per cent of the total load they would lose if allowed to remain in the oven until the equilibrium point was reached. This was found to be sufficient for all practical purposes so that 15 hr has been adopted as our standard, since it permits a ready overnight check. Preliminary tests showed that these springs are not practical above 400 F, with the stress above 75,000 psi. Some springs tested showed 8 to 10 times more loss between 400 and 500 F than between 300 and 400 F, but only 1.5 times more between 300 and 400 F than between 200 and 300 F.

In ordinary clutch service, these springs probably would never reach a temperature of 400 F but, in order to check this point, a number of clutches returned from actual field service were examined. Some of these appeared to have had normal usage, while others evidently had been abused. Other clutches of known qualities were given severe road tests. From the average load losses of the springs, the probable temperatures reached were found to be in the 200 to 300 F range. As a consequence, it was decided that, for an accelerated test, 350 F would be adopted as a standard. At this temperature, springs still react somewhat along theoretical lines.

Together with numerous other tests, twenty-five different types of pressure springs, as received from various vendors, were all given a 20-min treatment at 750 to 775 F and then pressed twice under 300 lb. They were then carefully selected for load, clamped between test plates at their respective working heights, and placed in the oven at 350 F for 15 hr.

Of the twenty-five springs tested, seventeen came within the stress range of 75,000 to 96,000 psi. The maximum load loss on the springs of this group was $5^1/_2$ per cent. The other eight springs, ranging from 96,000 to 106,000 psi, showed a gradual increase to 9.5 per cent. This does not mean that all of the springs fell exactly on a straight line between these points but that an approximate average was obtained. We found several unexplainable discrepancies, such as the author mentioned in his paper.

As the result of the foregoing tests, the design drawings for these springs now call for a maximum allowable load loss due to heat. Routine checks are made daily on six springs from each production shipment. These are placed in the oven the day they are received and allowed to remain for 15 hr overnight, thus making them available for production the following day if they are satisfactory.

While we feel that only a beginning has been made from our findings up to the present time, we believe that an ordinary grade of oil-tempered wire, in a spring of reasonable design, given the proper manufacturing attention, will provide a clutch spring which will be satisfactory for all practical purposes. Further, we believe that still greater improvement can be made without the use of more expensive steels.

Author's Closure

Mr. Zeidler's remarks are very interesting and timely. The results, assuming his mean stress to be 85,000 lb per sq in. for the first 17 springs, check our results within 1 per cent and are lower due to the time interval he used. On the 8 springs whose mean stress is 100,000 lb per sq in., curve 13 gives a 10 per cent loss in load. This is within $^1/_2$ per cent of Mr. Zeidler's figures. We consider this an excellent check on the utility and accuracy of our work. We are extremely glad that Mr. Zeidler has come forward with this information.

Turbine Discharge Metering at the Safe Harbor Hydroelectric Development

By J. M. MOUSSON,[1] BALTIMORE, MD.

This paper discusses the suitability, calibration, and reliability of certain piezometer systems installed in low-head units of high capacity. An account is also given of a research to determine and develop a suitable type of flowmeter to be operated by the differential pressure from these piezometer systems for continuous integration, indication, and graphic recording of unit and plant discharges. The type of equipment installed is presented in detail, as well as its adaptation as an automatic guide to operation, resulting in appreciable benefits through higher operating efficiencies.

Introduction

Although continuous automatic accounting of unit discharge is not new, several recent improvements and developments have entirely changed the aspect of desirability for apparatus of this kind, as many of the shortcomings of earlier installations, limiting their usefulness, have been successfully overcome.

While, in some plants, automatic water accounting has been carried out for years, the necessary equipment has often been regarded as a luxury, particularly, as its sole purpose was usually confined to collecting runoff data at the project site to augment records of existing gaging stations or, perhaps, replace those of stations rendered inoperative in a project area due to construction of a particular plant. Since the accuracy of river gaging is essentially not very high, and decidedly lower than that required for turbine-discharge measurements for acceptance tests, it has been standard practice to keep unit- and powerhouse-draft records by means of computations based on power output. At the same time, however, it has been generally recognized that the installation of input-measuring apparatus would be highly desirable, if and when unit-discharge and station-totalizing equipment of sufficient accuracy and within economical reach were available to serve as a yardstick for plant operation, both as to proper and efficient loading of the units and to detect troubles affecting their efficiencies.

To illustrate the difficulty of the solution to this problem, it may be mentioned that, while equipment of this kind was contemplated at Safe Harbor at the outset of construction in 1930, as at that time already certain provisions had to be incorporated in the substructure, on the generator-room floor, in the conduit system, and in the control room, nevertheless, the various investigations and development work required a substantial amount of time and it was not until late in 1938 that suitable equipment was finally installed.

1—INVESTIGATIONS PURSUED

The various investigations carried out dealt not only with the exploration of the principle to be employed, but also with the possible consistency, sensitivity, and suitability of various apparatus. In the first place, it had to be established that the index method, based on differential piezometer deflection, is of sufficient accuracy as a basis of continuous water measurements.

Piezometer Installation

In each of the substructures of the six main units comprising the initial development, there were installed three piezometers of the Winter-Kennedy type[2] in the turbine scroll and two piezometers of the Peck type[3] on one of the stay vanes of the speed ring, Fig. 1. While one of the Winter-Kennedy taps was placed in the high-pressure low-velocity region, the two other taps were located radially opposite thereto at the speed ring in the low-pressure high-velocity region, one just above the speed ring and the other tapped in the crown of the speed ring.

The Peck piezometer locations are shown in Fig. 2, the impact tap in the nose and the low-pressure tap in the flank of the stay vane. In the first four main units to be installed, the Peck impact tap was located at the nose tip. On the fifth main unit it was placed at a slight angle to the longitudinal axis of the stay vane, $^3/_4$ in. from the nose tip, and on the sixth unit to be installed at a still larger angle, that is, 45 deg and $2^1/_{16}$ in. from the stay-vane tip. At the same time, some shift in upstream direction of the Peck low-pressure tap was also made on the latter two units.

In addition, two auxiliary piezometer openings were located at

[2] "Improved Type of Flow Meter," by I. A. Winter, Proc. American Society of Civil Engineers, vol. 59, part 1, 1933, pp. 565–584.

[3] "Two Methods of Measuring Water to Hydraulic Turbines," *Power*, vol. 77, March, 1933, pp. 126–127.

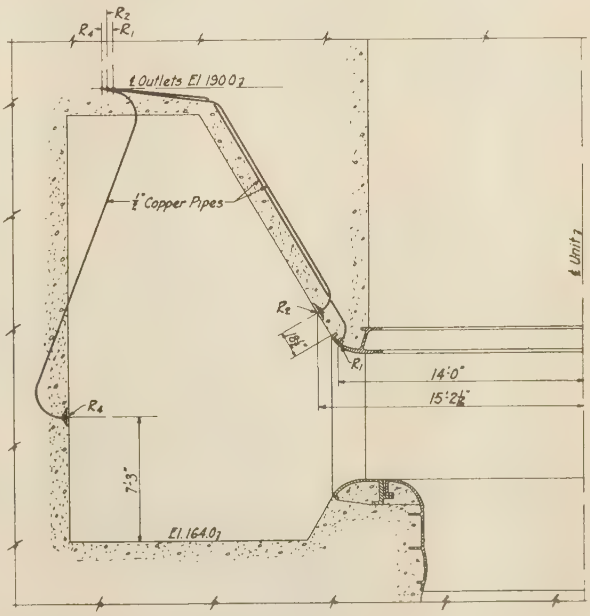


Fig. 1 Winter-Kennedy Piezometer-System Installation at Main Units
(Taps R_1, R_2, and R_4.)

[1] Hydraulic Engineer, Safe Harbor Power Corporation. Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the downstream nose of one of the intake piers of each main unit for possible use should pumping with the units ever be resorted to for peak storage requirements during low flow. Both service units were provided with two piezometers of the Winter-Kennedy type. To prevent air pockets, all piping leading to the individual piezometer openings was placed with a continuous slope and copper piping was used to prevent corrosion. In the pipe tunnel beneath the generator-room floor, a piezometer board with vertical glass tubes was installed at each unit where the deflections could be measured in feet of water.

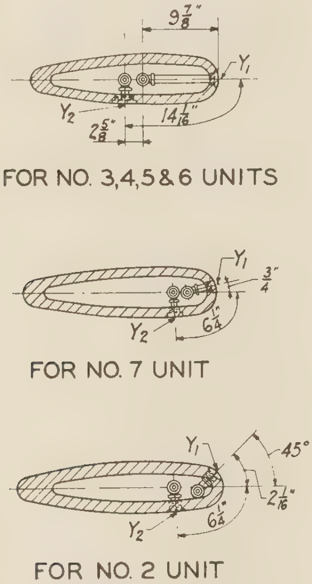


FIG. 2 LOCATION OF PECK PIEZOMETERS ON MAIN-UNIT SPEED-RING STAY VANES
(Taps Y_1 and Y_2.)

After placing each unit in service, it was essential, as a first step, to determine which combination of two piezometers would prove most consistent. This was done by plotting the differential pressure of any two taps against that of any one of the other possible pairs. From Fig. 3, it may be noted that the three Winter-Kennedy taps and the Peck impact tap showed a markedly better consistency than the Peck low-pressure tap Y_2, the latter being responsible for the erratic behavior in three of the plots. On the other five main units, the results were similar with the exception that even the Peck impact tap, located closer to the nose or at the very nose tip of the stay vane, was considerably less steady. For all main units, the Winter-Kennedy taps showed a high degree of consistency.

This result should not be interpreted as a general weakness of the Peck type of system. Investigating the origin of this erratic behavior, that is, through analysis of the results with the various Peck tap locations, as shown in Fig. 2, it was found that the cause for instability, particularly of the low-pressure tap, was rather in the design of the stay vanes than in the type of piezometer system. The Safe Harbor stay vanes are comparatively short and have a straight longitudinal axis. Since, on the one hand, the low-pressure tap was erratic in all units, irrespective of the shift upstream, and, on the other hand, the consistency and magnitude of deflection of the impact tap increased decidedly by the shift away from the nose tip, it could be concluded that the stay vanes of the speed ring were not pointed head on into the flow but at a considerable angle, causing a region of local disturbance on one side of the stay vanes, with the unstable region extending almost to the very tip of the vane. In the light of these results and, in view of the experience obtained elsewhere with piezometers of the Peck type, it would appear that a considerable improvement in stay-vane design is yet to be accomplished by lengthening, better streamlining, and curving these vanes. It is noteworthy

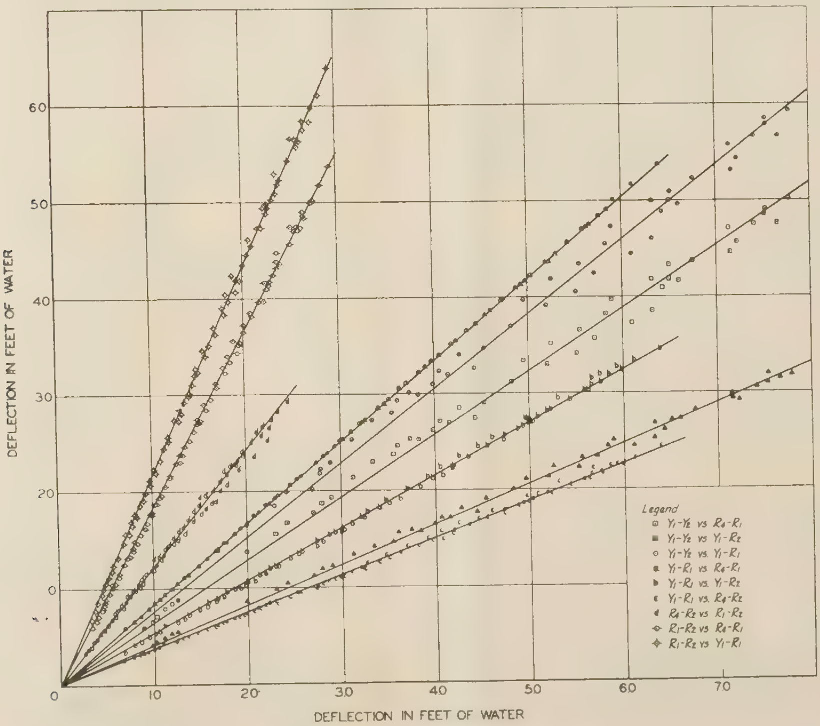


FIG 3 ANALYSIS OF PIEZOMETER DEFLECTIONS OBTAINED AT NO. 2 UNIT

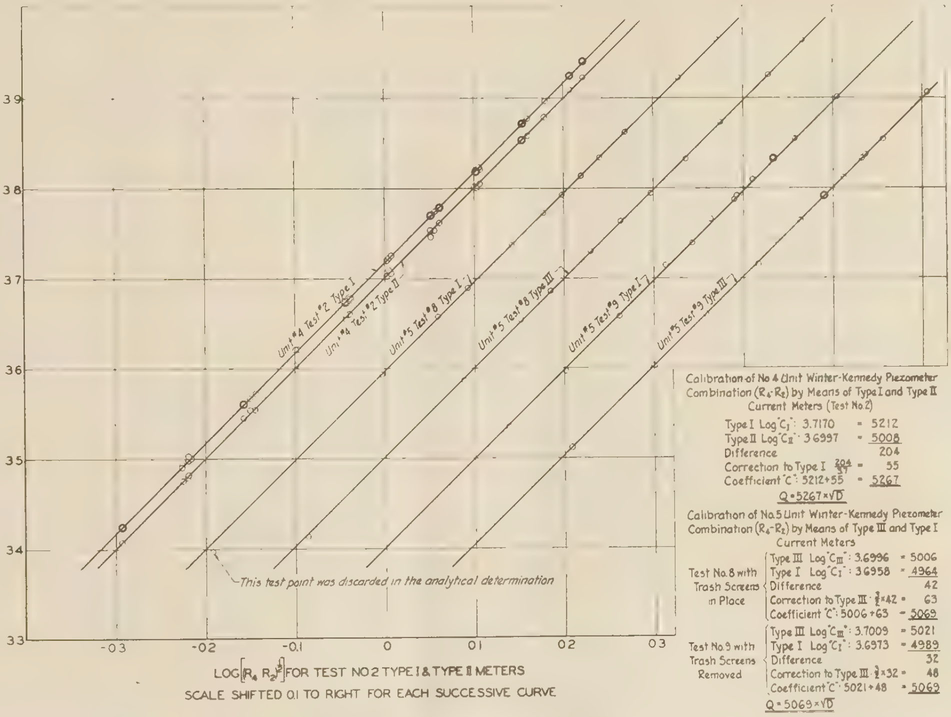


Fig. 4 Calibration of Piezometer Pair (R_4-R_2) at Nos. 4 and 5 Units by Means of Two-Type Current-Meter Method

that, in more recent installations, some improvements in this direction already have been made.

While the piezometers were used initially simply as a relative index to determine the proper relation between turbine-blade and guide-vane positions under various operating heads for the Kaplan main units and served as a basis for the cam designs controlling the gate-blade relation, these piezometers were calibrated for absolute-discharge measurements in course of the acceptance tests by means of the two-type current-meter method.[4] The results obtained for the piezometer pair (R_4-R_2) (refer to Fig. 1) of the Winter-Kennedy system of two main units are shown in Fig. 4. These curves relate piezometer deflection and discharge in accordance with the fundamental equation

$$Q = C \times D^a$$

where Q is the discharge measured in cubic feet per second and D the differential piezometer pressure in feet of water. The slope of the curves and their intercept at zero corresponding to the exponent a and coefficient C, respectively, were determined analytically, based on the method of least squares.

Of the six main units, three were tested by means of current meters.[4] The piezometers of the other units were calibrated indirectly by assuming their peak efficiencies to be identical with those of other units of the same design and manufacture actually tested. This procedure was also followed for the two identical Francis-type service units in testing one of them by means of current meters and assuming the peak efficiencies of both to be alike.

It is recognized that such a procedure is not absolutely correct because identical units have not necessarily identical peak efficiencies. However, based on experience available, it is believed that the error thus introduced will not exceed 1 per cent for any one unit and that the average for the entire station should be even closer, because the actual efficiencies of these units might be higher or lower. The calibrations of the piezometer pair (R_4-R_2) of the Winter-Kennedy systems on the six main and the two service units are given in Table 1.

TABLE 1 CALIBRATION OF PIEZOMETER PAIR (R_4-R_2) OF WINTER-KENNEDY SYSTEMS

Main unit no.	Coefficient C	Departure from average, per cent	Calibration procedure
2	5107	−0.80	Current meters
3	5090	−1.13	Based on No. 5 unit
4	5267	+2.30	Current meters
5	5070	−1.52	Current meters
6	5185	+0.72	Based on No. 4 unit
7	5170	+0.43	Based on No. 5 unit
Average	5148		
Service unit No.			
41	330.8	−0.86	Based on No. 42 unit
42	336.6	+0.86	Current meters
Average	333.7		

Recognizing the fact that piezometers are very sensitive and greatly affected by local disturbances, due to irregularities of the water passage, as well as due to minute changes in the shape of the piezometer opening, the differences in the coefficients are relatively small.[5] The variation in the exponent a of the equation ($Q = C \times D^a$) was also very small, varying between 0.500 ± 0.005, so that for all practical purposes the square root was found to be of sufficient accuracy.

To guard against unexpected trouble in the future, which would render one or the other piezometer unreliable or useless, all taps were calibrated during these tests. For instance, Fig. 5 shows the calibration of No. 2 unit Peck impact tap Y_1 and Winter-Kennedy low-pressure tap (R_2) combination (Y_1-R_2).

The calibrations of the piezometers also permitted arriving at some conclusion regarding the degree of consistency and relative precision of these systems, Table 2. The consistency or average

[4] "Water Gaging for Low-Head Units of High Capacity," by J. M. Mousson, Trans. A.S.M.E., vol. 57, 1935, pp. 303–316.

[5] "Piezometer Investigation," by C. M. Allen and L. J. Hooper, Trans. A.S.M.E., vol. 54, 1932, pp. 1–11.

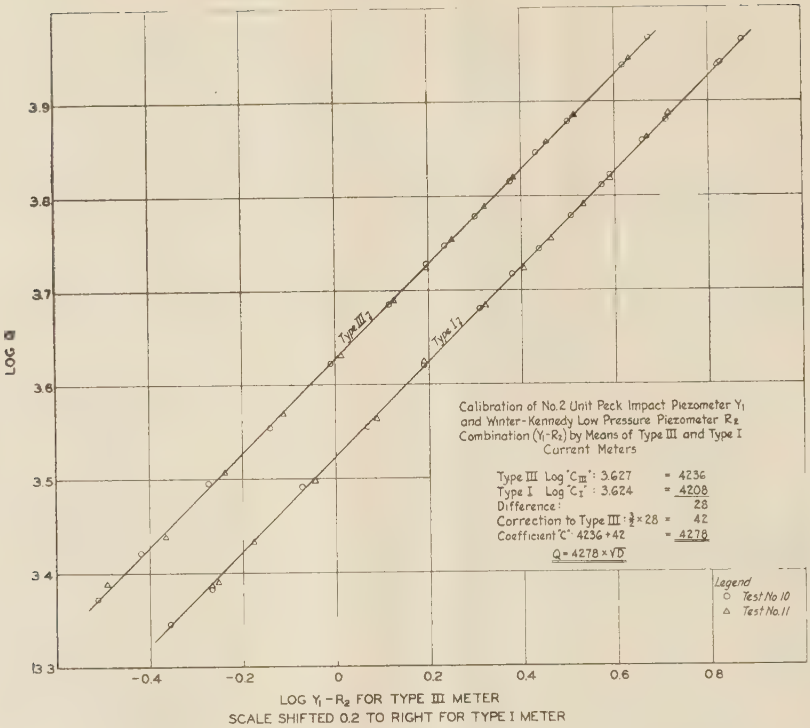


Fig. 5 Calibration of Piezometer Pair (Y_1-R_2) at No. 2 Unit by Means of Two-Type Current-Meter Method

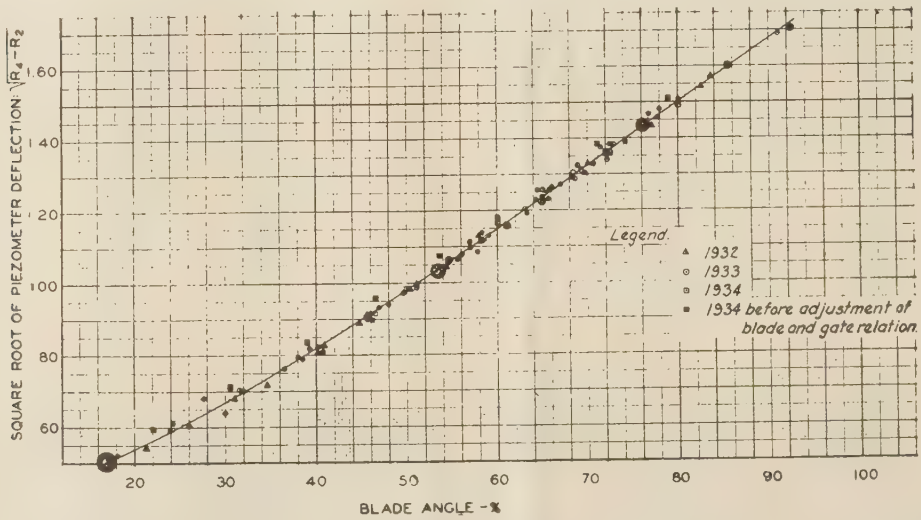


Fig. 6 Check on Consistency of Piezometers Using Turbine-Runner Blade Opening as a Parameter

departure of one test point was determined analytically to ±0.25 per cent for the measurements with the type I current meter, shown in Fig. 4. The mean departure of one measurement was ±0.34 per cent. The mean departure of all measurements was ±0.06 per cent and the relative precision ±0.04 per cent. Considering that these values include not only the errors of the piezometer system but also those of the current-meter measurements, it is believed that the accuracy of the systems is fully adequate as a basis for continuous-flow measurements.

As a next step it was essential to see whether or not the piezometers would maintain their calibrations over a period of years. The results for one unit and one pair of piezometers of the Winter-Kennedy system are shown in Fig. 6, using the runner-blade angle of the Kaplan units as a parameter. In this instance a somewhat wider dispersion of the test points as compared with that in Figs. 4 or 5, is to be expected as the measure of blade angle is not very accurate due to the inevitable lag in the blade-operating mechanism. The results indicate, however, that during the period of observation no change in calibration had taken place. Of particular interest are the test points obtained in 1934, prior to proper adjustment of the blade-gate relation which for some reason had become slightly incoordinated.

TABLE 2 DETERMINATION OF CONSISTENCY AND RELATIVE PRECISION OF PIEZOMETER PAIR (R_4-R_2); CALIBRATION OF NO. 4 UNIT

Test Run	Turbine Discharge for Type I Current Meter Q_1 c.f.s.	Square Root of Piezometer Deflection $\sqrt{R_4-R_2}$	$\frac{Q}{\sqrt{R_4-R_2}}$	Departures δ	Departures Per Cent	δ^2
1	7518	1.448	5192	-20	0.38	400
2	5946	1.140	5216	+ 4	0.08	16
3	4752	.917	5182	-30	0.58	900
4	4768	.915	5211	- 1	0.02	1
5	3687	.708	5208	- 4	0.08	16
6	3735	.718	5202	-10	0.19	100
7	3182	.604	5268	+56	1.07	3136
8	3145	.605	5198	-14	0.27	196
9	3163	.608	5202	-10	0.19	100
10	5245	1.006	5214	+ 2	0.04	4
11	6594	1.268	5200	-12	0.23	144
12	6571	1.262	5207	- 5	0.10	25
13	8729	1.664	5246	+34	0.65	1156
14	8697	1.666	5220	+ 8	0.15	64
15	8415	1.613	5217	+ 5	0.10	25
16	8401	1.612	5212	0	0.00	0
17	7875	1.510	5215	+ 3	0.06	9
18	7444	1.424	5228	+16	0.31	256
19	7426	1.427	5204	- 8	0.15	64
20	6611	1.278	5173	-39	0.75	1521
21	6651	1.276	5212	0	0.00	0
22	5987	1.153	5193	-19	0.36	361
23	5994	1.155	5190	-22	0.42	484
24	5887	1.127	5224	+12	0.23	144
25	5858	1.127	5198	-14	0.27	196
26	5289	1.011	5231	+19	0.36	361
27	5301	1.015	5223	+11	0.21	121
28	4726	.906	5216	+ 4	0.08	16
29	4712	.906	5201	-11	0.21	121
30	4174	.797	5237	+25	0.48	625
31	4173	.801	5210	- 2	0.04	4
32	3632	.697	5211	- 1	0.02	1
33	3634	.697	5214	+ 2	0.04	4
34	3115	.595	5235	+23	0.44	529
35	3091	.595	5195	-17	0.33	289
36	2657	.510	5210	- 2	0.04	4
37	2664	.510	5224	+12	0.23	144
			Avg =5212		Avg =0.25	$\Sigma\delta^2$ =11537

Results

Consistency or average departure of one measurement = $\pm 0.25\%$

Mean departure of one measurement = $\pm\sqrt{\Sigma\delta^2/(n-1)} = \pm\sqrt{11537/36} = \pm\sqrt{320.472}$ $= \pm 17.90 = \pm 0.34\%$

Mean departure of all measurements $= \pm\sqrt{\Sigma\delta^2/n(n-1)} = \pm\sqrt{11537/(37 \times 36)} = \pm\sqrt{8.661}$ $= \pm 2.94 = \pm 0.06\%$

Relative Precision of all measurements = $0.674 \times 0.06\% = \pm 0.040\%$

This may be regarded as one example demonstrating the degree of sensitivity of piezometers and how useful they may be to detect improper operating conditions.

Flowmeter Investigation

During 1934 and 1935, three types of flowmeters, each employing a different principle, were investigated in detail to determine which type would meet the rigid requirements or could be further developed to a satisfactory stage. Aside from a minimum amount of maintenance desired, the chief requirements stipulated were a high degree of accuracy and sensitivity over the useful range and the possibility of totalizing the unit flow automatically for the entire station, as well as metering characteristics permitting short duration tests on each unit to determine its efficiency. The basic principles employed by these types of meters were as follows:

For the first type of meter the differential pressure of two piezometer taps served to establish flow in a system, the intake being the high-pressure tap and the exit the low-pressure tap, the rate of flow through this system varying with the differential pressure or discharge through the turbine. The meter consists of a drum about 10 in. diam and 5 in. long with the axis of the drum or cylinder in a horizontal position. A vertical partition divides the drum into two half-cylindrical chambers. This partition supports the hollow central core of the drum. If the drum were split open its cross section would be similar to a wheel with two spokes. In the central hollow core of the drum, there is located a knife-edge bearing or a ball bearing to allow the drum to swing back and forth. The two drum chambers are interconnected through an orifice located near the lower end of the partition, that is, near the drum periphery.

By utilizing the flow through the system to displace mercury from one half-cylindrical drum chamber into the other through the orifice, there results a rotational movement of the drum around its own axis. Water displaced by the mercury in the second drum chamber is discharged through the low-pressure tap. When reaching a certain predetermined limit of tilting, a four-way cock operated by a mercury switch is turned 90 deg, changing the feed from the high-pressure tap to the other drum chamber and also connecting the low-pressure tap to the opposite chamber, thus reversing the flow of mercury and, accordingly, the direction of rotation of the drum. In continuous operation, a cyclic rotational drum movement is obtained similar to a pendulum motion; and the larger the differential pressure, the shorter the time required for each cycle. A counter, operated by the limit mercoid switches mounted on either side of the drum, records the number of drum swings and can be calibrated to serve as a flow integrator through the turbine.

The second type of meter employed the differential-piezometer pressure to lower or raise a float or dome also through displacement of mercury, the dead weight of the float being balanced by a counterweight supported by a cable fed over a pulley. By means of gears, the motion of the pulley shaft may be utilized for instantaneous-flow indication. The integration of flow is accomplished through a clock-operated disk driving a small wheel attached to the cable. The cable movement changes the position of the small wheel and places it at a certain distance from the disk center. While at a high rate of flow, the small wheel is placed close to the disk periphery and, therefore, operating under a high gear ratio, it is placed in the disk center at zero position of the meter and, consequently, does not rotate at all. The small wheel driven by the disk in turn operates an integrating counter. At the same time, a graphic record of the unit discharge can be obtained by means of a pen recording the cable movement or position on a drum making 1 revolution per 24 hr.

The third meter type employed a radically different principle. It is based on the fact that the centrifugal force exerted by a flyball system has the same relation to the rate of rotation that the differential pressure has to the rate of flow. The essential parts of this instrument are a tilting mercury manometer and a motor integrator carrying a flyball system, so arranged that the centrifugal force due to rotation of the integrator is opposed to the force of the tilting manometer, Fig. 7. A mercury switch operated by the beam of the tilting manometer controls the motor speed, maintaining a balance between the centrifugal force of the motor integrator and the piezometer-differential force acting upon the tilting manometer. Since the force due to the piezometer differential varies with the square of the flow being measured and the centrifugal force of the integrator also varies with the square of the integrator speed, the two square laws accordingly cancel, thus leaving a direct relation between flow and integrator speed. A counter geared to the motor integrator shows revolutions in terms of flow.

While the first type of flowmeter referred to was found to be extremely accurate even for very low differential pressures, that is, low turbine discharges, its main disadvantage lay in necessitating a continuous flow through the piezometer piping system. For the measurement of gas, steam, filtered water or any refined fluid, there would be no danger from plugging up the piping, but with silt-laden river water, such as carried by the Susquehanna River, there was great danger of rendering the entire piezometer piping

system useless, even with frequent flushing by compressed air or filtered water. At the same time, this apparatus did not lend itself particularly to totalizing, because a like periodicity of the drum motion on different units would not correspond to equal unit discharges, due to the difference in the piezometer calibrations. Theoretically, it could be compensated for by introducing different gear ratios for the individual counters or by changing the size or location of the orifice connecting the two drum chambers or even through adjusting the amplitude of the cyclic rotational drum movement. In view of the various disadvantages and complications, this type of meter, though accurate, could not be given any further consideration.

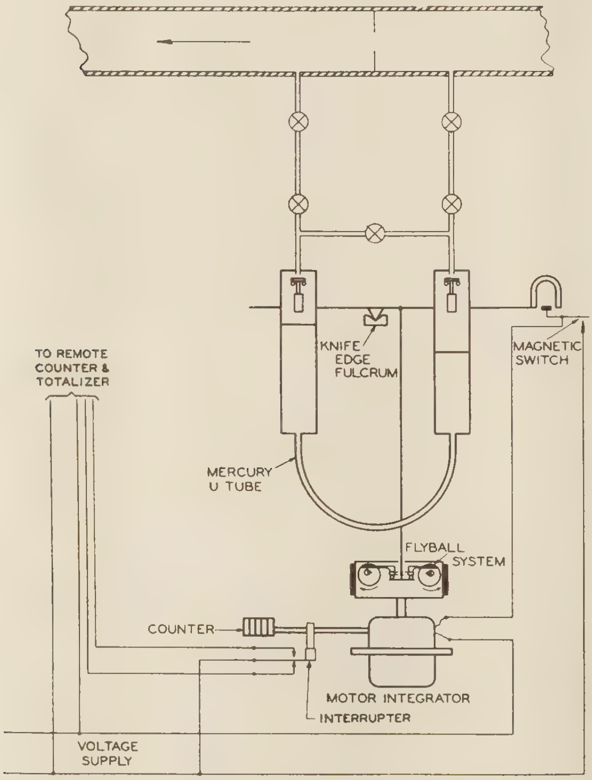


Fig. 7 Diagrammatic Sketch of Type 3 Flowmeter
(By courtesy of the Leeds & Northrup Company.)

Two flowmeters of the second type were purchased and installed temporarily on separate units in the pipe tunnel beneath the generator-room floor. Prior to shipment, these meters were calibrated by the manufacturer for the respective piezometer systems. As a first step, hourly readings of the meters were compared with analytically determined unit discharges based on output. As expected, the flowmeters showed consistently larger unit discharges, the discrepancy being more the greater the fluctuation in loads carried by the units. With the units operating on hand control and blocked to generate at a constant output, there was close agreement between metered and analytical discharges. These results may be attributed to the concave shape of the unit-efficiency curves.

Next, these flowmeters were used to make turbine-efficiency tests of 5- and 10-min duration, the flowmeters being read every 15 sec and the watt-hour-meter-disk revolutions and the time in seconds being recorded by a chronograph. On each unit, the test points thus obtained spread considerably over a band about 3 per cent in terms of efficiency. This discovery led to analysis of the instrument errors by making standard water-column tests. Three major sources of errors were revealed. A first error was traced to the eccentricity of the integrating disk. This error was not constant but had a periodic sinusoidal characteristic completing the cycle in $^1/_2$ hr, corresponding to the time required for 1 revolution of the disk. While for one of the flowmeters the amplitude of this sinusoidal-error curve varied between +0.63 and −0.38 per cent, the other flowmeter showed disk-error variations between −0.82 and +1.39 per cent. The second error, which could not be controlled, was the variable frequency of the station-service system from which the clock driving the integrating disk obtained its power supply. Since the frequency varied about 1 per cent, it was sufficient to make the disk error inconsistent with time, by causing a phase shift in the disk-error curves.

The third source of error was due to the lap of the counter gears and the weight of the rotating countersweep hand, its weight tending to accelerate the motion in the downstroke and retard it in the upward swing. This phenomenon superimposed another error of periodic characteristic upon the first error referred to. To make matters more complicated, the frequency of the second periodic error was not constant but varied with the discharge, being a function of the speed of the countersweep hand and, therefore, decreasing with increasing discharge.

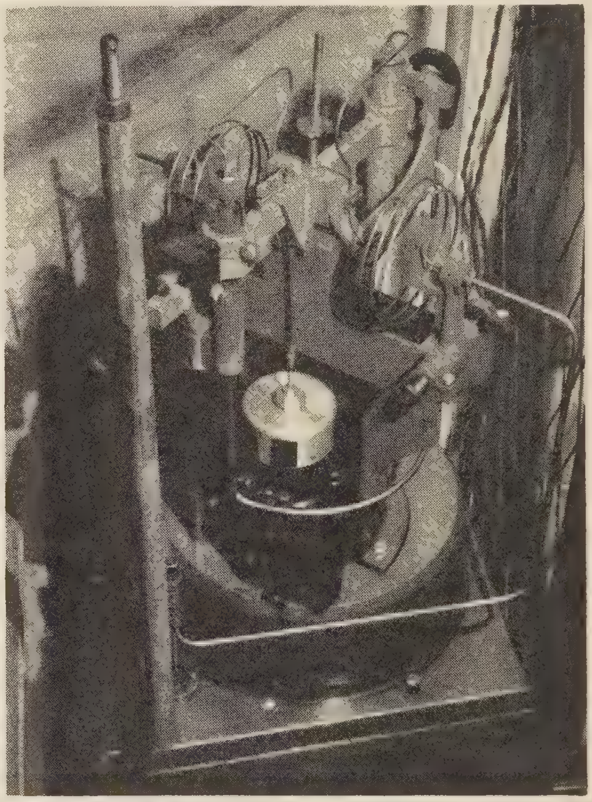


Fig. 8 Temporary Installation of Experimental Type 3 Flowmeter at No. 5 Unit

Although results of short-duration efficiency tests, using compensating measures for the errors referred to, gave greatly improved results, it was realized that such procedures were too complicated to be adopted as a routine measure on all units, because the effort involved, in analyzing instrument errors for meters to

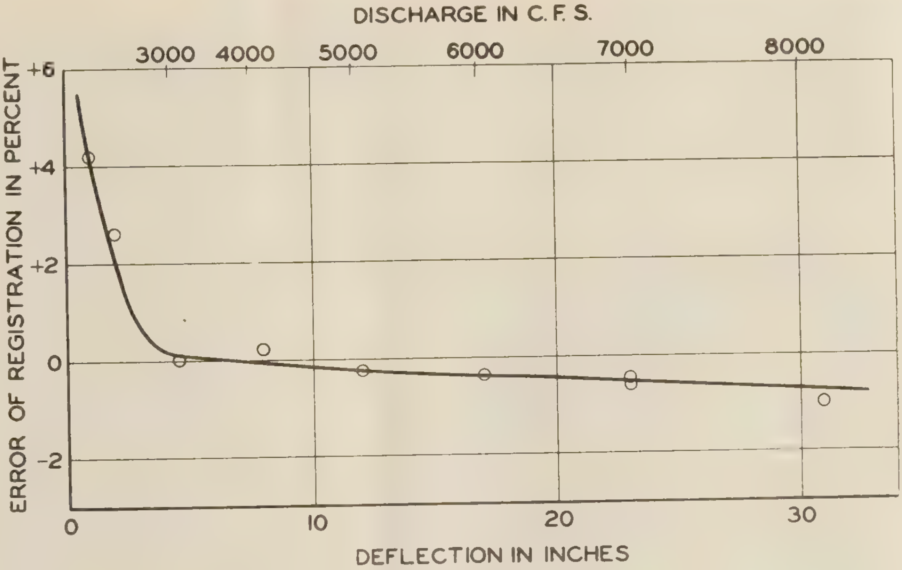


Fig. 9 Error Curve of Experimental Type 3 Flowmeter

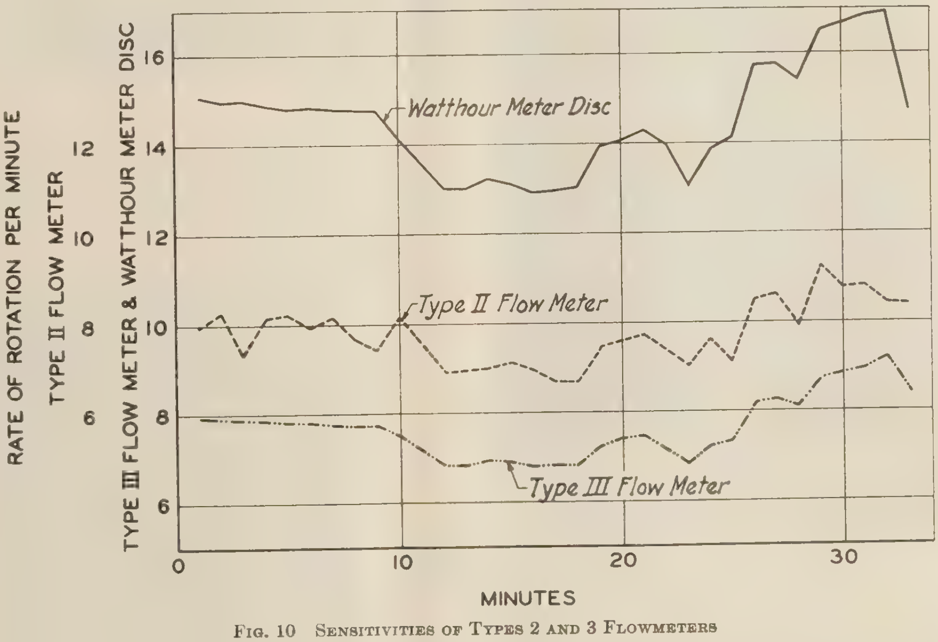


Fig. 10 Sensitivities of Types 2 and 3 Flowmeters

be installed on all units and the use of difficult and complicated compensating procedures for each meter, was far too great in comparison with the accuracy of the results obtained.

An experimental flowmeter of the third type was installed temporarily on one of the units, Fig. 8, and operated in parallel with one of the meters of the second type. Hourly readings on both meters were compared with each other as well as with discharge computations based on power output. The two flowmeters agreed within 0.5 per cent in average, the third type of meter being closer to the analytically determined discharge.

Short-duration efficiency tests on the third type of meter showed a very small spread of test points and a very good agreement with the results of the turbine acceptance tests. The execution of these tests could be simplified considerably by being able to record the flowmeter countershaft revolutions, Fig. 7, by means of electrical impulses on the chart of a recorder simultaneously with the revolutions of the watthour-meter disk and second impulses. The watthour-meter-disk revolutions were obtained by means of a photoelectric-cell arrangement, a small electric bulb being placed on one side of the disk and the photoelectric cell on the opposite side. The beam of light causing the impulses fell through the balancing hole in the watthour-meter disk. The second impulses were obtained also by means of a photoelectric-cell arrangement mounted on the master clock for frequency control of the system, the beam of light being cut by the pendulum.

Next, the meter errors of the third type of flowmeter were analyzed by means of the standard water-column tests. As may

be seen, the error curve as shown in Fig. 9 had the typical shape of a rotational integrating device. While the test points were rather consistent, nevertheless it was concluded that further improvement of the meter should be carried out to flatten and lengthen the horizontal leg of the error curve and improve its accuracy to such a degree that even analytical compensating measures would not be required for short-duration efficiency measurements on the turbines.

Additional tests were carried out to determine the responsiveness and sensitivity of the second and third types of flowmeters with varying load on the generating unit. The results, shown in Fig. 10, demonstrate the consistency of the third type of flowmeter, as it follows the watthour-disk-revolution indications consistently in contrast to those of the second type.

In view of the fact that the totalizing with the third type of meter was a simple electrical problem and well-established principle, it was decided to use the third type of flowmeter for the Safe Harbor installation, provided satisfactory improvements were made by the manufacturer in the error characteristics.

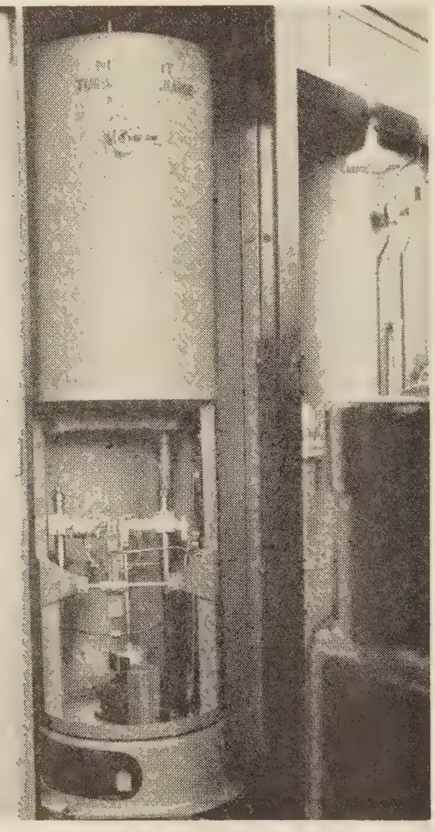


Fig. 11 Typical Flowmeter Installation

2—FLOWMETER EQUIPMENT INSTALLED AT SAFE HARBOR

During 1936 and 1937, various studies were made on remote unit-discharge-totalizing equipment. The manufacturer's attention was drawn also to the possibility of using this type of equipment as part of unit- and station-efficiency indicating-and-recording apparatus. By 1938, the plans for such an installation had crystallized to a point where it was felt safe to proceed with the installation of the flowmeter equipment for all units, as well as the flow-totalizing apparatus for the entire station.

The flowmeters selected were installed in cabinets originally provided on the generator-room floor, located adjacent to and forming an integral part of the gage boards of each unit, Fig. 11. This installation comprised eight flowmeters, one for each of the six main units and one each for the two service units. On all units the flowmeters were connected to the Winter-Kennedy piezometer pair (R_4-R_2) and calibrated, based on the data given in Table 1.

It should be noted that the error characteristics of these meters had been materially improved, so that no correction of any sort had to be applied over the entire range of turbine discharge actually used. The improvement in the error characteristics can best be realized by comparing the check calibrations of three flowmeters after installation, Fig. 12, with the results obtained with the experimental flowmeter in 1934, which is shown in Fig. 9.

Another desirable advantage of these meters is that the checking or recalibration is greatly simplified by calibrated weights to be hung on one arm of the tilting mercury manometer, thus eliminating the use of standard water columns. As demonstrated by the data plotted in Fig. 12, the results obtained by each method are, for all practical purposes, identical.

The power supply for the flowmeter motor integrators was obtained from the 120-v 60-cycle station-service system at outlets

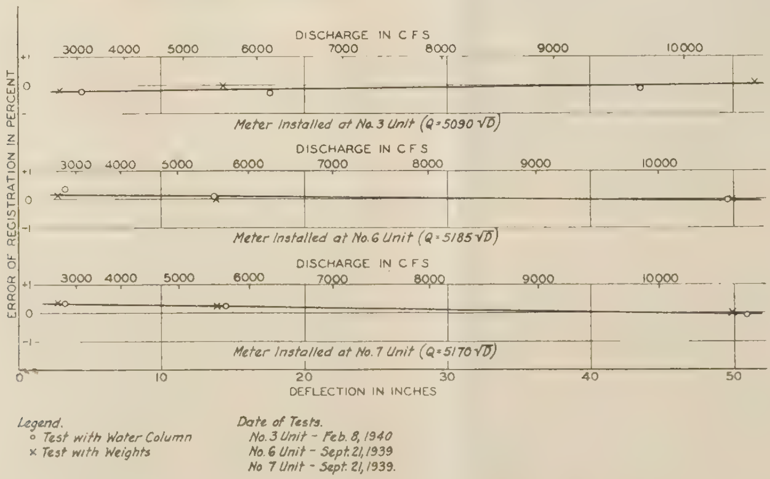


Fig. 12 Check Calibrations of Three Type 3 Flowmeters After Installation

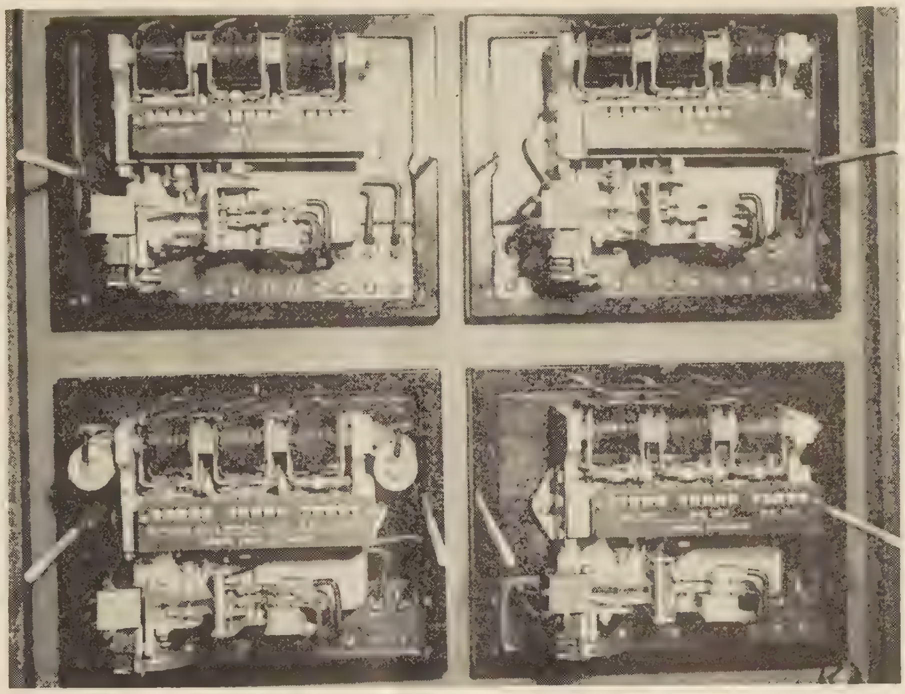


Fig. 13 Discharge Totalizing Relays

available at each unit gage board. The totalizing apparatus was installed on a panel of the relay board in the control room, Fig. 13. Its principal parts consist of four impulse totalizing relays. Three of these serve as unit-discharge totalizers for a group of three turbines each and one as master totalizer for the entire station, giving the sum total of the three unit totalizing relays. The spare position on the first totalizing relay will be used for the flowmeter at No. 1 unit, now being installed. While the input-output ratio of the unit totalizing relays is 5:3, the master totalizer has a ratio of 3:1. Since each impulse sent out by the interrupter on the countershaft of the individual flowmeters represents 20,000 cu ft, each impulse received by the station total discharge counter from the master totalizing relay corresponds to 100,000 cu ft. The station-total counter is mounted below the totalizing relays on the same panel, Fig. 14. Individual unit discharges can be read on the individual impulse counters of the unit totalizing relays. Mechanical counters were provided on all flowmeters in order to facilitate the checking of impulse transmission and relay operation, as well as for rechecking the calibrations of the flowmeters themselves.

Fig. 14 Totalizing Relays and Station Total Discharge Counter Installed on Relay Board in Control Room

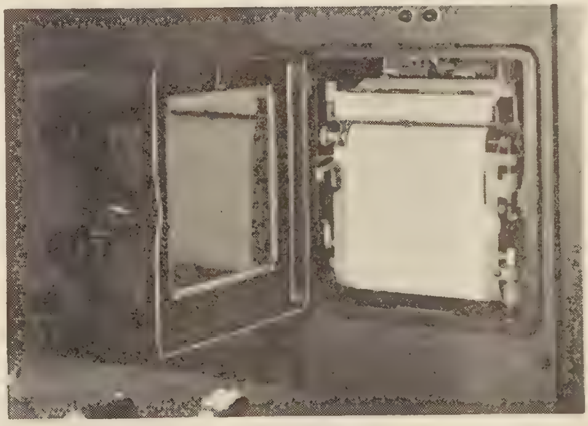


Fig. 15 Station Total Discharge Indicator and Recorder Installed on Instrument Board in Control Room

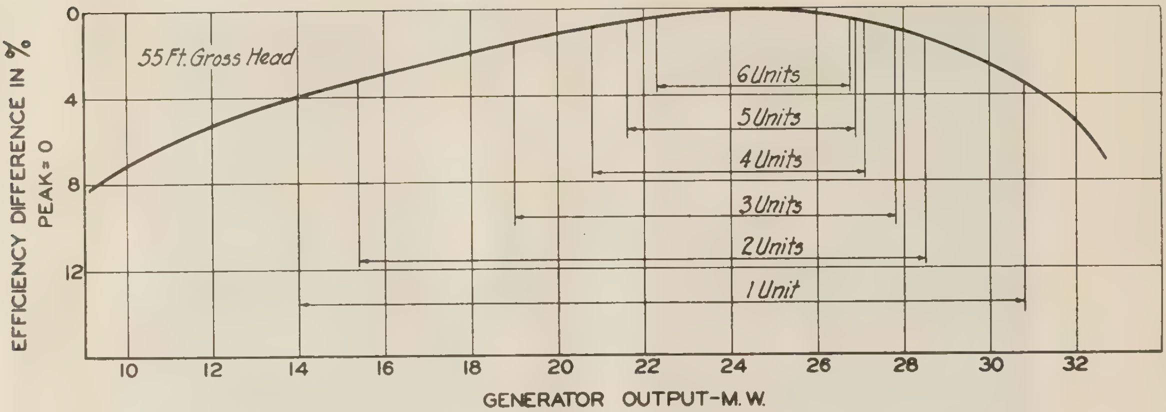


Fig. 16 Typical Loading Schedule for Safe Harbor Main Units

A total station discharge graphic recorder and discharge indicator, combined in one instrument, was installed on the instrument panel located opposite the totalizing-relay panel, Fig. 15. The upper range limit of the graphic recorder and the indicator was chosen as 80,000 cfs, representing the approximate maximum station draft of the Safe Harbor development for a number of years in the future.

3—BENEFITS OBTAINED THROUGH FLOWMETER INSTALLATION

During 1939, various investigations were made based on the data obtained by the flowmeter installation. The operators were required to read the individual unit discharges and the station-total draft every hour on the hour, together with the unit and station integrating watthour meters, as well as forebay and tailwater indications. The operators, however, were still charged with computing the individual unit and total-station drafts based on power output, as had been standard practice. It was felt that a long-term comparison was essential to obtain the proper basis for continuous-flow records at Safe Harbor, the transition period furnishing the ratio between computed and automatically recorded station drafts under the various seasonal loading schedules. Once sufficient data have been accumulated, it is expected that the operators will be relieved altogether from computing the discharge based on output.

The data obtained by the operators were also used to investigate unit and station operating efficiencies. An investigation of this kind was all the more essential, as during approximately 290 days of the year the available river flow at Safe Harbor is less than the station draft required with all six main and two service units installed operating at maximum capacity. After placing the seventh main unit in service, which is now under construction, the corresponding period will increase to 305 days. It was interesting to note that the availability of an input yardstick had a decidedly stimulating effect on the operating personnel. While, during the first month of flowmeter operation, that is, January, 1939, the ratio of actual loss in generation to expected loss was greater than unity on all but 8 days, this ratio did not exceed unity during 18 days in May, 1939, under similar river-flow conditions and loading schedules. It has been estimated that an improvement of this magnitude is responsible for an increase in generation of at least 0.3 per cent, or approximately 1,700,000 kwhr per year with the present installation of six main units and 1,900,000 kwhr per year with the seventh main unit placed in service, so that the flowmeter installation will pay for itself in a very short time.

With a close continuous check on unit operating efficiencies available, it was also possible to keep the losses due to trash on the intake screens appreciably below those which must have been prevailing during previous years. Prior to the installation of the flowmeters, the screen losses were determined from time to time by measuring the screen head loss. Now, as soon as any of the units show a drop in operating efficiency, as indicated by the hourly readings, the screen losses are determined independently. After cleaning the racks, the operating efficiency invariably increases to the expected level. Although it is difficult to estimate the increase in station economy due to this means of obtaining an earlier indication of the loss in efficiency due to plugging up of the screens, nevertheless, it is believed that the benefits thus derived are substantial.

Since the availability of the particular type of flowmeters permitted short-duration turbine-efficiency tests to be carried out by one man, therefore justifying itself as a routine measure, it was also possible to investigate in detail the efficiency characteristics of each main turbine over the entire range of operating heads. Such a procedure was particularly desirable as these turbines are of the Kaplan type, requiring an adjustment of the cam controlling the gate-blade relation for the various operating heads, the operators being required to change to a new cam setting after each 1 ft of change in head. While these compensating devices on all main units were originally designed and calibrated, based on a minimum of information due to the costly testing procedures even with the use of the index method with piezometers, the flowmeters available made it possible to check and recalibrate these compensating devices with a large amount of detailed information obtained with a minimum of effort. The scope of the work involved can best be realized by mentioning that, although five of the six main units were of identical design, nevertheless, the characteristic of each unit was found to be sufficiently different from the others to warrant individual cams, consequently requiring individual calibration of the cam-adjustment device for variation in head. The results of this investigation reflected favorably upon the operating efficiencies of the individual units and the station as a whole.

As a next step, a detailed study was undertaken to determine the magnitude and duration of avoidable inefficient operation in percentage of total operating time. From the ideal loading schedule for the main units, as shown in Fig. 16, and valid for a gross head of 55 ft, it is apparent that the band of permissible load variations of each unit decreases with increasing number of units on the line. With capacity requirements above the most efficient station operating range with all available units operating,

all individual unit loadings are increased by equal amounts up to the point of maximum capacity.

By analyzing the chart of the total station discharge recorder, Fig. 15, in the light of the operator's log, it was noted that considerable periods elapsed between the time of placing units on or off the line and the ideal loading schedule. The losses thus sustained, though by no means excessive, when compared with some other stations were nevertheless appreciable, amounting to about 10 per cent of the total operating time on the average.[6,7] It was realized that there was considerable room for improvement provided proper means were available for giving instantaneous warning when reapportioning of load to individual units is required. It is obvious that in this connection some thought was again given to efficiency-indicating-and-recording apparatus, but another and far simpler and less expensive solution was discovered.

4—INSTALLATION OF LOAD-LIMIT LIGHTS

The characteristics of the Kaplan-type main turbines installed at Safe Harbor are such that the most efficient discharge range of these units is, for all practical purposes, independent of the head if the loading schedules, valid for each head similar to that in Fig. 16, are adhered to. Thus the discharge, with one unit operating within the permissible load range, varies between 4000 and 8200 cfs irrespective of the head, and the discharge ranges

TABLE 3 DISCHARGE LIMIT SETTING FOR LOAD LIMIT LIGHTS

Discharge range, no.	Discharge-range setting, cfs	Main units to be operated, no.
1	0– 4000	0
2	4000– 8200	1
3	8200–14900	2
4	14900–21400	3
5	21400–27500	4
6	27500–33900	5
7	33900–40800	6
8	40800–48000	7

with any given number of units operating are also constant, i.e., independent of the head for all practical purposes. In view of these characteristics and taking proper account of station-service unit draft requirements, it was possible to provide for an automatic and instantaneous load-limit indicating apparatus as an integral part of the total station discharge indicator and recorder, shown in Fig. 15.

Essentially, this device consists of a contact-making cam arrangement controlling two warning lights, one located on the operator's desk and the other on the instrument board above the station total discharge recorder. For each load range between two discharge limits, Table 3, there is available one contact-making cam assembly independently adjustable as to what part of the total-discharge range it will control. A control switch is provided with one position for each discharge range, that is, number of units to be operated, connected so as to keep the light extinguished when set for the number of units to be in operation for best efficiency, as long as the discharge is in the corresponding range. If the discharge crosses the limits of this range, the lights will be lighted from the contact assembly of the adjacent range either until the switch has been reset to the number of units, corresponding to this new range, or the discharge has returned within the range. With the control switch being kept set correctly, that is, corresponding to the number of units in operation for best efficiency in each discharge range, the illumination of the lights will indicate inefficient operation.

[6] "How We Raise Hydro Efficiencies," by E. B. Strowger, *Electrical World*, vol. 103, April 14, 1934, pp. 535–538.

[7] "Waterwheel Testing and Operating Records of Plant Discharges," Proceedings National Electric Light Association, vol. 85, 1928, pp. 872–904.

It may be noted that eight discharge ranges have been provided, the reason being that the seventh main unit is now being installed and that an indication is also desirable when the station as a whole, with all seven main units operating, has reached the upper limit of the range of most efficient operation.

In addition, the scope of the total discharge station indicator and recording instrument will be increased by means of adding a load-operating-range scale, each division of this scale corresponding to the permissible range of discharge for a certain number of units in operation as shown in Table 3. By means of this improvement, it will be possible to observe at a glance how many units should be in operation at any time. When reaching a load limit as indicated by the warning lights and observing the shape of the discharge curve plotted by the station-discharge recorder, it also will be immediately apparent whether an upper or lower limit has been reached, requiring one unit to be put on or off the line, respectively.

To keep a definite record of inefficient operation, the station total discharge recorder is also to be equipped with an additional pen element operating simultaneously with the load-limit lights. This added provision will also enable the operators to ascertain the duration of the period of inefficient operation prior to noticing the lighted load-limit lamps, so that the allowable 10-min interval of borderline operation is not exceeded. Some inefficient operating time is necessarily unavoidable and, for the present and some time past, we have felt that a 10-min period of allowable inefficient operating time is reasonable.

The installation of this load-limit light apparatus is now in progress and it is expected that, due to its availability, avoidable inefficient operation will be reduced to a negligible amount, resulting in an additional and substantial increase in operating efficiency and station output.

Discussion

M. M. Borden.[8] The type 2 flowmeter referred to was not constructed with certain precise operations, involving gear-centering and tooth-spacing in particular, which are applied to instruments the totalized flows of which are to be read at intervals of a few minutes rather than several times a day.

A case in point is taken from the record of one of several such meters, which were furnished for an electric power station. In this instance the maximum errors of the totalizer when read at 5-min intervals varied from —0.8 to +0.9 per cent with an average for a 90-min period of +0.16 per cent.

For 10-min intervals between readings, such point errors varied from +0.65 to —0.65 per cent and the 90-min average was +0.16 per cent.

For 30-min intervals between readings, the point errors were from a maximum of +0.3 to —0.1 per cent, with a 90-min average of +0.1 per cent.

The errors were determined by comparing the readings of the fast-moving hand of the totalizer with its 4-in. graduated circle with the record of the water weighed in the laboratory tanks.

The type 2 instrument permitted comparison of the instantaneous rate of water flow with the corresponding instantaneous indications of the electrical output and of the head on the wheels.

The water-flow rate indication of this meter is made without the use of gearing and may be read by a pointer moving in front of an equally spaced flow scale of whatever radius is required.

While the W-K relationship appears to have a normal flow of 0.5 for n, the operating principle of the type 2 meter allows it to be furnished with a uniformly spaced flow scale for any value of n which the particular field rating might necessitate.

[8] Chief Engineer, Simplex Valve & Meter Company, Philadelphia, Pa. Mem. A.S.M.E.

E. S. Bristol.[9] This paper presents an interesting review of the steps taken in a persistent investigation that finally resulted in a flow-measuring installation of a rather unusual nature. It is of interest also to note how the various obstacles were overcome through careful study and how the information yielded by the final metering system was analyzed to obtain improved station performance.

Additional information with respect to the type 3 flowmeter described by the author, will make more apparent the characteristics contributing to the degree of accuracy reported. Referring to Fig. 7 of the paper, it is seen that the flowmeter is a force balance in which a force dependent upon piezometer pressure difference is opposed to centrifugal force from a rotating-flyball system. The meter is a relay-type mechanism, in which the balance arm functions only as a detector to regulate electric-power supply to the integrator motor. A knife-edge support is provided for the balance arm which is not required to operate any indicating, recording, or integrating elements, but which merely functions to actuate a magnetically operated mercury switch. The alternate closing and opening of the mercury switch results in an on-off control of the integrator motor, such that its speed oscillates slightly above and below the required average value for any particular pressure differential. The tilting manometer thus has a continuous rocking action, similar to that of many speed governors, which reduces to a minimum any tendency of the mercury to stick to the manometer tubes as well as any frictional effects.

The flyball system, actuated by the integrator motor, is of the neutral type, such that force transmitted to the manometer arm is independent of the flyball angular position over the working range. This characteristic avoids change in calibration when the manometer arm assumes slightly different average positions, as required to change the on-off time cycle of the mercury switch in maintaining required motor speed, despite variations in voltage, frequency, etc.

The integrating element of this type of meter inherently possesses the same accuracy as the meter itself, since direct coupling of the integrator to the variable-speed motor, driving the flyball system, avoids the introduction of any intermediate errors.

Separate means of adjustment are provided for calibrating the high and low ends of the meter range, thus making it possible to match closely the characteristics of the primary-flow element, such as the turbine-scroll piezometers employed at Safe Harbor, or the venturi tube, flow nozzle, or thin-plate orifice more commonly used. The high-range adjustment consists of a threaded rod for changing the point of attachment of the vertical flyball link to the horizontal manometer arm. The low-range adjustment consists of a moving balance weight on the manometer arm. By means of these adjustments, the meter calibration can be readily changed in the field to suit an experimentally determined coefficient of the primary element, in applications where facilities are available for checking the latter in its service location. Figs. 9 and 12 of the paper indicate the nature of the variations which can be made in the meter calibration. The experimental meter of Fig. 9 was slow at the higher flows, so that the manometer balance weight was offset in the increase direction to improve the over-all relation. The three calibration curves of production meters, in Fig. 12, show much improved settings at high flows, with both high and low deviations at low flow, depending upon the particular low-range-adjustment setting. As pointed out by the author, once the relation between water-column readings and check-weight readings has been determined, the latter can be used in routine accuracy checks with resultant saving in maintenance time.

The relay equipment for totalizing station-water flow is of the standard impulse type employed for electrical-demand metering, with minor modifications to suit the high rate of operation required. Flow is totalized every 2 min, so that a high rate of impulses per minute is necessary in order to obtain reasonably close setting of the totalizing recorder. The pen of this recorder moves at the expiration of each 2-min interval to a position corresponding to average rate of station-water flow during that interval. The totalizing action employs positive forward and return electrical impulses, with corresponding forward and return solenoids on the totalizing relays. As a result, no false counts occur if a transmitting contact chatters and produces more than one impulse in the same direction. After each forward impulse, the associated return impulse must go through to reset the receiving element, before a successive forward impulse can be of any effect.

The author refers to the possible use of the flowmeter equipment as a component of efficiency-measuring equipment for individual generating units or for the complete station. To obtain an indication or record of efficiency, elements must be added which will properly combine effects representative of electrical output and hydraulic head with the water-flow measurement and provide an ultimate indication of the ratio of electrical output to the product of water flow multiplied by head. These operations can be performed electrically, using an emf from a thermal converter or torque balance to represent electrical load, an emf from a slide-wire, positioned in accordance with rate of flow, and an emf proportional to head, as derived from float-actuated slide-wires at the forebay and tailrace. By applying these emf values to suitable potentiometer recording equipment, a continuous record can be obtained of the efficiency of an individual unit or of the entire station.

In closing, it may be mentioned that the type 3 flowmeter is not restricted to hydraulic applications, but is also employed in steam-flow service.

F. Nagler.[10] The water-power industry has been all too slow in analyzing its own performance. Its system seems to be much less exact than that of a flour mill, a country grocery store, or a gold mine. All too frequently, however, because of the difficulty of sampling the ore, mining operations are tabulated on the basis of adding the bullion produced to the assumed or measured gold content in the tailings, that sum being reported as the "head." Water-power management has not been so greatly different in the conduct of its own affairs.

The ideal state would be to charge to the plant the flow in the river and credit to the plant the kilowatthours produced. The apparatus methods described in the paper are, apparently, sufficiently directed to that very end.

Is it not inevitable that any piezometer located on the nose of the vane will be more erratic, under variable-flow conditions, than one located on a surface where the flow is directed? This does not refer so much to the variation of flow from the operation of the guide vanes, as to variations resulting from influences further upstream. Typical sources would be the condition of the racks but, more particularly, the condition of operation of adjacent units. In any event, careful observation of the indicated results should permit attention to be called quickly to any abnormal flow condition which might be harmful.

Apparently, Fig. 16 of the paper tells quite a story, not so much for a plant containing six units, but particularly for plants which contain more. Flatness of the efficiency-gate-opening curve naturally plays less and less part in efficient operation of plants as the number of units increases. Should this be applied still further to the regulation of units, a series of curves would

[9] Engineering Department, Leeds & Northrup Company, Philadelphia, Pa. Mem. A.S.M.E.

[10] Chief Engineer, Canadian Allis-Chalmers, Ltd., Toronto, Canada. Life Member A.S.M.E.

result very much as shown in Fig. 7 of a paper[11] by the writer on speed regulation.

J. F. Roberts.[12] This paper should be of great interest to engineers who have tried to keep accurate discharge records at hydroelectric plants. Apparently, the author's organization has been successful in obtaining the cooperation of the operating engineers. In his earlier experience the writer frequently encountered opposition or at least lack of interest in this regard. Since turbine flowmeters invariably show a greater discharge, as compared with computed discharges based on kilowatthour output, the operators sometimes preferred the latter method, as it gave them credit for a higher operating efficiency than actually existed.

The desirability of turbine flowmeters is now universally recognized by operators as essential in large modern hydroelectric plants. How many modern steam plants are built at the present time without accurate coal, feedwater, and steam flowmeters? Electrical engineers would not think of omitting both integrating and recording watthour meters, yet some of these same engineers formerly belittled the use of turbine-discharge flowmeters, preferring to rely on the unit-performance curves made up when the units are new and under ideal test conditions.

The Tennessee Valley Authority has had excellent results with the Winter-Kennedy type of taps shown in the author's Fig. 1. One set of these taps was calibrated on a 16-in. test model of a 45,000-hp 48-ft head, fixed-blade propeller turbine, obtaining the following calibration: $Q = 6562\ D^{0.481}$ where D is the deflection in feet of water, the quantity of water being measured by a weir.

Gibson tests on similar taps on a 66,000-hp 165-ft head Francis turbine gave the following equation for two similar units:

$$\text{Unit 1}\begin{cases} Q = 1693.4\ D^{0.5026} & \text{for } R_6 - R_3 \\ Q = 1367.7\ D^{0.5087} & \text{for } R_6 - R_1 \end{cases}$$

$$\text{Unit 2}\begin{cases} Q = 1659.3\ D^{0.5103} & \text{for } R_6 - R_3 \\ Q = 1325.2\ D^{0.5088} & \text{for } R_6 - R_1 \end{cases}$$

where D is the deflection in inches of mercury.

J. W. Scoville.[13] The stay vanes of a speed ring are a necessary evil, and their angle and shape have been objects of considerable investigation in so far as turbine efficiency is affected. The angle is necessarily a compromise, since a turbine has to operate at any gate opening. The angle and shape are such that the best efficiency is not reduced nor maximum output affected adversely. Necessarily no consideration has been given to the effect on the Peck piezometers. The writer has noticed that the coefficient for the Peck taps varies with gate opening on Kaplan turbines in several plants which have been tested. This fact does not necessarily preclude their use for index testing in connection with the determination in the field of the proper blade-gate relationship of a Kaplan turbine. The Winter-Kennedy system is equally suitable for this purpose.

The author mentions but does not stress the fact that the piezometers were used for such index testing. It is possible by such methods to obtain the correct blade-gate relationship of a Kaplan turbine without going to the expense of a water-measurement test.

As the author points out, if the deflection between Peck or Winter-Kennedy taps is measured, Q is equal to $C \times D^a$ where a is practically 0.5. If a test is made at various blade angles at several gate openings, during which the head, kilowatt output, and deflection are measured, curves can be plotted as shown in Fig. 17 of this discussion.

The ordinates $KW\sqrt{D}$ are proportional to the unit efficiency if plotted for a constant head. In effect, such a curve is an over-

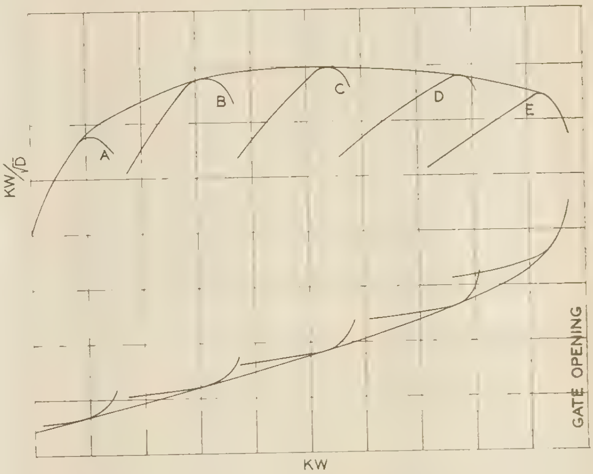


Fig. 17 Curves Showing Relation Between Head, Kilowatt Output, and Deflection, Resulting From Tests Made at Various Blade Angles and Several Gate Openings

all power-efficiency curve at an unknown scale. The tangent point of the envelope to the individual curves at the several blade angles A to E determines the proper gate opening at these blade settings. If, as the author suggests, the peak efficiency is estimated, the efficiency curve is then determined, as well as all necessary data for the proper operation of the unit. Thus, the advantages of a field test may be obtained on a unit where it is impossible to make an accurate water measurement. Such an index test may be made on any plant at a saving in cost over a more extensive one in which a water measurement is made.

E. B. Strowger.[14] The author shows that at Safe Harbor the exponent a in the equation representing the Winter-Kennedy deflection-discharge relation was determined by experiment to be 0.5, this equation being $Q = C \times D^a$. With a equal to 0.5 it is apparent that, in the case of the Safe Harbor units, the force due to the piezometer differential varies with the square of the flow. Since centrifugal force varies as the square of the speed, the author was able to utilize an integrator carrying a flyball system, so arranged that the centrifugal force due to rotation of the integrator is opposed to the force of a tilting mercury manometer, resulting in a direct linear relationship between flow and integrator speed. If the value of the exponent had been found to be other than 0.5, the relationship between flow and integrator speed would not be linear and the integration of the flow would have been more complicated. Possibly in this case a slight correction in the flyball system could be made to produce the desired direct relationship.

Table 4 of this discussion is presented to show a number of Winter-Kennedy tap calibrations which have been made by the Gibson method of testing on 50 units in 12 different hydroelectric power plants. The exponent a for these taps is shown to vary from a minimum of 0.476 to a maximum of 0.538 and the arith-

[11] "Changing Requirements in Hydraulic Turbine Speed Regulation," by F. Nagler, Trans. A.S.M.E., vol. 52, 1930, HYD-52-2.

[12] Principal Mechanical Engineer, Tennessee Valley Authority, Knoxville, Tenn. Mem. A.S.M.E.

[13] Assistant Chief Engineer, S. Morgan Smith & Company, York, Pa. Mem. A.S.M.E.

[14] Hydraulic Engineer, Buffalo, Niagara and Eastern Power Corporation, Buffalo, N. Y. Mem. A.S.M.E.

TABLE 4 RATINGS OF CERTAIN WINTER-KENNEDY TAPS BY THE GIBSON METHOD

$(Q = C \times D^a)$

Plant	Unit	Date of Test	No. of Test Runs	Taps Used	C	a
1	1	6/18/31	26	-	865.9	0.529
2	4	9/10/31	45	-	1923.3	.505
3	1	9/25/30	45	-	949.0	.503
4	1	4/21/32	29	-	34.1	.500
4	2	4/22/32	29	-	34.1	.500
4	3	4/23/32	29	-	34.1	.500
5	2	3/15/33	30	(R4-R1)	2759.9	.508
5	2	3/15/33	30	(R4-R2)	3313.1	.508
5	9	3/11/33	30	(R4-R1)	2644.3	.518
5	9	3/11/33	30	(R4-R2)	2989.8	.518
5	A	3/13/33	31	(R4-R0)	557.9	.521
5	A	3/13/33	31	(R4-R1)	599.5	.521
5	11	8/16/34	32	-	2734.0	.513
6	1	8/15/33	26	(R4-R2)	646.9	.535
6	1	8/15/33	26	(R4-R3)	739.1	.535
6	2	8/16/33	25	(R4-R2)	650.1	.535
6	2	8/16/33	25	(R4-R3)	754.9	.535
6	3	8/17/33	25	(R4-R2)	668.1	.538
6	3	8/17/33	25	(R4-R3)	740.3	.538
6	4	8/18/33	25	(R4-R2)	647.8	.538
6	4	8/18/33	25	(R4-R3)	753.1	.538
7	5	9/18/33	42	(Note 1)	2728.9	.496
7	2	10/22/35	47	(Note 2)	1212.9	.521
8	1	4/16/37	30	(R2-R1)	977.5	.508
8	1	4/16/37	30	(R3-R1)	835.7	.508
8	1	4/16/37	30	(R4-R1)	686.0	.508
8	2	4/19/37	31	(R2-R1)	1012.2	.503
8	2	4/19/37	31	(R3-R1)	893.0	.503
8	2	4/19/37	31	(R4-R1)	710.9	.503
8	3	4/21/37	30	(R2-R1)	1029.6	.495
8	3	4/21/37	30	(R3-R1)	870.9	.495
8	3	4/21/37	30	(R4-R1)	732.8	.495
8	4	4/23/37	22	(R2-R1)	1035.5	.505
8	4	4/23/37	22	(R3-R1)	894.5	.505
8	4	4/23/37	22	(R4-R1)	721.2	.505
9	3	10/12/37	35	(R5-R2)	817.1	.508
9	3	10/12/37	35	(R5-R4)	941.8	.515
9	8	10/15/37	35	-	354.1	.508
9	8	10/15/37	35	-	436.7	.476
9	5	10/11/38	21	(R6-R4)	873.1	.505
9	5	10/11/38	21	(R6-R3)	801.8	.508
9	5	10/11/38	21	(R6-R2)	724.6	.505
10	2	10/20/37	32	(R6-R1)	1310.5	.508
10	2	10/20/37	32	(R6-R3)	1653.7	.510
10	1	10/25/37	32	(R6-R1)	1354.6	.508
10	1	10/25/37	32	(R6-R3)	1704.8	.503
11	2	11/19/37	28	-	2653.7	.500
11	3	11/30/37	28	-	2614.2	.505
11	1	12/1/37	28	-	2628.4	.503
12	4	1/28/38	37	-	1709.7	.481

Note 1. Sum of three deflections. (R4-R2) + (X2-R2) + (X1-R2)

Note 2. Function of four deflections. 2(X1 + X2 + R4 + Y1 - 2R1 - 2R2)

metic average of all values shown is 0.511. While the theoretical value of the exponent is probably 0.5, in many cases the character of the flow in the vicinity of the taps or the condition of the tap equipment may be such as to cause the value of the exponent to depart slightly from the theoretical value. Attention is particularly called to the values of a for unit 8 of plant No. 9 where the test on one set of taps showed a value of 0.508 and the same test on another set showed a value of 0.476.

I. A. WINTER.[15] The writer has had occasion to check the performance of type 2 and type 3 flowmeters, and finds that both instruments are capable of a high degree of accuracy and reliability with, apparently, the advantages of integration slightly in favor of the type 3 meter and the advantages of indication and servicing slightly in favor of the type 2 meter. That part of the paper which the writer is best qualified to discuss is the performance of the prime mover or the differential pressure taps, located on opposite sides of the turbine scroll case, of which considerable data have been accumulated.[16]

The Winter-Kennedy piezometer system, shown in Fig. 1 of the paper, depends upon the effect of centrifugal force of the water as it flows about the vertical axis of the unit, and therefore registers as a function of the flow past the piezometer section only and is not affected by the coefficient of friction of the walls of the conduit, the angle of the turbine gates, or the head on the power plant. Whenever possible, pertinent data relating to the performance of the taps, with respect to these factors, have been obtained and it may be said that the results have been highly satisfactory.

An example of the comparison of performance of the prime mover and the type 2 meter is illustrated in Fig. 18 and Table 5

[15] Senior Engineer, United States Bureau of Reclamation, Denver, Colo.

[16] Ref. (2) of paper.

TABLE 5 PERFORMANCE OF DIFFERENTIAL-PRESSURE TAPS AND FLOWMETER UNDER VARYING HEAD

Servo-motor-piston stroke, in.	d In. mercury at 524-ft head	d Reduced to 453-ft head by Equation [12][16]	d From curve for 453-ft head	Q At 524-ft head determined by manometer	Q By flow-meter dial, type 2	Departure, per cent
0.00	0.00	..	..	...	...	...
2.17	0.55	0.48	0.46	330	335	+1.5
3.08	1.19	1.03	0.99	485	480	—1.7
3.97	2.15	1.86	1.85	640	636	—0.7
4.86	3.36	2.91	2.91	775	790	+1.8
5.77	5.08	4.40	4.40	935	940	+0.5
6.67	6.56	5.68	5.83	1070	1082	+1.2
7.33	7.85	6.79	6.98	1165	1175	+0.8
8.02	9.15	7.92	8.23	1255		
8.87	10.68	9.24	9.60	1350		
8.45	10.20	8.82	8.95	...		
7.56	8.36	7.23	7.41	..		
6.67	6.73	5.82	5.83	1080		
5.98	5.40	4.67	4.67	970		
5.31	4.26	3.69	3.69	865		
4.42	2.75	2.38	2.38	710		
3.53	1.74	1.51	1.35	580		
2.26	0.88	0.76	0.50	425		
0.00	0.00	..	..	...		

No readings taken

NOTE: The original calibration of the flowmeter taps was made October 15, 1937, at a gross head on the plant of 453 ft. The meter register was calibrated to agree with these data. The check test by the index method was made on December 27, 1939, and the head on the plant at that time had increased to 524 ft.

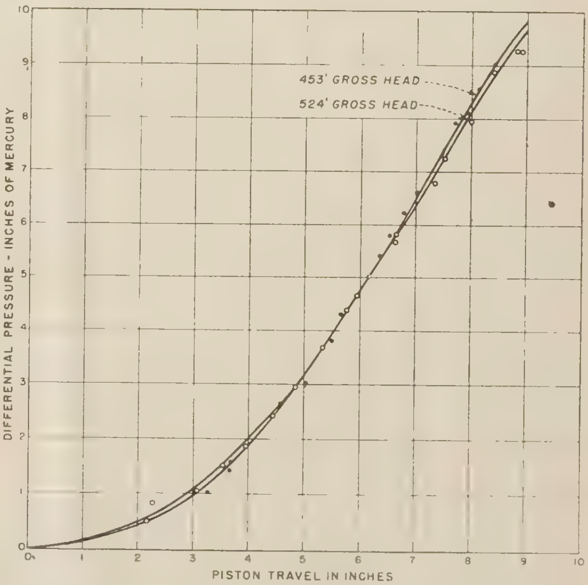


FIG. 18 RELATION OF TURBINE-SERVOMOTOR-PISTON TRAVEL TO DIFFERENTIAL PRESSURE CORRECTED TO A COMMON HEAD OF 453 FT

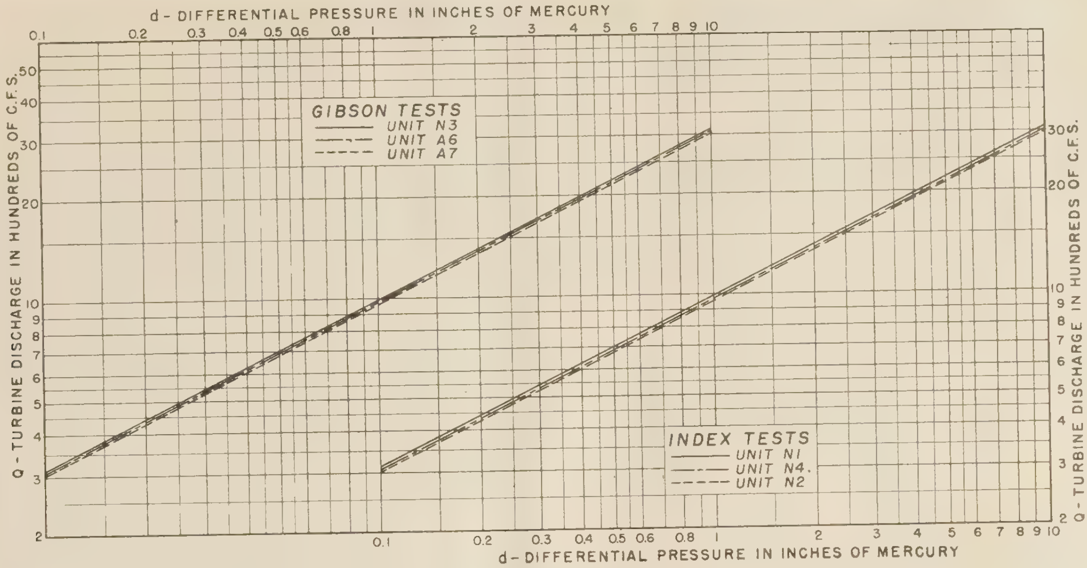


FIG. 19 COMPARISON OF FLOWMETER PERFORMANCE FOR IDENTICAL TURBINE-SCROLL CASES

TABLE 6 FLOWMETER CONSTANTS AND DEVIATIONS FOR FIG. 19

Unit	Constant c in equation $Q = cd^{0.5}$	Deviation from average c = 972 for 6 units, per cent
N1	988	+1.6
N2	953	—2.0
N3	986	+1.4
N4	968	—0.4
A6	976	+0.4
A7	962	—1.0

of this discussion, showing the relation of turbine-servomotor-piston travel to differential deflection in inches of mercury, as observed on differential-pressure taps, installed in unit A-8 at the Boulder power plant. The curve for gross head of 453 ft was determined by Mr. Gibson, using the time-pressure method of water measurement. The curve, designated as 535-ft gross head, represents comparable measurements when reduced to a common head of 453 ft in accordance with Equations [11] and [12],[16] stating that the deflection readings may be reduced to a common head directly as the ratio of the common to test heads or H_c to H_t. This curve shows very good agreement between the original calibration made in 1937 and the check test made in 1939. The maximum deviation of these curves is about 2 per cent, which is to be expected, since there is a change in the coefficient of discharge of the turbine, due to the constant speed of the runner under varying heads and, also, there is a change in the coefficient of discharge of the turbines in the opposite direction, due to the roughness of the turbine orifices increasing in the two-year period between tests, thereby lowering the coefficient of discharge. The index test at 535-ft head included readings of the flowmeter dial, for comparison with the observed quantities determined by the manometer readings shown in Table 5. This table shows the maximum deviation between the flowmeter dial and the manometer reading to be plus $1^1/_2$ per cent and the average throughout the range of the curve is less than 1 per cent. This would appear to be a satisfactory performance of the meter after two years of continuous operation and without special adjustment for the tests.

Fig. 19 of this discussion is of special interest in comparison of the discharge-differential-pressure relation as obtained on units of the same design. This figure shows results of calibration made on six units with identical scroll cases at the Boulder power plant, each developing in excess of 115,000 hp, when operating at a head of 475 ft or higher. Three sets of flowmeter taps were calibrated by the time-pressure method of water measurement (Gibson tests), and three sets were calibrated by the index method, using one unit tested by Mr. Gibson as a basis and assuming the other units to have the same coefficient of discharge.

It is not to be expected that the performance of the taps for similar units will be in better agreement than the data shown in Fig. 19, due to the lack of exact cross-sectional area of the casings at the metering section, and the lack of similarity of the runner and turbine gates which affects the coefficient of discharge at different points around the turbine speed ring. It is also likely that the coefficient of discharge may vary as much as 5 per cent for similar turbines, due to differences in orifice areas in the runner, inherent with the difficulties of producing large steel castings. The change in discharge of the unit due to a change in the controlling-turbine areas may be accounted for by making precise calibrations of the runner, in order to apply the proper correction factor because of the lack of similarity when extrapolating test data from similar units.

A study of accumulated data for various tests of turbine-flowmeter installations indicates that exponent a of 0.5 gives more consistent results than exponents determined by other flow means. This is in agreement with the author's conclusion, based upon the study of results obtained by means of precise current-meter measurements. With these data as precedent, it is recommended that exponent a, used in the flowmeter calibrations, be accepted as 0.5 and the constant for each individual run be determined on this basis with the general equation for the entire range of flow determined by means of weighted averages so that the greater degree of accuracy obtained for the higher water measurements may be given full weight in the final equation. The data given in Fig. 19 are determined upon this basis.

AUTHOR'S CLOSURE

In making more pertinent data available, the various contributions to the discussion constitute a valuable addition to the paper. Mr. Borden's comments confirm the conclusions regarding the type 2 meter, as the amplitudes of the sinusoidal-error curves for the two meters of this type investigated are of the same magnitude as for the meter mentioned by Mr. Borden.

While the amplitude for one of the type 2 meters was found to be (0.63 + 0.38) or 1.01 per cent, that of the other was (1.39 + 0.82) or 2.21 per cent. Based on the 90-min test period and readings at 5-sec intervals, Mr. Borden's meter, of the same type picked at random, apparently has an error amplitude of at least (0.8 + 0.9) or 1.7 per cent.

Regarding Mr. Bristol's contribution, it may be pointed out that the type 1 meter may also lend itself to efficiency-indicating-and-recording apparatus, as the electrical impulses obtained from the mercoid switches limiting the swing of the drum, Fig. 20 of this closure, could be used for the positioning of slide-wire elements. At the same time, it should not be lost sight of that

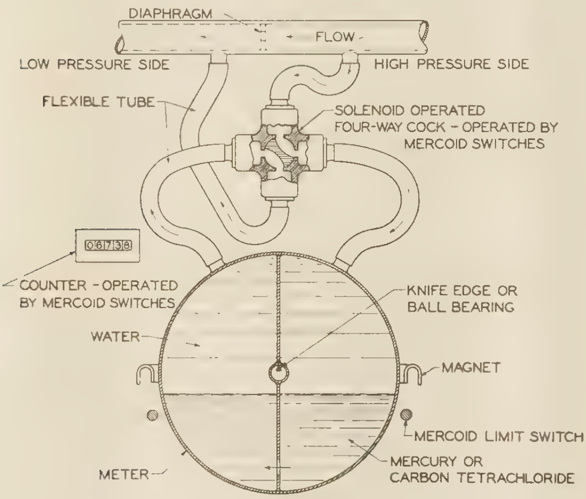


Fig. 20 Schematic View of Type 1 Meter

load-limit indications, based upon hydraulic input or efficiency-indicating-and-recording apparatus, are not the only two alternatives to improve operating efficiencies. Alarm devices, similar in principle, may be based upon power output rather than upon hydraulic input and, if need be, with manual or automatic adjustment for head. Still other alarm devices may take advantage of gate opening using the stroke of the servomotor piston of the gate mechanism as a primary element.

Mr. Nagler touches upon a subject of wide interest. For many years there have been two schools of thought. One of these advocates that only vital machinery together with the very minimum of auxiliary equipment be installed with the intent of simplifying operation and keeping the investment as small as possible without impeding service. Those adhering to this point of view usually claim that many of the more modern power stations are over-equipped with auxiliary instruments and apparatus of all sorts, complicating operation unnecessarily. The second school and no doubt the more progressive one believes that the omission of certain apparatus is false economy and that auxiliary instruments are justified, provided they either pay their own way through increased generation or by making data available which may ultimately lead to improved service through furnishing the basis for performance investigations and thus contribute to an advancement in the art. It is thought, however, that no single policy may be advocated as, depending upon local conditions, either course may be well justified.

Mr. Roberts very aptly points out that the metered turbine discharge will be invariably larger than the discharge determined indirectly, based upon output. Since this difference is inherent with the convex shape of unit-efficiency curves, it seems rather puzzling why any operator should object to metering.

Regarding Mr. Scoville's discussion, it is believed that the best speed-ring design is very likely also the most favorable to the installation of Peck piezometers. The most advantageous shape of the stay vanes necessarily is such that it produces no vortexes which would adversely affect the turbine efficiency. It is under these ideal conditions, free of vortexes, that piezometers, in this case those of the Peck type, are most reliable.

Index testing was not particularly stressed as it had been discussed in detail in a previous paper[4] to which reference was made in this paper. However, a calibration of the piezometers by means of a quantitative water-gaging method, suitable to local conditions, is thought to be essential for a complete field test. Exclusive reliance on a step-up formula from laboratory results is as yet not feasible if a high degree of accuracy is desired for input data obtained by means of piezometers. With duplicate units of identical manufacture, the piezometer calibrations of at least one unit should be obtained by means of absolute water measurements, with the taps of the other units to be calibrated indirectly by assuming the efficiencies of both to be alike.

The question raised by Mr. Strowger for a possible adjustment on the type 3 meter for exponents differing by a small amount from 0.5 has been answered by Mr. Bristol. An adjustment is possible on the flyball system.

The statement made by Mr. Winter, regarding the performance of type 2 and type 3 flowmeters is interesting, particularly so because our experience has been different in that the type 3 meter was found to be not only more accurate but, at the same time, needed considerably less servicing than the type 2 meter.

In view of the conclusions reached with respect to the magnitude of the exponent, the additional piezometer-calibration data made available by Messrs. Roberts, Strowger, and Winter are of utmost importance. It is believed, however, that equal weight should not be given to each individual piezometer calibration cited. While some of the calibrations were obtained by means of graphic procedures, others were arrived at by means of analytical computations, using the method of least squares. Only the latter method is thought to be sufficiently accurate to warrant serious consideration, if a high degree of accuracy is desired. At the same time, the calibration method employed must be given some weight, together with the number of test points, magnitude of test-point dispersion inherent in the various absolute water-measuring methods, as well as local conditions. In this connection, attention may be drawn to the remarkable consistency of the results obtained with the water-gaging method employed at Safe Harbor, as it compares most favorably with that of any other method known at this time. Whether or not the exponent is actually 0.5 or a value very close to it may depend to a certain extent upon local conditions but there seems to be an agreement that for practical purposes an exponent of 0.5 may be entirely adequate in most cases from a metering point of view.

Speed Regulation of Kaplan Turbines

By J. D. SCOVILLE,[1] YORK, PA.

In this paper hydraulic-turbine speed-regulation data are presented and compared with calculated performance. Field tests made at the Bonneville and Guntersville power plants are cited. The effect of air admission during load rejection is discussed. Runaway-speed tests in the field are compared with model data.

THE problem of speed regulation has been the subject of numerous publications, although but meager field-test data have been presented for comparison with theoretical performance. It is the purpose of this paper to present the results of tests made on the adjustable-blade or Kaplan turbines at the Bonneville and the Guntersville hydroelectric plants, two recent power projects developed, respectively, by the United States War Department and by the Tennessee Valley Authority.

Kaplan turbines require special consideration in the matter of speed-regulation studies for three reasons:

1 The blades of the runner change pitch simultaneously with the movement of the gates for change of load.

2 The runner must frequently be placed below tail water to prevent cavitation.

3 The runaway speed is higher than that of other types of turbines.

Obviously, if the machine remains connected to a large system after a load change, the frequency variation cannot be calculated, since the flywheel effect of all the connected rotating machinery is one of the determining factors and is unknown. Consequently, when load is rejected from or imposed on a hydroelectric unit, its regulation must be calculated either on the basis of being connected with rotating machinery of limited extent and of known characteristics or on the basis of an isolated generator, operating on a load with no contributing WR^2 effect. The tests to be described were made in such a way that only the WR^2 of the rotating elements of the unit itself affected the regulation. When the load of a hydrogenerator is changed, the speed change depends upon the mechanical characteristics and condition of the governor, the flywheel effect of the turbine and generator, the hydraulic conditions at the plant, and certain hydraulic characteristics of the turbine itself.

Calculating Speed Drop

The speed drop for a sudden increment of load on a unit is

$$S_d = \frac{81{,}000{,}000 \times HP \times T}{WR^2 \times N^2}$$

where S_d = speed drop, per cent
HP = load change
T = time of gate movement, sec
WR^2 = flywheel effect of rotating elements, lb-ft²
N = normal speed of unit

The product of this formula is an approximation, since in it the assumption is made that the input to the generator, during the transition period, increases in a straight line. This is not quite true, but the error is not serious.

$$S_r = \frac{S_d}{1 + S_d/(N_r - 1)}$$

where S_r = speed rise for same load as given, rejected, and with the same time of gate movement
N_r = runaway speed expressed in relation to normal speed

If the turbine is operating in an open flume, the regulation calculated from the foregoing formulas will apply. If there is a closed channel or penstock leading to the wheel, pressure changes will occur during the gate movement and the speed rise or drop will be increased. The average pressure change during the load change should be used.

$$S'_d = \frac{S_d}{(1 - \Delta H)^{3/2}}$$

and

$$S'_r = S_r(1 + \Delta H)^{3/2}$$

where S'_d = speed drop, corrected for pressure drop
S'_r = speed rise, corrected for pressure rise
ΔH = average pressure drop, or rise as the case may be, expressed as a decimal

The maximum pressure change can be obtained (1)[2] by the use of the Allievi charts or other methods. This maximum pressure rise should be used in the design of penstock and casing, but for the purpose of correcting speed regulation, the average pressure rise should be used. Hence, T is the total time of gate movement when applied to the Allievi charts. The actual change in discharge should, of course, be used to compute the change in velocity. The momentum formula is also applicable in obtaining the average pressure rise, where

$$\Delta H = \frac{LV}{TGH_0}$$

For partial load changes the time of gate movement will not be in proportion. There is a "dead time" which includes the time to transmit the speed change from the flyballs to the pilot valve, to the relay valve, to the gate servomotor and, in the case of the Kaplan, to the blade servomotor, also to accelerate the oil in the piping and the mass of the gate mechanism. This should not be confused with the sensitivity of the governor. Table 1 shows approximately how the governor time varies with load.

TABLE 1 VARIATION OF GOVERNOR TIME WITH LOAD

Gate per cent	Governor time, sec					
100	1.5	2.0	3.0	4.0	6.0	8.0
75	1.35	1.7	2.5	3.2	4.7	6.2
50	1.2	1.5	1.9	2.4	3.4	4.4
25	1.05	1.2	1.4	1.7	2.2	2.7
10	0.95	1.0	1.1	1.2	1.4	1.6

It is recognized that the formulas given are approximate. If a more exact method of computation is desired, the step-by-step method of computation may be used, in which the energy transfer between the water column and the runner is considered during short intervals of time during the load change. In using this method account must be taken of the relationship between the blade and gate movements and also of the lag of the blades, caused by the necessity of accelerating the blade mechanism and the oil

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Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

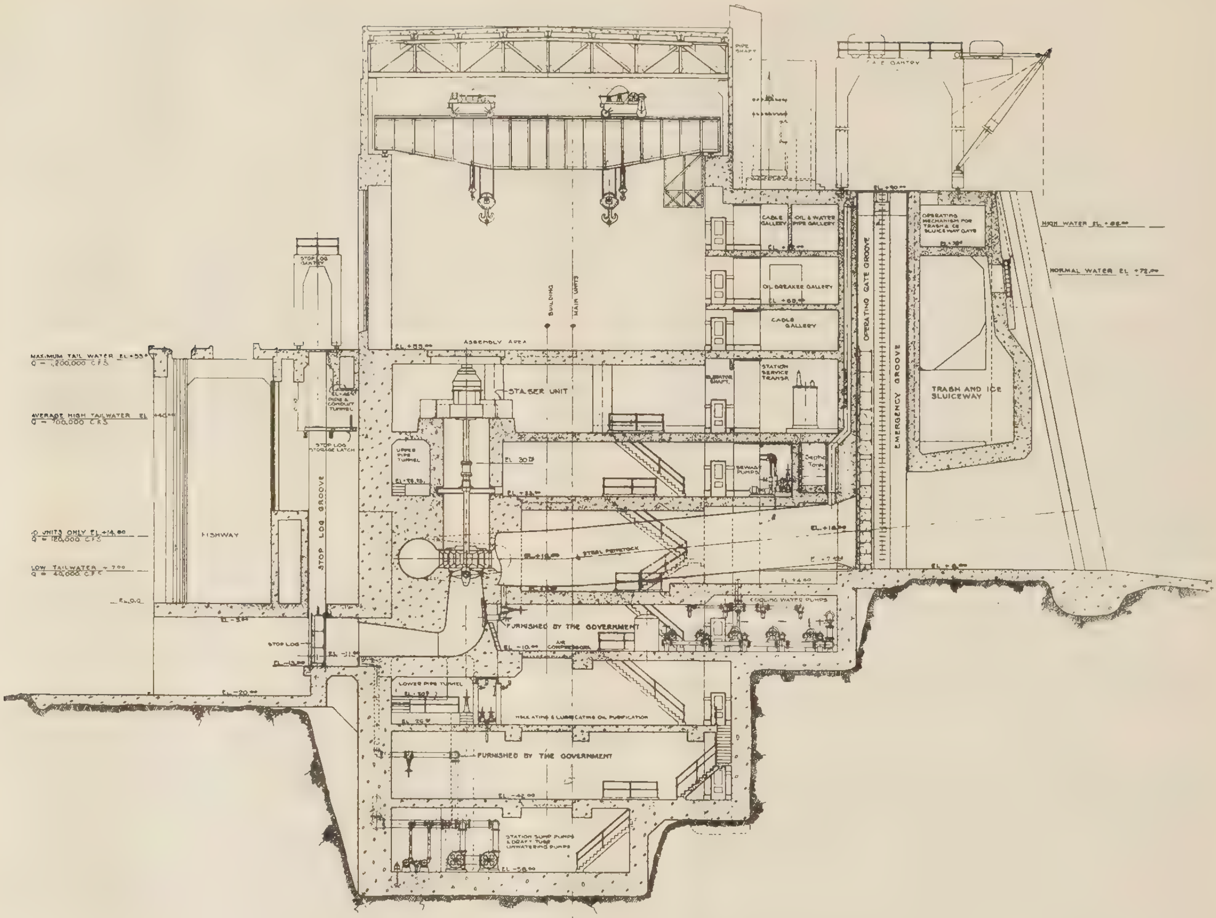


Fig. 1 Cross Section Through Station Service Unit, Bonneville Project

in the piping. For a detailed discussion of this method, reference is made to a paper (2) by Strowger and Kerr.

Application of Method to Bonneville Turbines

The formulas and data have been applied to the Bonneville turbines and compared with field-test data obtained during the initial period of operation. These tests were made by the turbine manufacturer in cooperation with the United States Army Engineers, by whom this project was built. At that time, 1938, there were two Kaplan turbines of 66,000-hp capacity, driving 48,000-kva generators and one 5000-hp Kaplan wheel connected to a 4000-kw generator. A cross section through the station-service unit is shown in Fig. 1. The main turbines are the highest powered Kaplan wheels in the world and operate under a maximum head of 69 ft. For this reason, a substantial submergence was necessary to prevent cavitation. Fig. 2 is a cross section through the powerhouse, which shows the intake, turbine, and draft tube.

Water Rheostat Used to Load Generator

During the early operating period, the transmission lines were not completed so that, in order to load the generators, it was necessary to build a water rheostat capable of absorbing about 50,000 kw (3). With this apparatus, both load-on and load-off tests could be made without the introduction of any flywheel effect other than that of the machine itself.

Procedure for Testing

The station-service unit was tested first. The blades of this runner can only be adjusted with the machine shut down. Therefore, tests of speed rise and drop were made at several blade positions. The blades were adjusted to a given position, the generator loaded to a predetermined output, and then the circuit breaker was tripped. The speed rise was recorded by a Horne tachograph which was connected electrically to the generator through a transformer and which, by means of a flyball-operated pen, recorded the variation of speed during the transition period. A recording Bristol pressure gage was attached to the penstock adjacent to the turbine scroll to obtain the pressure change during the gate movement.

After a load-rejection test was made and the data obtained, the machine was again brought up to normal speed and voltage, and the circuit breaker closed. Since the rheostat was in the same position, instantaneous load-on regulation data were thus obtained from the Horne tachograph and the Bristol pressure gage. Altogether 34 load rejections and additions were made, at five blade settings and several gate openings each. Fig. 3 summarizes the data. It will be noted that while the computed speed rise checks fairly well with the test results, the speed drop from test is substantially less than the computed value. In this connection, it should be noted that, when some load less than the maximum turbine output is imposed, the gates by overtraveling make available more power input to the generator to prevent

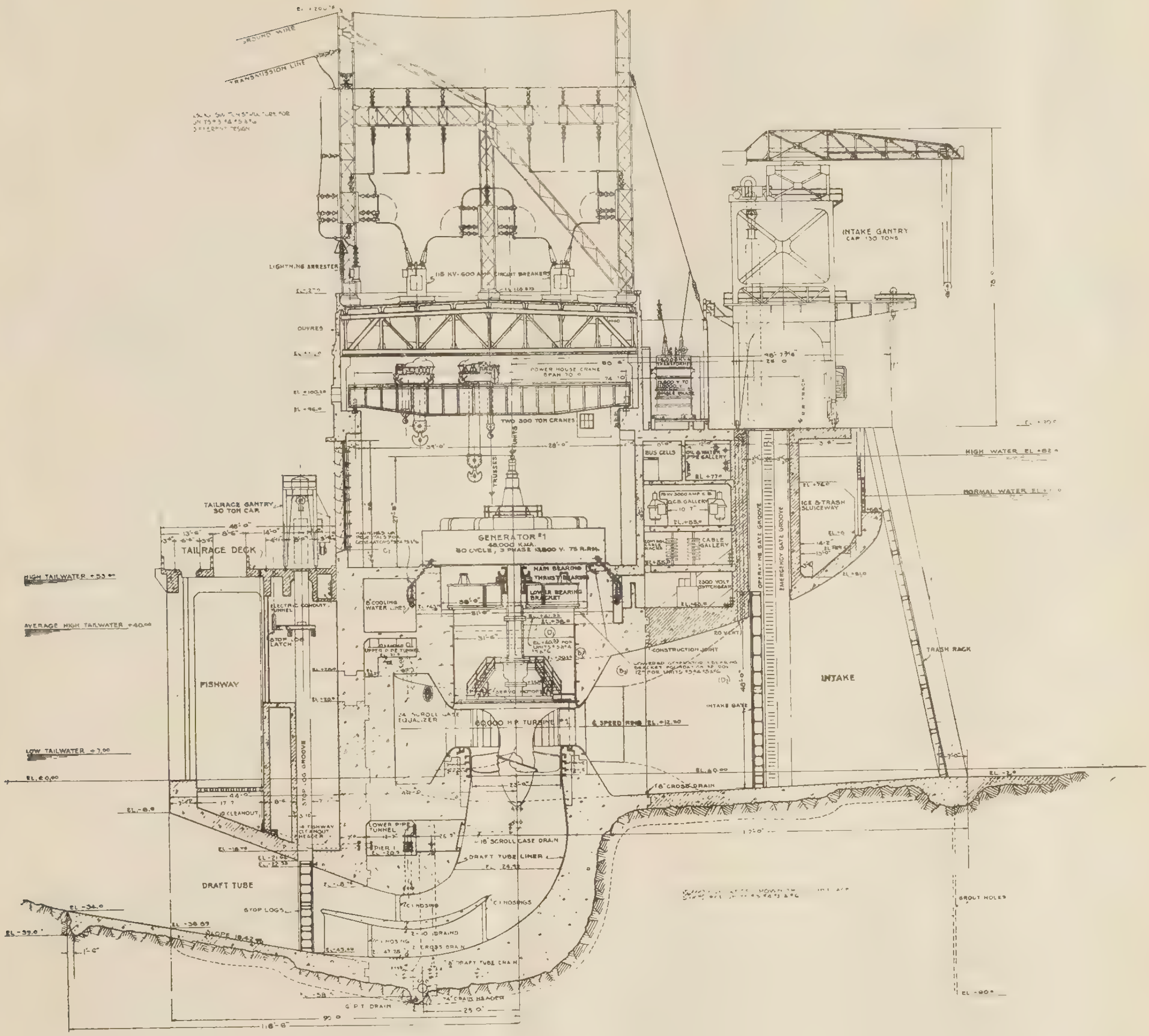


Fig. 2 Cross Section Through Powerhouse; Bonneville Project

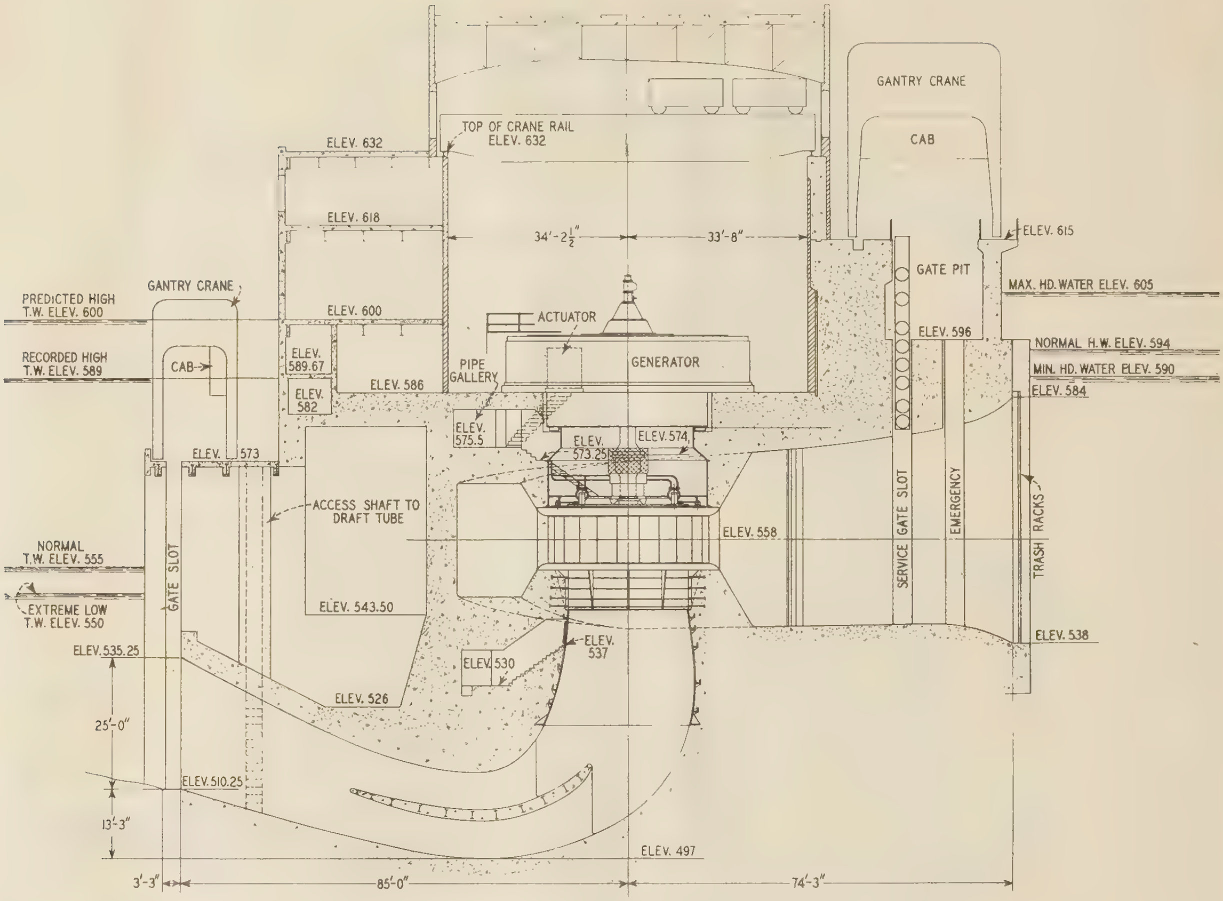
speed drop; hence, the calculated value is on the safe side for all loads except for the maximum at each blade setting. Of course, it must also be realized that the mechanical condition of the governor has considerable bearing on the actual speed change. It should again be pointed out that, since the service-unit runner had manually adjustable blades, the tests cited were, in effect, on a fixed-blade runner at several angles.

Tests on Main Units

The main units are completely automatic Kaplans, that is, the blades move with the gates, taking predetermined positions for each gate opening. Tests were made on both of these units in the same manner as on the service unit, using the Horne tachograph and the Bristol recording pressure gage to obtain the necessary data. The water rheostat provided the load and was quite satisfactory except that, due to its location in the forebay, surges set up by the larger sudden load-on conditions caused severe power swings. More than 32,000 hp could not be taken on for this reason. Up to 60,000 hp, load rejections were made with no trouble, since the surges occurred after the rheostat was separated from the line.

Fig. 4 shows the test results compared with the computed speed change and indicates fairly close agreement. The load-on regulation is relatively not quite so good as for the service unit. This can be explained by the fact that the blades do not move quite as rapidly as the gates and by the dead time in starting them. They are in the flat position before load is imposed and must open before any substantial amount of power can be developed. The blades of the service unit were open and ready to assume load.

It was found that, for large load rejections, the governor did not again come to rest without several oscillations. Fig. 6 illustrates this on unit No. 1, showing the record of speed change versus time taken by the Horne tachograph. As a matter of fact, the unit at first got completely out of governor control and the gates oscillated over their entire range and had to be stopped with the hand load limit. The reason for this is probably as follows: Since the runner is considerably submerged, after the gates



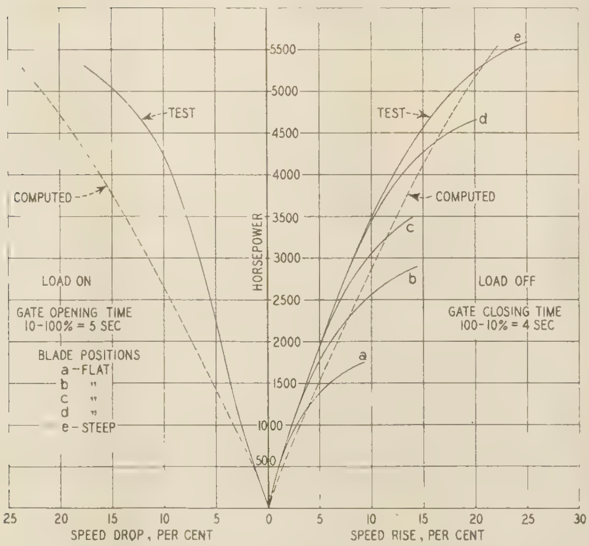


Fig. 3 Summary of Load-Rejection and Addition Data

Fig. 5 Section Through Powerhouse; Guntersville Project of the Tennessee Valley Authority

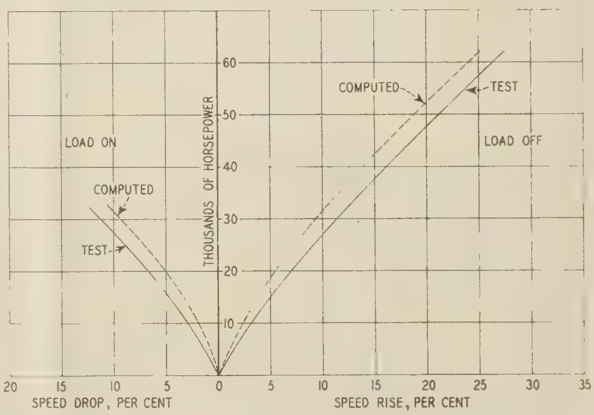


Fig. 4 Load-Test Results Compared With Computed Speed Change

are closed, the runner operates as an axial-flow pump against a shutoff head equal to the barometric pressure plus the submergence. It attempts to pump the water between the runner and gates downward, and absorbs a large amount of power in doing so. This causes a rapid deceleration in speed, which requires a large gate movement and initiates hunting of the governor. This condition was much improved by the installation of a low-gate-limit stop which prevented the gates closing below the speed-no-load position when load was rejected, thus preventing the speed from reaching an excessively low value.

A slow closure at the end of the servomotor stroke aids the situation, if it is so designed as to permit the rapid opening of the gates the instant the governor so requires.

The introduction of air into the space between the runner and the gates destroys the vacuum there and reduces the braking effect on the runner so that the deceleration after the gates are closed is not so rapid.

Tests at Guntersville Plant

The foregoing was illustrated during tests on the turbines in the Guntersville plant of the Tennessee Valley Authority. These are Kaplan wheels of 42,000-hp capacity under 42-ft head at 69.2 rpm, driving 30,000-kva generators. Fig. 5 shows a section through one of the units in this plant.

As there was no water rheostat available, only load-rejection tests were made. The speed rise during load rejection was measured by the same Horne tachograph as was used at Bonneville. A Bristol recording pressure-vacuum gage was attached to the turbine top plate at A, Fig. 7. A spring-loaded air valve is connected at about the same elevation and so arranged that, when the vacuum under the top plate reaches a predetermined value, the valve opens and admits atmospheric air.

Fig. 8 shows the variation in pressure under the top plate at A on unit No. 3 during load rejections of from 7000 to 27,000 kw. It is shown here that, at the higher load rejections, a vacuum of 26 in. of mercury was reached during closure of the gates but that, by the admission of air, it quickly dropped to about 8 in.

A second load rejection of 27,000 kw was made on this unit, with the air valve closed, the effect being shown in Fig. 9. The vacuum reached 30 in. of mercury and then slowly dropped to a fluctuating value of 16 to 20 in., after an intermediate sharp drop to 8 in. It is this rapid change in vacuum which sometimes causes the rotating element to jump from the thrust bearing. The admission of air reduces this tendency considerably.

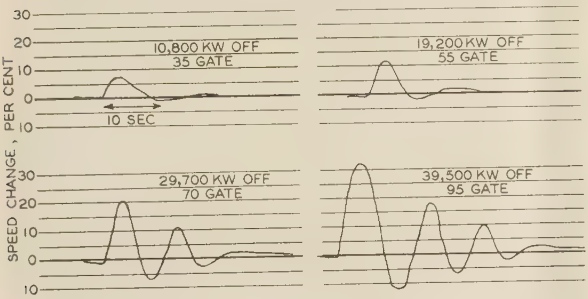


Fig. 6 Record of Speed Change Versus Time; Unit No. 1

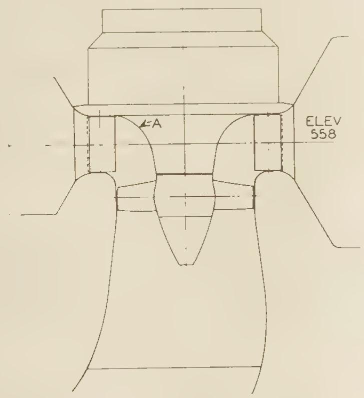


Fig. 7 Recording Pressure-Vacuum Gage Attached to Turbine Top Plate at A

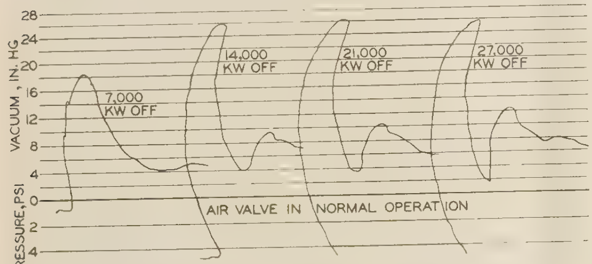


Fig. 8 Variation in Pressure Under Top Plate at A; Unit No 3, During Load Rejections of 7000 to 27,000 Kw

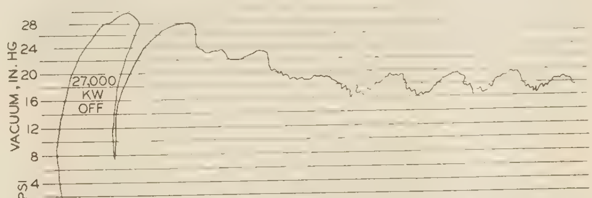


Fig. 9 Effect of Load Rejection of 27,000 Kw on Unit With Air Valve Closed

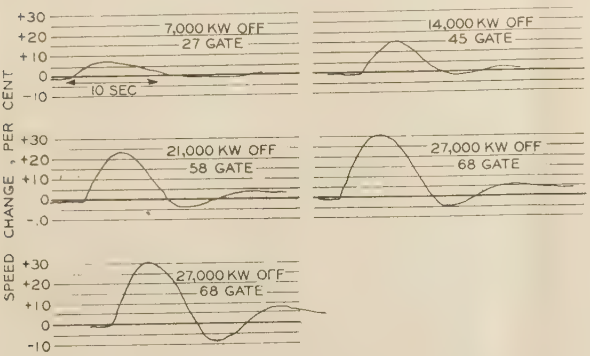


Fig. 10 Speed-Time Charts From Tachograph

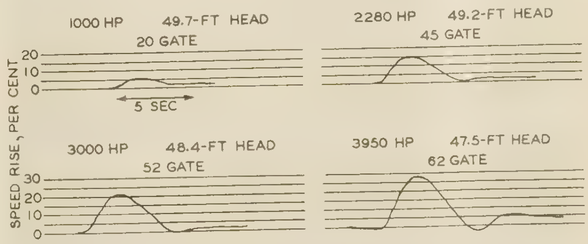


Fig. 11 Example of Normal Speed Regulation

The speed-time charts from the tachograph are shown in Fig. 10, the first four diagrams being for load rejections with the air valve in normal operation and the fifth curve for load rejection with the air valve closed. It will be noticed that the oscillation of speed below and above normal, after closure, is substantially less with the admission of air.

Fig. 11 is given as an example of normal speed regulation. This is a small unit not comparable in size with either Bonneville or Guntersville but shows the type of speed-time curves which are desirable. These are about the character of curves which would be obtained on a Francis turbine during load rejections.

In the event of governor failure, when the gates are wide open and, if for some reason, they remain in that position after the load has been tripped off, runaway speeds can be reached. Maximum speed occurs if the gates are wide open and the blades about half open. This is an abnormal condition, as the blades should be in the steep position at full gate. Under the latter condition, the overspeed is much reduced. The maximum speed which can be reached occurs at some intermediate gate-and-blade position and is about 85 per cent of the value of full-gate—half-blade position.

Overspeed tests were made on both turbines at Bonneville and the results checked within, 1 per cent and 5 per cent, the predicted values from model tests. It was found from these tests that runaway speed is affected by plant σ, where

$$\sigma = \frac{\text{Barometer} - (\pm H_s)}{H}$$

in which

H_s = suction head on runner; minus if runner is below tail water, and plus if above

H = head operating on turbine

During the runaway-speed tests in the field, it was noticed that there was a large opening tendency of the blades. This is a desirable feature since, at large blade angles, the overspeed is still further reduced.

Conclusions

The problem of speed regulation of large Kaplan turbines involves careful consideration of a number of factors which are not of particular concern in the ordinary Francis turbine. It is hoped that this paper has illustrated some of these and that more information will become available, as it is of interest not only to turbine and governor manufacturers but also to plant operators.

BIBLIOGRAPHY

1 "Symposium on Water Hammer," A.S.M.E., 1933.

2 "Speed Changes of Hydraulic Turbines for Sudden Changes of Load," by E. B. Strowger and S. L. Kerr, Trans. A.S.M.E., vol. 48, 1926, pp. 209–232.

3 "Water Rheostat Is Test Load for Bonneville," by R. L. Earnheart, *Electrical West*, vol. 81, Sept., 1938, pp. 42–44.

4 "Speed Regulation in Hydroelectric Plants," by W. F. Uhl, Trans. A.S.M.E., vol. 34, 1912, pp. 379–432.

5 "Effect of Speed Regulation and Water Hammer on the Design of Relief Valves, Penstocks and Surge Tanks," Proceedings N.E.L.A., Serial Report of Hydraulic Power Committee, vol. 84, 1927, pp. 587–612.

6 "Mechanics of Hydraulic-Turbine Pressure Regulation," by Arnold Pfau, Trans. A.S.M.E., vol. 52, 1930, paper HYD-52-4, pp. 29–45.

7 "Water Power Engineering," by H. K. Barrows, McGraw-Hill Book Company, Inc., New York, N. Y., chap. 10, "Speed and Pressure Regulation," pp. 519–543.

8 "Hydro-Electric Handbook," by W. P. Creager and J. D. Justin, John Wiley & Sons, Inc., New York, N. Y., 1927, chap. 28, "Hydraulic Turbines," by W. M. White, pp. 576–658.

Discussion

J. M. Mousson.[3] This paper deals almost exclusively with load rejection. Although this phase of speed control is by no means the most important, it is perhaps the most spectacular. A great deal is claimed for air admission and yet, in the light of the data presented, there could perhaps be some doubt as to the effectiveness of air. Apparently, the author's case rests on a comparison between the two tests made with and without air admission at one 42,000-hp turbine of the Guntersville development under load rejection of 27,000 kw and shown in Fig. 10 of the paper. Based on these data alone, not much can be said in favor of air admission. While a 3 per cent gain in speed minimum may be observed with air, the speed maxima as well as the number of speed oscillations are identical in both tests.

Believing in the beneficial effect of air and to support the author's conclusion, some data obtained with an oscillograph on No. 3 unit at Safe Harbor in 1934 are presented. This turbine is rated at 42,500 hp under a head of 55 ft and operating at 109.1 rpm.

A comparison of the first two graphs in Fig. 12 of this discussion shows that, whereas, a load rejection of 18,000 kw without air injection produced 3 distinct cycles of speed oscillation, only 1½ cycles could be noted with air admission. This is indeed a striking example of the benefit of air injection, as it accomplished a 50 per cent reduction in speed oscillation and an almost equal reduction in time required to obtain speed equilibrium.

From a comparison of governor performance at Bonneville and Guntersville presented in Figs. 6 and 10 of the paper, respectively, it is evident that there must be either a fundamental difference in the governor design or in hydraulic conditions. Does the author attribute the improved performance at Guntersville to the gate-limit stop mentioned by him which prevents the gates from closing below speed-no-load position, together with provisions for a slow closure at the end of the servomotor stroke, so designed as to permit the rapid opening of the gates, if required by the governor?

While air injection did show very beneficial results at Safe Harbor, equally satisfactory governor-performance improvement was obtained with a cam mechanism installed on the governor of

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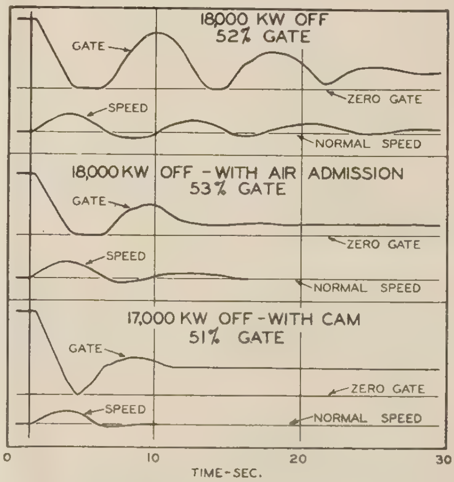

Fig. 12 Speed and Gate-Opening Characteristics Under Load Rejection From Oscillograph Records

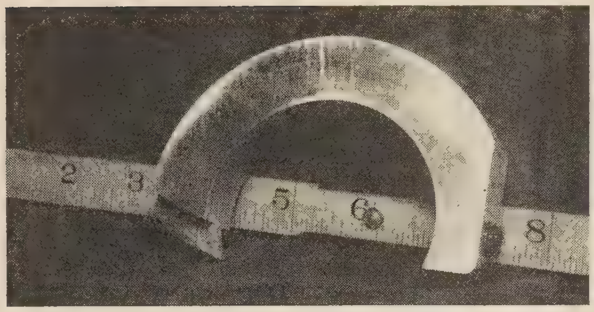

FIG. 13 SHAPE OF CAM TO PREVENT HUNTING

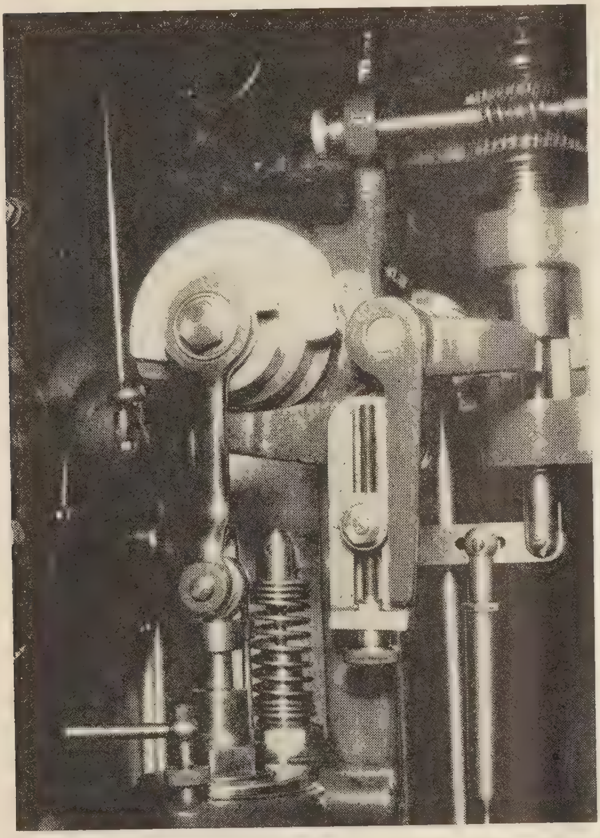

FIG. 14 CAM MOUNTED ON COMPENSATING CRANK OF SAFE HARBOR GOVERNOR

unit No. 3, introducing a large speed drop for the light-load range. The third graph in Fig. 12 shows the speed oscillation obtained with this device under load rejection of 17,000 kw. This cam device is mounted on the compensating crank, Figs. 13 and 14 of this discussion, in such a manner as to contact with the equalizing mechanism at a gate opening of 15 per cent on the closing stroke, causing an upward push on the speed rod and opening the pilot valve in the direction to open the gates. While the gates are still rapidly moving in the closing direction, forces are thus set up in the governor system opposing the closing motion, resulting in a dissipation of stored kinetic energy in the moving mass by the time the gates reach zero opening. The gates at once assume speed-no-load position with no further oscillation. In effect the cam increases the inherent speed drop from approximately 2.5 per cent to approximately 10 per cent below 15 per cent gate.

After the tests on unit No. 3, all of the governors at Safe Harbor were equipped with a cam of this type. Subsequent experience in operation has fully justified their installation. It is believed that, while the method of air injection is satisfactory, its application as a continued feature to operation is not economical, due to the large reserve of compressed air required for emergency. The use of a cam mechanism or some other mechanical feature accomplishing the same results as described is preferable.

F. NAGLER.[4] The type of comparisons made by the author are not encountered as frequently as they should be for the good of the hydroelectric-power field. These comparisons form an excellent answer to the criticism that the basic speed-change formula, given at the beginning of the paper, is only approximate. It is very definitely, however, a workable approximation, certainly as accurate as necessary for the purpose in question.

Three of the four variables in that formula, that is, $WR^2 \times N^2$, divided by HP, are constant for any particular unit and regulation depends principally upon them. It is, ordinarily, much simpler to speak of them as the regulating constant, since they represent the ability of a unit to regulate the speed. Expressed in the inverted form immediately preceding, they ordinarily vary between 5,000,000 and 10,000,000. It is of special interest that certain sizes and speed ranges of units must have additional flywheel effect added to the generator to raise this regulating constant to a feasible figure, whereas, other types, particularly in the high-speed and large-capacity ranges, inherently possess a regulating constant sometimes larger than is necessary. This is dictated by generator design.

Putting this regulating constant in the denominator of the author's basic speed-drop formula leaves the speed regulation dependent upon the governor time T. Modern governor guarantees mean little or nothing beyond the statement that, if the regulating constant is a certain amount, if the unit is disconnected from the load, and if a certain governor time is used, certain figures, which have practically nothing to do with the regulation of a system, can be computed.

Usually, the governor time, T, ranges somewhere between 2 sec and 10 sec. Most governors are purchased on the basis that they will close the gates in from 2 to 4 sec. Some of the largest operating companies, thereafter, normally adjust all of their governors so that they cannot close in a shorter time than 7 or 8 sec. Here we have a situation carried over from the days of isolated units, entirely comparable with insisting on a whip socket on the automobile dashboard. Bids may be compared on the basis of a few per cent difference in speed-change guarantees in the face of the realization, should thought be given to it, that these figures will probably be doubled in actual operation and probably with definite improvement to the system frequency.

The writer would suggest that, in the future, consideration be given to a simple statement of the regulating constant as an indication of the influence of the unit on system regulation and a statement of the guaranteed minimum governor time as an indication of the governor capacity. The absurdity of present-day governor guarantees will be somewhat reduced thereby.

Referring to the formula for average pressure rise immediately preceding Table 1, a somewhat closer approximation of the pressure rise and pressure drop, through the ranges usually experienced in practice, is obtained by adding 10 per cent to ΔH for pressure rises and subtracting 10 per cent from ΔH for pressure drops. This brings the approximate formula fairly closely in line with Allievi.

[4] Chief Engineer, Canadian Allis-Chalmers, Ltd., Toronto, Canada. Life Member A.S.M.E.

It is significant that the author's charts of speed regulation show no results with load thrown on. This is logical because, in actual plant operation, except in connection with isolated units, as on a mining load, it is seldom that a unit ever has to meet sudden large increases in load. It is of interest that the author made tests with the load thrown on. The writer has used a water rheostat similarly, but was rather intrigued with the fact that, whereas, a load of, say 20,000 kw could be dropped instantly by pulling a switch, the same load was not picked up if the switch was immediately reclosed. There seemed to be a definite lag in the ability of the water rheostat to pick up the load it had just dropped. It would be of interest to have the author's comments on whether he observed any similar action in connection with load thrown on.

In conclusion, the writer would again emphasize the desirability of some change in the attitude of both purchasers and manufacturers in making governor guarantees. The formulas presented by the author have served their purpose for about 30 years, during which time probably 90 per cent of our turbine-horsepower capacity has become connected to the larger systems. Those systems operate the year round with 1 or 2 per cent maximum speed change. The units cannot receive large increases in load and, if they lose their load, they are disconnected from the system and have no influence on it. It may logically be concluded, therefore, that governor guarantees up to 25 or 30 per cent speed change are a clumsy attempt to effect a set of regulating conditions which may be much more simply and accurately expressed by regulating constant and governor time. Progressive purchasers are already looking at these basic figures, and greater progress would probably be made in the hydraulic-turbine field if more attention were directed toward them rather than to outmoded and academic speed-regulation tables.

J. F. Roberts.[5] This paper covers quite fully the various factors which must be coordinated in order to obtain satisfactory regulation of Kaplan-type turbines. In the case of the Guntersville turbines of the Tennessee Valley Authority with which the writer was particularly interested, it was possible to vary both the rate of movement of the runner blades and of the wicket gates, and, thus, in the field, determine that combination which would result in the most satisfactory combination.

As finally adjusted, the wicket gates were set to open and close in from 8 to 10 sec with the runner blades opening in about 8 sec, but requiring about 40 sec to close. While faster operation of the wicket gates is possible, it was found that on such a large system the frequency of the system varies slowly even for a sudden loss of 30,000 kw. More rapid operation of the wicket gates imposes heavy loads and shocks on the wicket-gate mechanism without apparent improvement to the system-speed regulation. It also causes greater pressure variations, particularly in the draft-tube water column which, while possibly not dangerous to the structures, are unpleasant and, in this case, apparently unnecessary.

It is interesting to note that the author's tests demonstrated it to be almost impossible to obtain a flat blade angle of the runner at full wicket-gate opening and runaway speed. This means that on future projects it may be possible to make an appreciable saving in generator costs, since generators designed for full gate and flat-blade runaway must be designed for about 270 per cent of normal speed, as compared with about 200 per cent speed for full gate and steep pitch on the runner blades.

E. B. Strowger.[6] The author has presented test data on the

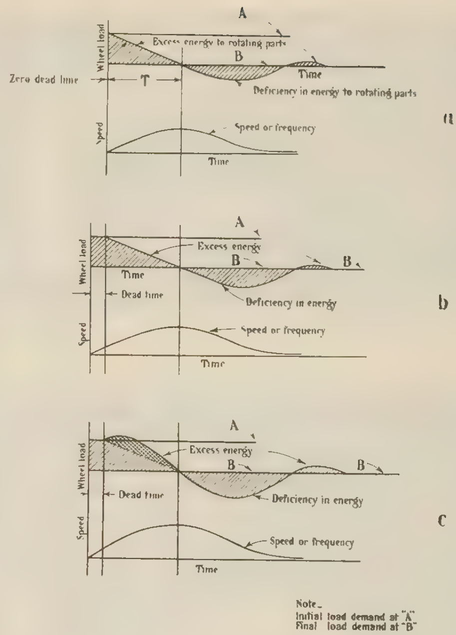

Fig. 15 Excess Energy Caused by Load Change

speed regulation of Kaplan turbines which are of value to hydraulic engineers engaged in water-power development, since but little if any such data have been published in the technical press to date. His approach has been through an approximate equation for reasons stated in the paper. In discussing the matter, he has referred to a paper (2) by S. L. Kerr and the writer, dealing with the step-by-step method of computing speed rise or speed drop for sudden load changes, in which a rational approach has been attempted and in which application of this method has been made to Francis turbines with good results. The writer therefore wishes to discuss the possibility of the application of this step-by-step method to Kaplan turbines.

Let us first consider a few simple but fundamental relationships. Referring to Fig. 15 of this discussion, assume an isolated unit operating on a load with little or no connected Wr^2 so that the only steadying influence on the speed would be the flywheel effect of the generator rotor. Suppose a load change takes place, as shown in Fig. 15 (*a*), the load dropping from A to B, and let us assume that the length of penstock and of draft tube is negligibly small. The turbine load then decreases approximately as a straight line with respect to time, as the governor closes the turbine gates to the new position of load demand. In this case, the excess rate of energy transfer from the penstock to the turbine above that demanded is gradually decreased as the governor moves the gates toward the new position and a rise in speed of the rotating parts takes place. This rise in speed actually makes the gates overtravel so that for a short time a deficiency of energy is transmitted to the rotating parts, followed by another small amount of excess energy, after which the travel of the gates has become damped to the extent that equality is again established between rate of energy supply and rate of energy demand.

In the case assumed, i.e., with the unit operating on an isolated load having no Wr^2, the excess energy produced in the penstock must be balanced by the deficiency of energy, before the initial speed is re-established. It should be noted that, if no governor adjustment is made, the speed of the unit will not return exactly to its initial value, but the final value of speed will be slightly higher, depending upon the load dropped and the in-

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[6] Hydraulic Engineer, The Niagara Falls Power Company, Buffalo, N. Y. Mem. A.S.M.E.

herent speed-drop setting of the governor. This can also be seen in Figs. 6, 10, and 11 of the paper. The speed rises to a maximum value at the end of time T, as shown in Fig. 15 of this discussion. This maximum is not reduced by the subsequent overtravel of the gates either for loads off or for loads on. The overtravel, however, brings the speed back approximately to the normal value, after the maximum speed is reached faster than would be the case with no overtravel.

Fig. 15 (b) shows a similar load change with the introduction of the element of governor dead time, and indicates that dead time increases the excess energy which must be absorbed by the rotating parts and, therefore, results in a greater change in speed.

Now, passing to the usual installation where a physical length of penstock and draft tube must be considered, as illustrated in Fig. 15 (c), the rate of change in energy transfer from runner to rotating parts is not uniform and, consequently, the turbine load plotted with respect to time is not a straight line. In fact, the power produced by the turbine actually rises at the beginning of gate movement and then decreases as shown. That part of the excess energy shown double-hatched in Fig. 15 (c) is caused

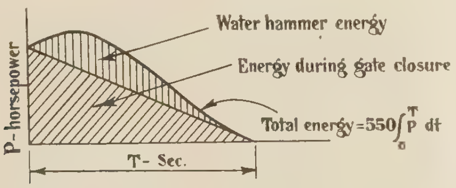

Fig. 16 Same Diagram as Fig. 15 (c), Except Governor Dead Time Is Considered to Be Zero

by the water hammer, produced in the penstock and draft tube, due to the destruction of velocity, and may be a rather large percentage of the total excess which must be absorbed. The speed of the isolated unit changes, as shown on the diagram.

Now let us consider Fig. 16 of this discussion, which presents the same diagram as shown in Fig. 15 (c) except that the governor dead time is considered to be zero. The energy absorbed by a rotating mass in changing the speed from N_1 to N_2 may be expressed by

$$\frac{Wr^2}{5870}(N_2^2 - N_1^2)$$

The expression $550 \int_0^T Pdt$ represents the total energy delivered from the water column to the runner during the gate closure and may be equated with the foregoing expression, representing the energy absorbed by the rotor. If the resulting equation is then solved, we obtain the following expressions for percentage of speed rise and percentage of speed drop.

$$\text{Speed rise, per cent} = \frac{N_2 - N_1}{N_1} \times 100$$

$$= 100\left[\sqrt{\frac{3{,}229{,}000\int_0^T Pdt}{Wr^2 \cdot N_1^2} + 1} - 1\right] \ldots [1]$$

Speed drop per cent

$$= 100\left[1 - \sqrt{1 - \frac{3{,}229{,}000\int_0^T Pdt}{Wr^2 \cdot N_1^2}}\right] \ldots [2]$$

It should be noted that these expressions contain the same variables as the first equation given by the author but that the expression is radically different in form. The question now arises as to how to evaluate the integral in these equations. For turbines with appreciable length of penstock, the evaluation of this integral depends upon the water hammer produced, the relation between runner efficiency and gate opening, the relation between runner efficiency and speed, and the rate of gate motion.

For open-flume settings and where the load varies in a straight line with respect to time, the integral may be closely approximated in the case of Francis runners by

$$\int_0^T Pdt = \frac{1}{2} HP \cdot T \ldots\ldots [3]$$

where HP = initial load in horsepower for speed rise and final load for speed drop

Consequently, for open-flume settings, these relations become for Francis wheels

$$\text{Speed rise} = 100\left[\sqrt{\frac{1{,}614{,}000\, HP \cdot T}{Wr^2 \cdot N_1^2} + 1} - 1\right] \ldots [4]$$

$$\text{Speed drop} = 100\left[1 - \sqrt{1 - \frac{1{,}614{,}000\, HP \cdot T}{Wr^2 \cdot N_1^2}}\right] \ldots [5]$$

In the case of Francis runners, the step-by-step method of calculation lends itself to the evaluation of this quantity, because the power input to the runner can be calculated for each small interval of time during the gate motion. However, for Kaplan turbines, as pointed out by the author, there may be a lag of the motion of the blades behind that of the gates, which changes the steady-state gate-opening-efficiency relationship and, consequently, the quantity considered cannot be evaluated for this type of turbine without determining the effect of the lag and other factors which influence the efficiency of energy transfer from the water column to the runner during the gate movement. Possibly this effect can be determined experimentally for a number of Kaplan turbines and an average coefficient obtained. In the case of the Bonneville main unit, the writer attempted to do this for the full-load point on the curve of the author's Fig. 4, and obtained the relation $\int_0^T Pdt = 0.39 \times HP \cdot T$. This showed, in the case of this particular Kaplan, that, even with some water-hammer energy present, less than 50 per cent of the energy represented by the product of HP and T was effective in speeding up the rotor during the load rejection. This means that the area under a straight-line curve of horsepower versus time represents too much energy transfer and shows the error, in one particular case, of the assumption made in deriving the approximate equation, i.e., a straight-line variation of these quantities during the transient.

To compare a typical Francis speed-rise problem with that of the Kaplan turbine, the writer made some computations for a Francis turbine having about 690 ft of penstock and obtained the figure of 0.61 as the coefficient of the quantity ($HP \cdot T$) in Equation [4]. This is of the right order because for an open-flume setting of a Francis turbine, it would be reasonable to expect to obtain a coefficient of about 0.5.

As a check on the rationality of the equations presented here, the writer determined the coefficient of the product ($HP \cdot T$) in the case of the speed-drop problem of the Bonneville service unit shown by the author's Fig. 3. A coefficient was determined by using the test data for a load of 5000 hp and then, since this unit had manually adjusted blades, the speed-drop points for 75 per cent gate and 50 per cent gate were computed using this coefficient, but applying estimated values of power for the two gate positions. A curve of speed drop was obtained, which checked the experimental curve at the part-load points within about 2 per cent and the shape of the curve was thus much closer

to the test curve of Fig. 3 of the paper than that of the computed curve of Fig. 3.

In conclusion it may be possible to apply the step-by-step method to Kaplan turbines by obtaining a graphic record of the movement of both the guide vanes and the runner blades during load changes and with the use of complete turbine characteristic curves, by evaluating the energy transfer from water column to rotor. As an alternative it may be possible to obtain average values of the coefficient as described for wheels of a given manufacturer or of a given design for load rejections at say 100 per cent gate, 75 per cent gate, and 50 per cent gate and these data would enable one to predetermine the speed-rise power curve similar to Fig. 4 of the paper for any projected installation, knowing of course the size of the unit, governor time, Wr^2, etc. These suggestions are made in an attempt to apply a method of computation which takes into account the fundamental relations of energy transfer which take place during the load change between the penstock and the rotor.

Author's Closure

Mr. Mousson's discussion substantiates the author's contention as to the advantage of admission of air to a Kaplan turbine during load rejection. Fig. 14 shows an alternate method used at Safe Harbor to prevent the secondary speed and gate swings. It is believed, however, that this method is not as good as that of air admission. The author knows of at least one occasion where one of the Safe Harbor units jumped from the thrust bearing during load rejection. Air admission would eliminate that trouble. It should be pointed out that air is injected under pressure at Safe Harbor, whereas atmospheric air would serve the same purpose. In spite of about 15 ft of submergence of the runner, atmospheric air was readily taken into the No. 3 unit at Bonneville and produced speed-change diagrams during load rejection exactly similar to those illustrated in Fig. 10 of the paper which were obtained on one of the Guntersville units. This is evidence that the secondary swings in speed which occurred on unit No. 1 were eliminated by the use of air on unit No. 3. It is believed that this answers Mr. Mousson's question.

Figs. 3 and 4 show speed drop as well as speed rise for load change on one of the Bonneville main units and on the service unit. Mr. Nagler states that there seems to be a definite lag in the ability of the water rheostat to pick up the load it had just dropped. The curves of Figs. 3 and 4 indicate somewhat less speed drop than was computed. There may be something in Mr. Nagler's statement. However, this was about as close to instantaneous load on regulation as could be obtained on the tests.

The author agrees with Mr. Roberts that it is highly desirable to be able to vary the rate of blade and gate travel independently.

Mr. Strowger wants to attack the problem of speed change by computing the water-hammer energy during gate closure and adding it to the normal energy put into the runner during this period. He is, of course, quite right in this as the energy which produces speed change is the sum of these two minus the increased generator windage and friction. However, as pointed out in the paper, there is a variable lag in the blade movement in addition to the dead time of the governor. Lack of certain and definite information makes it difficult to apply Mr. Strowger's method on Kaplan turbines.

Development of the Automatic Adjustable-Blade-Type Propeller Turbine

By R. V. Terry,[1] Newport News, Va.

In this paper the design and testing of automatic adjustable-blade-type propeller turbine models are treated, followed by an analysis and discussion of the model-test results. Four installations are described, aggregating a rated output of about 44,000 hp in five units. Operating experience and results with field units are considered. Mention is also made of further development work now in progress. In conclusion a summary is given of the principal features of this new type hydraulic turbine.

Introduction

The idea of adjusting the pitch of the blades of hydraulic-turbine runners to suit the load is relatively old. Such a runner was described[2] in 1867. For about 25 years development has been in progress on the Kaplan adjustable-blade propeller type turbine. That type has reached a high degree of perfection in employing an oil-pressure system, associated with the governor-pressure system, to operate the runner blades in synchronism with the gates. From the many published articles on adjustable-blade turbines, the advantages of varying the pitch of the blades are well known. The principal advantages are as follows:

(*a*) Increase in capacity above normal.

(*b*) Sustained efficiency over the larger part of the load range.

(*c*) Reduction of the minimum head at which the turbine will develop power.

(*d*) Greater stability of operation, particularly at low and intermediate loads.

In 1928, the author began studies for an adjustable-blade type of runner wherein the blades would move automatically with changes in flow through the runner, as well as with changes of speed and head. It was his rather radical idea[3] so to pivot the blades with reference to their centers of pressure that they would tend to adjust themselves to the most efficient angle for each condition of gate opening and head for constant-speed operation.

Model Tests

Surprisingly satisfactory results were secured with the first models built and tested in 1930, which encouraged further experimentation. Other groups of tests have been conducted at intervals for the last 10 years. Altogether, more than 200 tests have been made on about 30 different models, using about 15 sets of blades, some of which were altered as the test program progressed. The model tests were made on $16^1/_2$-in-diam runners under heads varying from 9 to 12 ft, in the Newport News Hydraulic Laboratory.

It was early found that the blades must be pivoted considerably ahead of their centers of area. In fact this was predicted from a study of the characteristics of airfoils and hydrofoils. In order that the blades might have a tendency to adjust themselves automatically and with minimum operative moments, it was also found desirable to pivot them slightly ahead of their centers of pressure. This resulted in a tendency for the blades to open at all times, at least in normal operation at or near the best efficiency conditions. As the test program progressed, certain definite trends and characteristics were found which influenced subsequent designs. As the development gradually unfolded, unproductive leads were encountered, as well as new and desirable characteristics disclosed.

It is assumed that those interested are somewhat familiar with turbine design and with some of the terms used in aeronautics dealing with airfoils. In Fig. 1 (upper left) is given typical velocity diagrams for an average blade section. Diagram u_1, c_1, w_1 is for the inlet edge of the blade, while diagram u_2, c_2, w_2 is for the discharge edge. Terms w_1 and w_2 represent the respective velocities relative to the blade and w_a is the vectorial average of w_1 and w_2. The function $u_1 = u_2$ is the tangential velocity of the blade. Terms c_1 and c_2 are the absolute velocities at inlet and discharge, respectively.

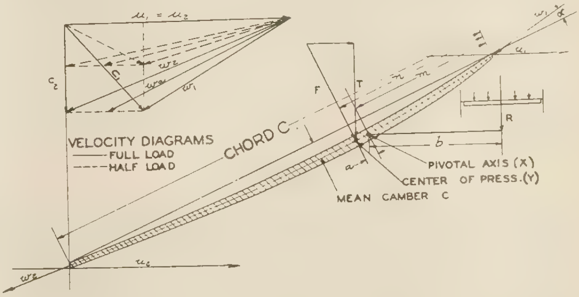

Fig. 1 Typical Blade Section and Velocity Diagrams

A typical blade section is also shown in Fig. 1, with chord C and mean camber c. The angle of attack α is the angle the inflowing water w_1 makes with the chord C. The center of pressure is shown at point y which is at distance n from the leading edge of the blade, measured along the chord. The pivotal axis is shown by point x which is at distance m from the leading edge of the blade. The equivalent total force on all the blades is represented by F, while T represents the axial component of F or the hydraulic thrust. The "hydraulic moment," tending to open the blades, is Fa, a being $n - m$. The hydraulic moment is balanced by a "reactive moment" Rb, tending to close the blades, produced by a balance piston in the runner hub acting on the blades with lever arm b, R being the total piston load.

An analysis of tests on airfoils similar in shape to turbine blades shows that the distance of the center of pressure (point y in Fig. 1) from the leading edge in proportion to chord length or n/C reaches a minimum value at a moderate angle of attack α, approximately the most efficient angle of attack in the case of a turbine blade. The curve of n/C plotted against angle of attack

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[1] Hydraulic Engineer, Newport News Shipbuilding and Dry Dock Company, Newport News, Va. Mem. A.S.M.E.

[2] U. S. Patent No. 67,994, issued to O. W. Ludlow in 1867.

[3] U. S. Patents Nos. 1,858,566 and 1,907,466.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

is fairly flat for several degrees variation in α from the value giving minimum n/C. But on either side of that value the center of pressure travels downstream, Fig. 2. A further analysis of the characteristics of airfoils shows that the location of the center of pressure varies with the proportional mean camber c/C. This

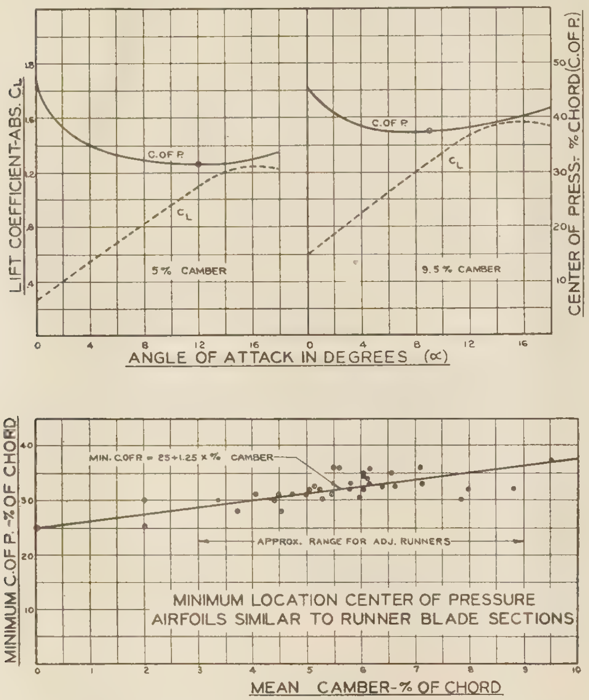

Fig. 2 Typical Characteristics of Airfoils

variation has been found to be fairly accurately expressed by the formula

$$n/C = 0.25 + 1.25 \times c/C$$

Thus, a blade section with a mean camber of 8 per cent would have its center of pressure located at 35 per cent chord, and should be pivoted at about 33 per cent chord. The proper camber of turbine blades gradually decreases from the hub to the periphery with a resulting change in the location of the center of pressure. We are primarily interested in the location of the center of pressure of the blade as a whole rather than in its location for each individual section. It has been found more practicable in producing simple blade shapes to place the axis further downstream near the hub and further upstream near the periphery, producing an over-all result as though detail considerations were given to pivoting each section upstream from its individual center of pressure. The runner blade is usually divided into four annular sections of approximately equal discharge and consideration given to the average result.

Fig. 3 shows a typical runner model with six blades, while Fig. 4 shows a sectional view of the model, and Fig. 5 a typical layout of a runner blade.

As may be expected, it was found that blades in echelon, grouped around a common hub, and enclosed in a housing, do not act exactly as they do in the case of single airfoils tested in infinite space. The echelon arrangement, together with the reactive nature of a turbine runner, tends to move the center of pressure somewhat downstream. This tendency has been found to increase with the camber, number of blades, and with blade width

Fig. 3 Runner Model With Six Blades; $16^1/_2$-In. Size

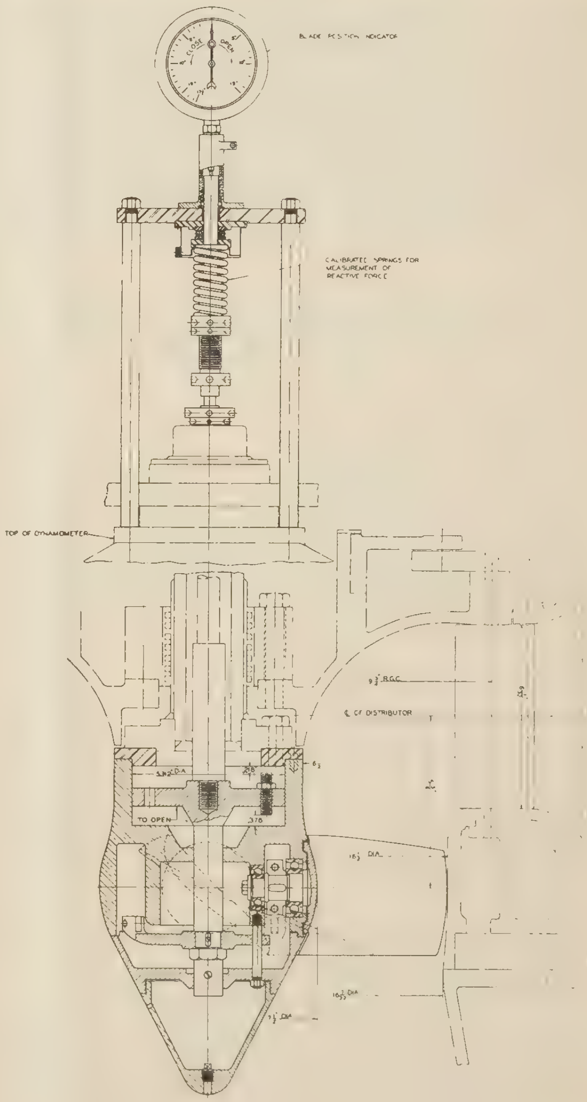

Fig. 4 Sectional View of $16^1/_2$-In. Test Model

or overlap. An idea of the degree of movement of center of pressure may be had from Table 1.

Tests for Model Development

The test procedure for developing a model was as follows:

1 Design the blades, based upon data secured from previous tests and from the consideration of the characteristics of airfoils.[4]

2 Test the model for speed, power, and efficiency at several blade positions, usually six, with the blades locked.

[4] "Aerodynamic Characteristics of Airfoils—VI," National Advisory Committee for Aeronautics, Tech. Report No. 315, 1929.

TABLE 1 COMPARISON OF ADJUSTABLE-BLADE RUNNERS FROM TESTS ON MODELS 16½ IN. IN DIAM

Relative blade camber	Normal			Small
Number of blades	4	5	6	6
Type no.	164	165B	166B	179D
Best unit speed, N_1	200	180	160	160
Specific speed, N_s	160	140	120	120
Pivot point, per cent of chord:				
At hub	34	37	40	36
At periphery	24	27	30	21
Average	29	32	35	28.5
Mean camber, per cent of chord:				
At hub	7.7	8.7	8.4	7.1
At periphery	2.3	3.1	3.6	2.5
Center of press, per cent of chord:				
At mean flow line	31.0	34	37.0	30.5
From airfoil tests	30.0	31	32.5	30.5
Discharge angle at periphery, deg	14.5	16	18.5	18.5
Chord angle at periphery, deg	17	19.5	22	21
Hub diameter, per cent of outside diameter	40	42.5	45	45
Blade width (plan) ÷ blade pitch:				
Hub	0.88	1.06	1.09	1.09
Periphery	0.77	0.88	1.00	0.91
Blade area, per cent of annular area	95	114	128	117
Head range (approx), ft	10–40	25–55	40–70	..

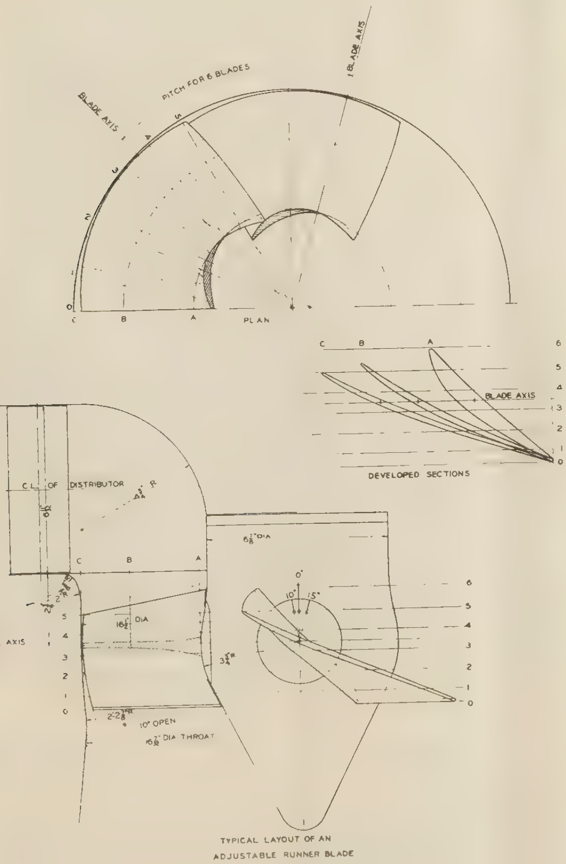

Fig. 5 Typical Layout of an Adjustable Runner Blade

3 Unlock the blades so that they are free to assume a balanced position at different operating conditions. Calibrated springs with various amounts of compression are used to produce reactive moments tending to close the blades.

4 Plot moment diagrams, one for each of several speeds, as illustrated in Fig. 6.

5 Study the required reactive moments for best efficiency operation, best gate-blade relation, for each speed, and reduce them to a constant speed for variable-head operation. In analyzing the moments, gravity effects are also considered.

6 Put the model in automatic operation with a suitable reactive moment with the wicket gates under governor control, driving an electric dynamometer, to determine its behavior under starting and runaway-speed conditions, as well as under normal power-producing conditions.

7 Finally, determine the movement of the center of pressure from the test data. This is done primarily from a consideration of the measured hydraulic moments and from the calculated thrust. However, some blades are tested with two locations of the axis and the center of pressure computed from the two sets of hydraulic-moment data.

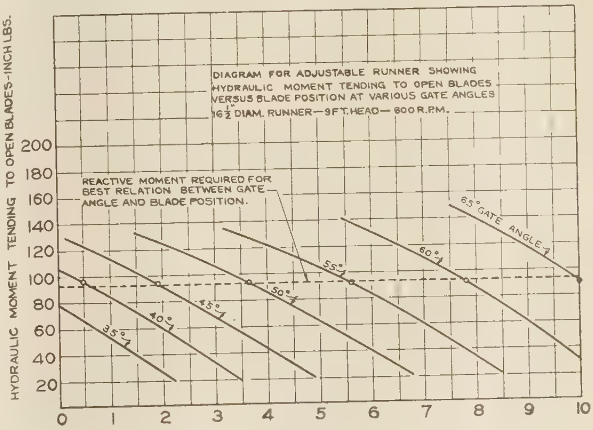

Fig. 6 Moment Diagram From a Model Test

From Fig. 5, it may be seen that the general shape of blades employed with these adjustable runners is quite simple. A cylindrical section approximates that of a portion of a parabola, with which uniform deceleration of the absolute whirl component of the water is theoretically obtained. The maximum camber is placed at about one third the chord distance from the leading edge of the blade. The face and back of the blades over most of their area are each produced by a system of straight lines (for horizontal sections) meeting at a common point, at the runner axis for the face and near the axis for the back. The blades are laid out for an intermediate angular position, usually for 60 per cent open position. The pitch is made uniform on each radial line. However, the inlet edge of the blade is extended at the hub, resulting in greater pitch at that point. Cambers much greater than those theoretically required from a consideration of velocity diagrams are selected to allow for the bending effect of the water streams before and behind the runner.

As a result of tests to date the following general conclusions may be drawn:

1 The blades tend to follow the gates, therefore tend to open as gates open and vice versa.

2 With proper blade shape and pivoting, a reasonably constant reactive moment may be used for operation at constant speed at or near best efficiency, for the various gate openings and for varying unit speeds as the head is changed.

3 At low unit speeds, high heads for constant speed, the required reactive moment for best efficiency operation decreases somewhat as the load increases, that is, with a constant reactive moment, the blades move less than required for perfect synchronism with the gates.

4 At high unit speed, low heads, the blades tend to move more than is required for the best gate-blade relation.

5 When properly pivoted, the blades have a strong tendency to open when starting the wheel from rest. This tendency is greatest with 4-blade, and decreases with 5- and 6-blade runners.

6 The blades have a strong tendency to open if the runner runs away at constant head. This effect is least with 4 blades and greatest with 6 blades. The hydraulic moment at runaway speed usually ranged from 4 to 5 times normal with the models tested.

At the present stage of development, it is not considered practicable or even possible to explain all the various characteristics by mathematics and hydrodynamics. The problem is too complicated, considering the numerous variables. However, a discussion of a few concepts is perhaps in order.

1 Why should the reactive moment required for best efficiency be fairly constant for varying loads and heads? To start with, the turbine drives an electric generator at a constant speed. The tangential velocity is relatively high compared with the variable axial velocity. The result is a fairly constant relative velocity w_a, which is the one that produces the predominant dynamic effect on the blades. For best efficiency under the various operating conditions, the angle of attack is naturally about the same, resulting in approximately the same center-of-pressure location. A constant relative velocity, together with a fixed angle of attack and center of pressure, would, of course, create a constant hydraulic moment.

2 Why should the moment increase as the gates are opened? As the gates open, the angle of attack increases. This causes a slight movement of the center of pressure downstream, and an increase in the normal force, resulting in increased moment. This change in conditions overbalances the reactive moment and the blades move open. However, as the blades move open, the angle of attack and normal force decrease and the center of pressure moves upstream until a new balanced condition is reached.

3 Why do the blades open when starting the turbine? With the runner at rest the water from the gates impinges directly against the runner blades at a very large angle of attack. Under this condition the center of pressure is located well downstream with reference to the axis of the blade. Because the blade is stationary and the angle of water deflection large, the force on the blade, per unit volume of water and head, is relatively high. This results in a large opening moment that overbalances the designed reactive moment at a relatively small gate opening. With 4-blade runners, the blade area is considerably less than the annular area between the hub and periphery. The blades of such runners have a very strong tendency to open when starting and the blades will go wide open. With 5-blade runners, and particularly those with 6 blades, the overlap of the blades may be such that the leading edge of one blade obstructs the impingement of water on the next blade. This results in less movement of the center of pressure and a smaller opening moment. Consequently, the blades of such runners do not open as much when starting.

4 Why do the blades open when the wheel runs away? Under this condition two more or less opposing actions take place. One consists of a reduction in the angle of attack which tends to reduce the blade force and moment. The other consists of an increase in the relative velocity, which tends to increase the force and moment approximately as the square of that velocity or speed. The latter action is predominant, resulting in rapidly increasing moment, particularly at the higher speeds. Here again there is a considerable change in characteristics as the number and overlap of the blades is increased. The opening tendency is stronger with the larger number of blades.

Tests were also made to determine the effect of the shape of the throat ring below the axis on the power and efficiency characteristics as well as on the hydraulic moment on the blades. These included spherical, cylindrical, and intermediate shapes. As was expected, the spherical shape had an effect in moving the center of pressure downstream as the blades approached their open position. It was found better to make the ring cylindrical for 4-blade runners and somewhat curved for the 5- and 6-blade runners. At the higher specific speeds, it was found that, for a given diameter of runner, the reduction in power caused by spherical ring more than offsets any gain in efficiency at the largest blade angles. When efficiency was plotted against power, the curve for a cylindrical ring was slightly above that for a spherical ring near full load. At low and intermediate loads, no difference was found.

The model tests have established a relation between the center of pressure and the axis of the runner blade which should direct attention to the advantages in utilizing the minimum force to move any type of adjustable-blade runner. When the blades are pivoted too far downstream, large moments are produced, particularly at runaway speeds. If the axis of the blade is moved upstream, closer to the center of pressure, the force needed to move the blades is smaller and the machine is safer from the standpoint of overspeed. However, without the use of antifriction bearings, there is a possibility of encountering the combination of blade-and-gate opening for maximum runaway speed and the generators have to be designed for such possibility.

With the new type of adjustable-blade turbine and with the blades properly pivoted ahead of the center of pressure on roller bearings, the theoretical maximum runaway speed cannot conceivably be encountered. The maximum runaway-speed combination occurs at high gate opening with the blades nearly closed and may be from 2.3 to 2.5 times the normal speed, or even higher. With the blades allowed to open fully the runaway speed is held within 1.8 to 2 times normal, the overspeed for which standard generators are designed. Also the force required to move the blades is only a small fraction of that needed when the blades are not appropriately pivoted with respect to the center of pressure.

Installations

By 1934, it was considered that sufficiently satisfactory results had been obtained in the laboratory to justify an experimental field installation. Arrangements were made at that time with the Kanawha Valley Power Company to furnish a 14-ft 9-in. adjustable runner instead of a fixed-blade runner at the Marmet plant[5,6] near Charleston, W. Va. This unit was placed in operation in 1935, and has been in continuous use since that time. It is rated 7600 hp under 23 ft head at 90 rpm and has developed about 8500 hp under that head. With the exception of the runner and a few minor differences, the adjustable-blade-runner unit is identical with the adjacent fixed-blade-runner turbine rated 6600 hp. Two other turbines of essentially the same design have since been put into operation for the same company, one in 1937 at the Winfield plant[7] rated 9150 hp at 26-ft head and one in 1938 at the London plant, both near Charleston, W. Va.

[5] "Design Features of London and Marmet Hydro Developments," by Philip Sporn and E. L. Peterson, *Power Plant Engineering*, vol. 41, 1937, pp. 80–87.

[6] "Automatically Adjustable Propeller Turbine," by R. V. Terry, *Power*, vol. 81, 1937, pp. 97–99.

[7] "Design and Operating Features of the Winfield Hydro Development," by Philip Sporn and E. L. Peterson, *Power Plant Engineering*, Feb., 1940, pp. 36–42, 46.

The Marmet turbine represented a rather large step-up in physical dimensions and power from the model, especially for equipment of such radical departure from past practice. The areas of the water passage were about 115 times and the power about 300 times those of the model. A blade of the Marmet runner is shown in Fig. 7, being machined in a 120-in. lathe. The general location of the blade axis with reference to the blade area is well

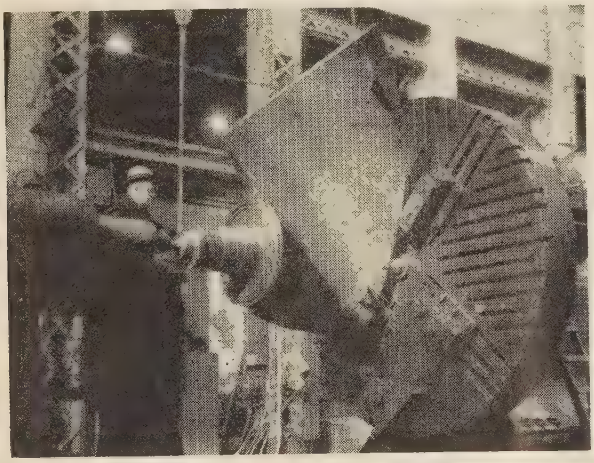

FIG. 7 BLADE FOR 14-FT 9-IN. MARMET RUNNER

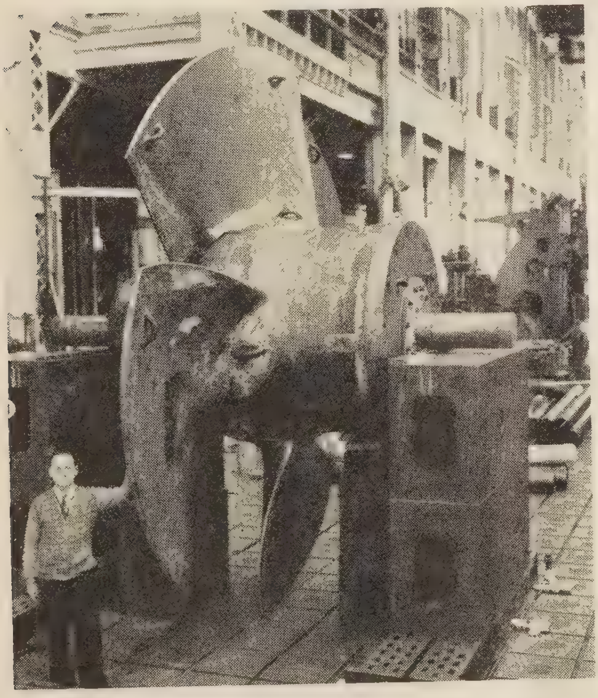

FIG. 8 BALANCING ADJUSTABLE-BLADE RUNNER FOR THE WINFIELD PLANT

illustrated. Another interesting feature illustrated is the relatively large size of the blade boss which is important in obtaining the proper strength of connection between the trunnion and blade. This is made possible by the use of an integral labyrinth seal. Fig. 8 shows the Winfield runner being balanced in the shop and Fig. 9 shows a sectional view of the Winfield unit.

With these three installations, the connection between the four blades of the runner and the balance piston was of the rack-and-gear sector type, each rack being guided in two bearings. The eight rack bearings cause some friction which creates a lag in the blade position with reference to the wicket gates and results in some differences between increasing and decreasing loads, amounting to about 10 per cent in gate opening or about 3 deg in blade position. These differences are not considered important. Field efficiency tests of those installations were not made because

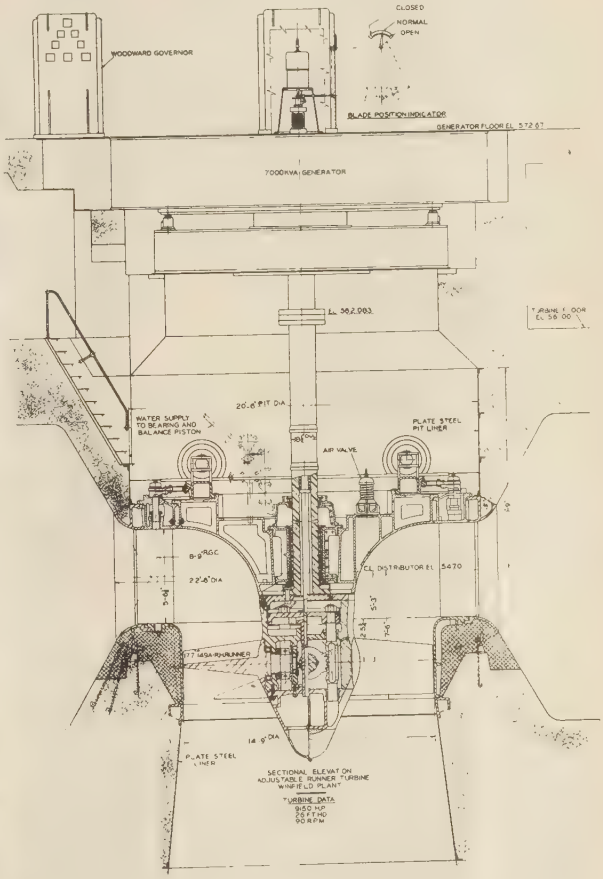

FIG. 9 SECTIONAL VIEW OF 9150-HP WINFIELD UNIT

of the difficulty of measuring the water accurately. However, the average blade position with respect to gate opening closely follows the best relation determined from the model tests. The power output plotted against gate opening approximates a straight line, which is a desirable characteristic.

The fourth installation was made during the last year at the Austin Dam plant of the Lower Colorado River Authority and consists of two units, each rated 10,000 hp, 200 rpm, 61 ft head. Preliminary field-test results show a maximum efficiency of 92 per cent.

INSTALLATION AT AUSTIN DAM PLANT

This installation may be considered typical and will be briefly described by reference to Fig. 10. The six runner blades are of cast steel with integral trunnions pivoted on roller bearings in a cast-steel hub. Each blade is pivoted on three roller bearings, two radial bearings, and a thrust bearing. The runner hub is divided into two compartments. The lower compartment, which is packed with grease, carries the bearings and blade-connecting mechanism. The upper compartment consists of a bronze-lined

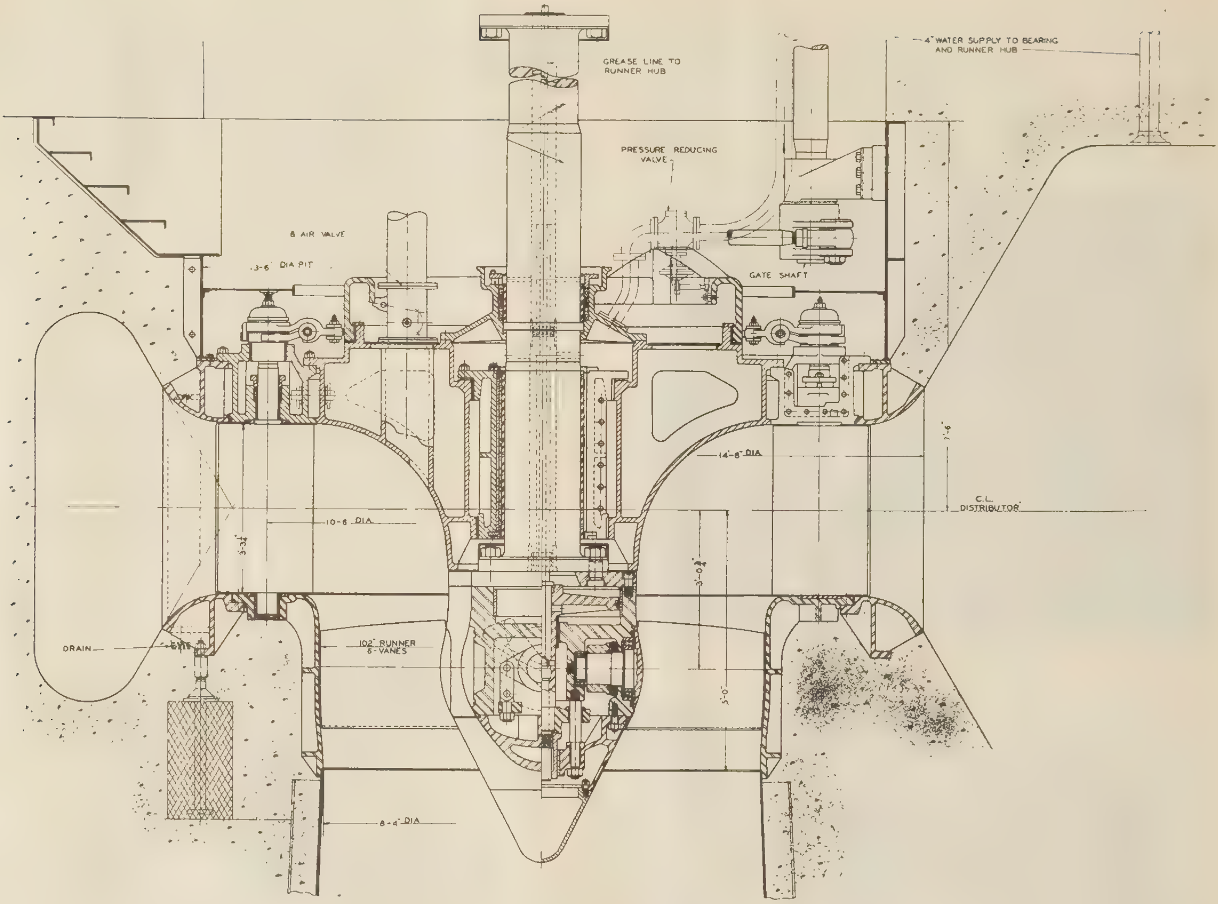

Fig. 10 Cross Section Through Austin Dam Turbine

cylinder to house the reactive piston, the integral stem of which projects downward through the lower compartment for connection to the blades by spider, links, and levers. The hub is also provided with top and bottom covers of cast steel, the top one being arranged for connection to the shaft. The stroke of the piston and spider limits the travel of the blades to 25 deg. The piston rod is guided in two bronze-bushed bearings made equal in diameter so that the reciprocation of the piston will not create a pumping action on the grease in the lower hub compartment. The space under the balance piston communicates with the draft tube through ports in the piston hub, hollow piston rod, and ports in the lower cap of the runner. Reactive pressure on top of the piston, tending to close the blades, is supplied from the headwater, through strainers and a pressure-reducing valve, to the bearing housing, thence through radial holes in the shaft and downward through the hollow shaft. This supply is combined with the water supply to the rubber bearing. The pressure supply is maintained constant by the pressure-reducing valve, the adjustment of which may be easily altered in the field as desired.

Grease is supplied to the runner hub through a pipe connected internally to the piston rod and extending upward through the generator and turbine shafts. This pipe also actuates the blade position indicator which is mounted above the generator. The top end of the pipe is provided with a swivel connection for greasing the runner hub while the turbine is running.

The runner hub is packed with grease during assembly. A heavy-bodied adhesive grease is used. It is expected that some water will enter the hub. This is, however, made difficult by the use of the heavy-bodied grease and by the use of a labyrinth-type seal at the blade bosses. The roller bearings are of the 15 to 18 per cent chromium type of corrosion-resisting steel with a Rockwell C hardness of about 55.

In addition to creating the reactive moment, the piston in the upper part of the runner hub also serves as a very effective dashpot. The inflow to and outflow from both sides of the piston must pass through restricted openings. This dashpot action helps to steady the blade movement as well as to limit the rate of movement.

The mechanism in the runners of all installations is simple and rugged. The blades are pivoted only slightly upstream from their normal centers of pressure, approximately 1 per cent of the nominal diameter of the runner. During normal operation this results in rather small moments about the blade axes, and relatively small forces on the internal mechanism, tending to reduce wear. The mechanism is, however, designed to carry forces about 5 times their normal value. This is done to provide for the larger forces which may occur during runaway speed and to take care of a hypothetical condition of having the total moments of all the blades concentrated on the mechanism of one blade.

Externally, all units have the same general appearance as those employing runners with fixed blades. The generators, governors, shaft couplings, and external mechanism of the turbines are of standard construction.

Operating Experience

The five units now in operation have been in continuous service

since installation of from 6 months to 5 years. No trouble of any consequence has occurred with any of these units. Highly satisfactory operating results have been obtained. All units govern very well. The three Kanawha Valley Power Company units are of the automatic type remotely controlled, and handle well under that type of operation.

As proved by operating experience and as may be seen from a study of the model-blade moment diagram shown in Fig. 6, the blade movement with respect to gate movement is inherently stable. For a given reactive moment and head, there is a rather definite balanced position of the blades for each gate opening. When the gates are moved by the governor, the blades become unbalanced until a corresponding movement of the blades restores the balance. The blades do not overtravel because such a movement sets up a restoring moment. For slow movement of the gates during normal governing, there is enough friction in the blade mechanism to prevent unnecessary movement of the blades. For large changes in load in either direction, the blade movement follows the gate movement rapidly with very little lag.

Perfect synchronism between gate-and-blade position, such as to result in the highest possible efficiency at all loads and heads, is not obtained. However, since blade adjustment is automatic for both head and load, the perfection of adjustment is considered adequate.

When any load up to full load is kicked off the unit, the blades close approximately as fast as the gates. This is a desirable condition of operation because, when the gates are closed, the runner tends to screw up in the water and thus to lift the entire rotor. The blades being in their flat or nearly flat position greatly reduce the lifting effect.

Lifting effect on shutdowns is also reduced by the admission of air through the crown plate. It is customary to provide each unit with two rather large air valves of different types. One is of the check type which is forced open by an adjustable cam as the gates close. That valve takes air through a pipe from the outside of the powerhouse and is also used for venting the turbine during normal operation under load at low gate openings. The other air valve is a spring-loaded check, taking air from the turbine pit, and is adjusted to open at about 15 ft of water vacuum. It opens only when the dropping of load produces a high vacuum under the crown plate.

With the air valves in operation and the runner blades reaching their closed position almost simultaneously with the gates, a highly satisfactory shutdown results. Practically no bump can be felt from the top of the turbine or generator when full load is dropped.

The turbine runners and throat rings have proved to be remarkably free of pitting from cavitation. This rather gratifying experience is attributed to several factors. The models were subjected to extensive cavitation studies. The runner blades are of simple curvature. The position of the gates and the sectional curvature of the top of the throat ring were selected so that the gates do not project over the curved part of the ring when in their full-load position. This tends to reduce inflow vortexes and to produce a more uniform velocity distribution.

Development Work in Progress

Additional studies and experimental work are in progress with a view toward further improvement and toward the extension of the upper head range. This also includes additional cavitation studies and further studies toward improvement of mechanical details.

Among the improvements in mechanical details might be mentioned the arrangement, shown in Fig. 11, for the automatic admission of free air to the draft tube at the runner cap. This is for

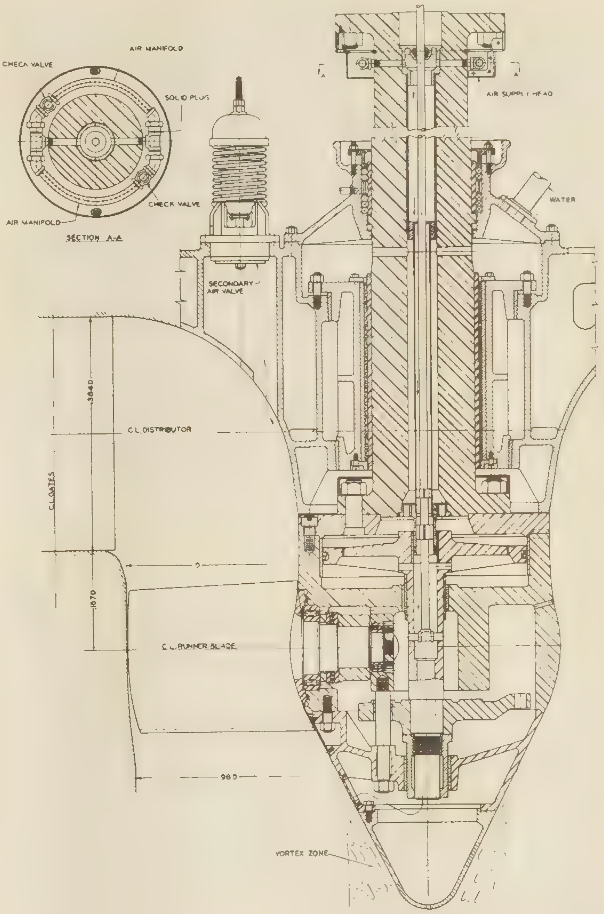

Fig. 11 Sectional View Showing Arrangement for Automatic Admission of Free Air to Draft Tube

the purpose of giving smoother operation in cases where the water leaves the runner with a considerable whirl. Such an arrangement is advantageous in preventing vibration in the case of runaway speed or operation at greatly reduced heads, high values of unit speed. Whenever a vortex forms at the runner cone in the center of the draft tube, a vacuum is produced, even with high tail water, such as to suck in free air through the tubular passages in the piston rod and turbine shaft. These passages terminate with check valves near the upper end of the turbine shaft. The check valves are adjusted so that they will admit air to the draft tube only when a strong vortex occurs at the runner cone.

Conclusion

In conclusion, a summary of the principal features of the new type of hydraulic turbine may be given as follows:

1 Blade-operating mechanism is entirely separate from the governor oil-pressure system and is controlled by the turbine water supply. No extra governor equipment is required and all oil pipes are eliminated.

2 Blade-adjusting mechanism is concentrated in the turbine proper, reducing interdependence of turbine, generator, and governor design.

3 Turbine and generator shaft couplings are of standard construction, the same as for fixed-blade turbines.

4 For normal blade adjustment, forces and moments required to actuate the blades are a minimum due to location of pivot point and use of roller bearings.

5 Blades open when starting turbine, tending to protect the main thrust bearing.

6 Blades open at runaway speed, appreciably reducing overspeed and cost of generator.

7 Use of heavy grease in hub versus heavy oil.

8 Inherent tendency for blades to adjust themselves to the condition of operation.

9 Blades follow gates rapidly on quick shutdowns, greatly reducing lifting effect and forces on the blades.

10 Provision for automatic suppression of vibration with free air in the case of overspeed.

Discussion

C. S. Adams.[8] The information contained in this splendid paper is evidence in itself of the great contribution that has been made to the progress of hydraulic-turbine design by the author. The amazing ingenuity of the inventive engineer should not be permitted, however, to overshadow the tremendous amount of mental and physical energy which has been expended by this man and his associates throughout the last 10 years in order to develop and perfect the original conception of the Newport News type of automatic adjustable-blade propeller turbine.

It was the writer's opportunity and pleasure, when serving as designing engineer for the Lower Colorado River Authority under Clarence McDonough, general manager and chief engineer for the Authority, to have been permitted to study the development of this turbine through the laboratory stage; to have viewed the first commercial installations on the Kanawha River in West Virginia; to have designed the Austin hydroelectric power plant around the two most powerful of these turbines yet built; and to have installed these two turbines and to have placed them into successful operation. Through each of these stages of development and application, the hydraulic engineer can find that this water-operated automatic adjustable-blade turbine has definitely proved a successful reality.

[8] Frederic R. Harris, Inc., Consulting Engineers, New York, N. Y.; formerly Designing Engineer, Lower Colorado River Authority, Austin, Tex.

The writer will endeavor to sketch briefly some of the items of special interest which concern the designer, constructor, and operator of power stations that embody this new type of turbine as well as to present a few brief facts relative to the Austin installation.

The invention of the Newport News type of adjustable-blade-propeller turbine has furnished the designing hydroelectric engineer with another useful tool which can be applied advantageously to assist in the solution of problems that occur in the design of hydroelectric power systems where the conservation and the efficient use of water in the system must be effected to the utmost. The Newport News turbine permits a simple, neat, compact, and relatively inexpensive installation to be provided at hydraulically proper locations in order to supply energy and power to small "off-peak" loads with a relatively high over-all plant efficiency.

To cite a practical application, each of the two Austin turbines is rated at 10,000 bhp at 200 rpm under a net head of 61 feet. This 20,000 bhp of adjustable-blade propeller turbine is an effective part of the Lower Colorado River Authority's total of 175,000 bhp which is now installed in four hydroelectric power plants on the Colorado River of Texas. In the ultimate design of this system, it was considered necessary to install the 20,000 bhp of adjustable-blade turbines at Austin so that the 155,000 bhp of Francis turbines along with over 2,000,000 acre-ft of firm water storage could be so correlated and coordinated that the over-all operating efficiency of the entire system under commercial loads could be at the highest possible value. A thorough study of the entire project with respect to hydrology, reservoir characteristics, and commercial power sales disclosed the definite necessity for an adjustable-blade installation.

The physical conditions at the Austin site rendered the installation of power machinery difficult. An unusually high tailwater condition during floods and the long length of spillway required for the dam in order to accommodate flood flows, coupled with the fact that the entire power plant had to be placed in

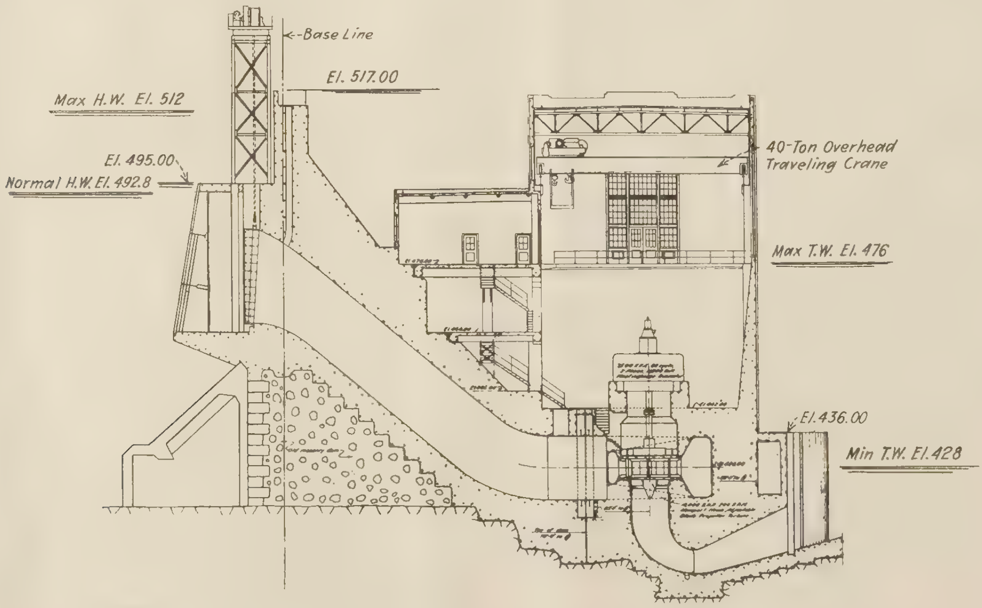

Fig. 12

the channel of this major stream, made necessary the most compact installation that could be developed. Figs. 12 and 13 of this discussion, showing the transverse and longitudinal cross-sections through the power station will enable this compactness to be observed.

The Newport News turbines lent themselves admirably to a neat and compact setting. An important factor in this particular power-plant design, in so far as the entire system was concerned, was that the demand capacity obtainable with the given diameter of runner was at a maximum with the adjustable-blade type. The design also permitted the use of 60,000-ft-lb torsional gate-shaft Woodward hydraulic governors in a pleasant-appearing cabinet form, as may be seen in Figs. 14 and 15 of the interior of the powerhouse, and a minimum of space was required on the generator-room floor and in the turbine pits for the governing installation.

The model tests on the Newport News runner revealed that both a lesser runaway speed and a lower starting hydraulic thrust existed on this type of runner than on the Kaplan type of corresponding characteristics and size. These features resulted in a saving in the electric generators together with an improvement in operating conditions. The author in pivoting each of the runner vanes slightly upstream from its center of pressure, has permitted these favorable characteristics to become innate properties of this turbine.

The incorporation of these turbines into the detailed design of the Austin power station entailed but slight additional difficulties, as compared with the Francis type or the fixed-blade-propeller type of turbines. The provision for the supply of water to the blade-actuating mechanism is essentially the only additional service that must be supplied. The governors and generators are essentially the same as they would be for fixed-blade units, although a hollow generator shaft is required for operating the blade position-indicating device atop the direct-connected exciters and for the admission of grease into the hub of the runner.

The installation of the Austin hydraulic-turbine units proceeded without delay. The assembly and, consequently, the dismantling of the units is a relatively simple operation and, since the runners were completely assembled in the shop prior to the field installation, a minimum amount of field-assembly work was required. No more working space in the powerhouse was required for erecting or dismantling these complete generating units than for comparable fixed-blade units. The governors were

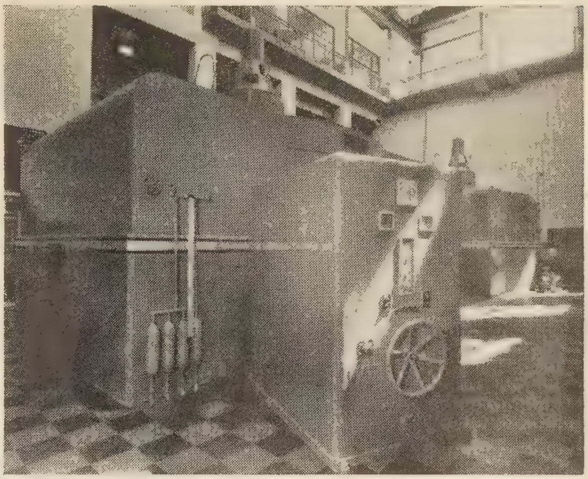

Fig. 14 Interior View of Austin Powerhouse, Showing Torsional Gate-Shaft Hydraulic Governor Cabinets in Foreground

Fig. 15 Another View in the Power Plant Showing the Two 10,000-Bhp Hydraulic Turbines

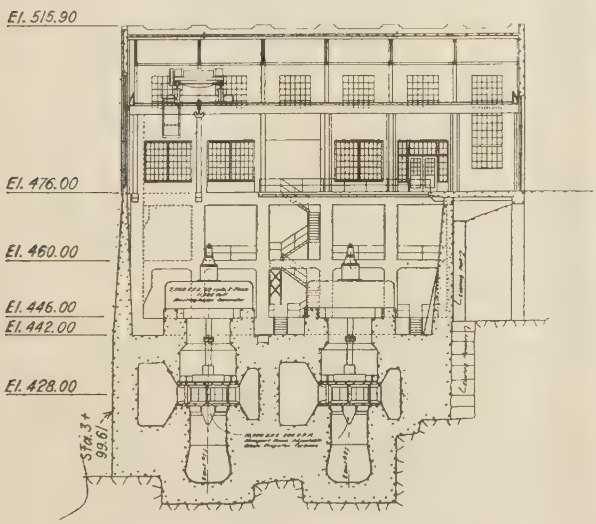

Fig. 13

shipped completely assembled in their housings, and it was only necessary to set them upon their anchor bolts and connect the servomotor connecting rods to the torsional gate shafts.

After the two units had been installed, a short operating period for each unit at about one-half rated speed enabled the Kingsbury thrust and guide bearings of the umbrella type on the Westinghouse generators and the water-lubricated rubber guide bearings of the turbines to be "run in" and to determine that the units were physically completed and also properly installed and serviced. At the end of this "run-in" period, each of the units was loaded with a water rheostat in order that it might be put through a vigorous and thorough mechanical testing before being placed into commercial operation, and in order that the operation of the runners could be observed and adjusted to insure performance characteristics comparable with those of the final laboratory model.

Load was applied to each generating unit in increments of about 1000 bhp; the unit was operated for about 5 min, then the circuit breaker was opened to remove the load; and the governor closed the regulating gates at the rate of 3 sec for a full gate stroke. As the units were started up after each run and as loads were increased to a maximum of 5 per cent above the rated capacity, it could be noticed that the blades opened almost to the fully open position, just as the regulating or wicket gates began to open, but the blades then closed slowly to the proper position. When the unit was disconnected from the bus, the blades closed at almost the same instant that the 3-sec governor closed the regulating gates. The quick-opening characteristic lessened the hydraulic thrust on the main bearing caused by the inrush of water on starting the unit, whereas the quick-closing characteristic denoted the sensitivity and fine mechanical balance of the individual vane bearings and linkage mechanisms for each of the six runner vanes of the 102-in-diam runner.

After the thorough initiation and adjustment with the aid of the water rheostat, the two units were placed in commercial operation on April 1, 1940. Although the flywheel effect or WR^2 of the Austin generator and turbine rotors is small and the flywheel effect of the rotating machinery of the connected loads is not great, the generating units governed perfectly. The units governed so well that the frequency-controlling equipment for the entire system will be moved to the Austin power plant. The Austin plant also serves as the dispatching point for the distribution of power to the Authority's customers which include the Texas Power and Light Company, Houston Lighting and Power Company, San Antonio Public Service Company, the City of Austin, 3500 miles of rural-electrification lines, and the Authority's own power district which is approximately the same area as the combined states of Massachusetts, Connecticut, Rhode Island, and New Jersey.

These two units have now been operating satisfactorily and continuously for 8 months. Outside of an occasional replacement of breaking pins on the wicket-gate mechanism, the units have functioned perfectly and on occasion have operated continuously at full rated load for a period of two weeks.

In July, 1940, each of the turbines was given a thorough preliminary field test in order to determine its performance characteristics. The Gibson method was used for determining the quantity of water used at each test point and all observations and computations were made essentially in conformance with the A.S.M.E. Power Test Code for Hydraulic Prime Movers. The units were tested between 9 a.m. and 8 p.m. with commercial loads. A reasonably flat horsepower-efficiency curve was obtained for each unit, but the author considered that the characteristic curves could be improved to conform more closely with the model curves by making a minor alteration to the pressure-regulating device which actuates the servomotor piston of the vane-position mechanism. This minor alteration has now been made and the load versus blade-angle relationships for each Austin turbine corresponds with those of the homologous adjustable-blade model over the entire range of load. The turbines will be given a final test in the near future in order to determine their performance curves.

The maximum over-all efficiency of one turbine, as determined from the preliminary field testing, occurred at 85 per cent of the rated capacity at a value of 92 per cent, whereas, the other turbine attained a peak efficiency of 90 per cent at 85 per cent of the rated capacity. The mechanical operation, the balance of the units, the functioning of bearings, and the operation of the complete generating units, as revealed during the field tests are all considered at or even above par with the other modern hydraulic-turbine installations at the Authority's power plants.

With the development, success, and the experience gained by the author in the design, manufacture, installation, and operation of the existing automatic adjustable-blade turbines, the hydraulic-turbine field should now be clear for the production of larger-capacity wheels of the same type. It is anticipated, moreover, that the talents and energies which have been utilized in the development of this successful invention will enable further advances to be made in the hydraulic-turbine as well as in related fields.

F. Nagler.[9] The author deserves a great deal of credit, as much for initiating a new policy of engineering presentation as for the extremely interesting type of turbine development indicated. In past years, few engineers were either willing or free enough from commercial considerations to be entirely open with engineering uncertainties, field troubles, and the like, to permit that frankness on which the most rapid progress may be realized.

The magnitude of the actual installations described in this paper is particularly noteworthy. The writer and J. F. Roberts did some work on automatically adjustable blades during the early '20's. This work attempted to get around the influence of variable friction, by hanging the blades on knife-edges instead of letting them revolve in the bearings. The idea was that they would be positioned by a combination of centrifugal force and water flow. The work was discontinued because of assumed insurmountable obstacles connected with weakness of structure and the wide variables introduced by changing friction, by variable-velocity conditions as the head changed, and, of course, by the variable velocity and direction imposed by guide-vane operation. The author, apparently, has worked out a feasible solution of these major difficulties.

It would be appreciated if the author would comment a little further on the life of the antifriction bearings on the runner-blade pivots. We know that tremendous forces exist on these blades; forces, as pointed out in a paper[10] by J. D. Scoville, of a magnitude sufficient to lift the entire rotor of generator and turbine. These forces come practically as a blow, Fig. 9[11] in the Scoville paper.

In a current paper[12] H. Styri brought out the point that, to insure commercial life of any ball or roller bearing, there must be an oil film present between roller and race. It is probably not particularly difficult to maintain such a film for a bearing that rotates, but the bearings in the hub of the runner are practically stationary throughout their life. The writer would expect certainty of breakdown of the oil film and inevitable peening action under such conditions. Undoubtedly, the field experience to date is the most effective answer to a question as to life of antifriction bearings under this duty. Further comment from the author on this point would be appreciated.

When a piston interconnecting and influencing the position of the blades is added, as shown in Figs. 9, 10, and 11 of the paper, and external control of the pressure behind that piston taken outside the shaft, are we not approaching closely an externally adjustable blade, the connection of which to the guide-vane motion would require a relatively minor addition in the form of links or cams?

It would be exceedingly interesting, if the author would show a comparison between the shape of the efficiency curves of the automatically adjustable-blade construction, and the Kaplan and the fixed-blade types.

[9] Chief Engineer, Canadian Allis-Chalmers, Ltd., Toronto, Canada. Life Member A.S.M.E.

[10] "Speed Regulation of Kaplan Turbines," by J. D. Scoville, Trans. A.S.M.E., vol. 63, 1941, pp. 385–394.

[11] Ibid, Fig. 9, p. 389.

[12] "Friction Torque in Ball and Roller Bearings," by H. Styri, *Mechanical Engineering*, vol. 62, no. 12, December, 1940, p. 886.

The use of air valves, as applied to this type of construction, is of particular interest to the writer since, at the time of presentation of this device,[13] but slight attention was paid to its utility. Several major field accidents would have been prevented had this device been adopted more generally. This is particularly true on the axial-flow types of turbines, where such positive uplifts as are reported by Scoville[10] are encountered. There are some instances reported of uplift with the Francis turbines, but the writer has not personally observed them. It would be of interest if the author would comment on this particular feature.

J. F. Roberts.[14] The description in the paper of the various tests and model and field experiments which have gone into developing the automatic adjustable-blade-type propeller turbine is both interesting and enlightening. The results should be very gratifying to the author and should result in a real saving to the users of this type of turbine.

The author's remarks concerning the use of air valves are interesting. The writer has always been a strong advocate of ample size of air valves, both for fixed- and adjustable-type propeller turbines and for Francis turbines. The point where the air is admitted to propeller-type runners has been found quite important and, in several cases, a change in the point of air admission has resulted in a material improvement. The writer has found that, if the air is admitted to a propeller turbine at the lowest point in the head cover, just above the top of the runner hub, the greatest benefit can be secured. In two cases it was found possible to get air into the turbine at the lower point when it would not take in air near the upper part of the head cover, due to a positive pressure existing at that point.

It has also been found unnecessary to use a cam or linkage to open the air valve at definite gate openings and close it at other gate openings since, both on Francis- and propeller-type turbines, the air is apparently beneficial whenever a vacuum exists which is sufficient to draw the air into the turbine. By providing the air valve with a light spring-loaded check valve so that it will close at zero pressure and prevent water from flowing out of the valve, but will open whenever a vacuum of 1 or 2 in. of mercury exists, the use of cams connected to the gate mechanism can be eliminated and better results obtained under both varying heads and varying gate openings.

J. D. Scoville.[15] The author's turbine is called an automatic adjustable-blade propeller, the blades moving to a new position as the gate opening is changed because of inherent characteristics of the runner. This is distinguished from the ordinary Kaplan turbine in which the blades are moved by an oil-operated servomotor, controlled by the gates. This might be called a controllable-pitch propeller.

There are certain differences between the two types which should be pointed out. The author states that, when the gates are opened on the automatic adjustable-blade turbine, the blades go to a large angle which make starting easy. On the controllable-pitch propeller, the blades are normally in their flattest position on starting, so that one would expect a large gate-opening requirement to start the unit. Such is not the case. Usually 10 to 15 per cent is sufficient. The unit reaches synchronous speed at about the same opening.

As the gates open further on the automatic adjustable-blade propeller, the blades follow. When the gates open on the controllable-pitch propeller, a cam on the gate shaft or connected to the gate servomotor controls the blade position. The shape of this cam is determined by index tests in the field so that the blades are moved positively to the best position. As the head changes, a new blade-gate relationship is required. This alteration can be taken care of by an adjustment in the cam position which requires only a minute or so, the turbine remaining in normal operation.

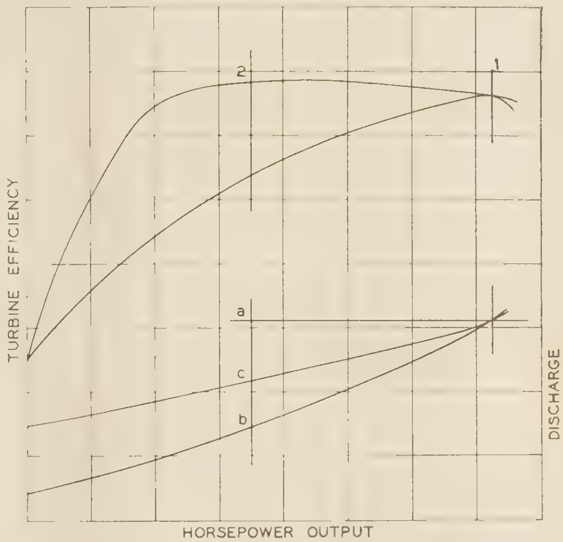

Fig. 16 Curves Showing Relation Between Load Changes and Discharge of Controllable-Pitch Propeller

The blades of a controllable-pitch propeller can be moved as fast as the gates for load changes. It is desirable, however, to restrict the closing of the blades. The reason will be evident from Fig. 16 of this discussion. If the load changes from 1 to 2 the discharge changes from a to c, if the blade-closing time is very slow, but from a to d if the blades close as fast as the gates. Therefore, the pressure change is correspondingly less. This is an advantage in a plant with a relatively long penstock. Usually the controllable-pitch propeller is so adjusted that the blades close in about 60 sec, so that, during a load reduction, it is in effect a fixed-blade runner.

If the controllable-pitch propeller runs away, the maximum theoretical speed will be reached with the blades in a relatively flat position and the gates wide open. It is only possible to obtain this condition if the blades are deliberately blocked at the flat angle. Even if the oil pressure fails and the gates open for some reason, the blades will open because of the unbalance, which the author points out. The opening tendency of the blades on the Bonneville units at runaway speed was about 4 times the force required to open the blades at normal speed. This means that the runaway speed of the controllable-pitch propeller cannot reach the theoretical maximum.

R. E. B. Sharp.[16] The Terry automatic adjustable-blade-propeller turbine presents a definite and valuable contribution to turbine design and to date turbines of this type have given a satisfactory account of themselves. The study and experiments, which the author has made on the hydraulic moments acting on

[13] "Operation of Hydro-Electric Units for Maximum Kilowatt Hours (The Turbovent)," by F. Nagler, *Engineers and Engineering*, vol. 42, 1925, pp. 148–156.

[14] Principal Mechanical Engineer, Tennessee Valley Authority, Knoxville, Tenn. Mem. A.S.M.E.

[15] Assistant Chief Engineer, S. Morgan Smith Company, York, Pa. Mem. A.S.M.E.

[16] Chief Engineer, I. P. Morris Department, Baldwin Southwark Division of The Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

runner blades, represent a valuable contribution in connection with the design of the adjustable-blade-propeller turbine, whether of the Terry type or of the usual Kaplan adjustable-blade-propeller turbine.

There is one factor, however, which the author does not mention, but which does have an effect on the net hydraulic moment acting on runner blades, and that is the centrifugal force acting on the blades.

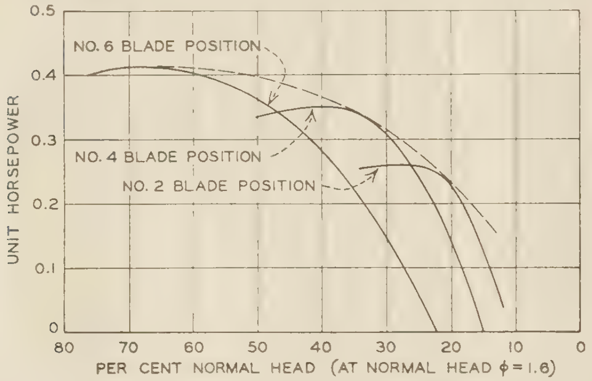

Fig. 17 Relative Power Developed by Kaplan and Terry Designs at Less Than Normal Head

Referring to the plan view of the author's Fig. 5, all particles in the blades are acted upon by centrifugal force which has a component tending to flatten them. The moment caused by this component is of course greatest at runaway speed. However, the net force mentioned by the author, namely, that resulting from a reduction in the angle of attack and that resulting from the increase in the relative velocity, increases at a greater rate than the centrifugal-force moment. The result is that, with this design of blade, the net moment is greatly increased, tending to open the blades.

When adjustable-blade-propeller turbines are operated at less than normal heads, the Kaplan type of turbine, with the blade-gate relation positively established by a cam, develops greater power than with the author's design; Fig. 17 of this discussion indicates this condition.

The curve marked No. 6 blade position represents the maximum power that can be developed in the steepest blade position for this particular runner. Similarly the curves marked No. 4 and No. 2 blade positions represent the maximum power which can be developed with the blades set in these flatter positions. It is noted that at 30 per cent of the normal head, for instance, the unit power developed for the No. 6 blade position is 0.15, whereas, for a flatter blade position, as read from the envelope, the maximum unit power is 0.315. This condition approaches the runaway-speed condition where the author's blades tend to take up their steepest blade position or angle. In order, however, to develop the maximum power possible from the unit, it is seen that the blades should be definitely brought to a flatter position.

It is the writer's experience, based on comparative tests, that a better performance as regards cavitation and maximum output obtainable are secured by having the blade axis located farther downstream than is indicated in the paper. This is due to reduced clearance between the throat ring and the runner blades in the vicinity of the trailing edge being secured with the axis so located. While this does increase the capacity required of the blade servomotor, it is considered desirable.

The greatest factor in determining the capacity of runner-blade servomotors of Kaplan turbines is the friction in the runner-blade mechanism in the hub. A minimum amount of friction is therefore quite desirable on that type of turbine and the use of antifriction bearings in Kaplan hubs would of course accomplish this result to a very great degree. The inevitable entrance of water into the hub at some time or other, however, dictates that these bearings should be of the stainless-steel type, such as the author employs. The slow oscillating movement of the runner blades with a turn of not more than 30 deg, with indefinite more or less stationary periods in conjunction with the live character of the load, tends to cause Brinelling or local strain hardening or grooving action on the races which, in the writer's opinion, has made the use of this type of bearing questionable, particularly in view of the present inability of bearing manufacturers to obtain a satisfactory stainless antifriction-bearing material.

The author has naturally employed bearings of ample proportions and liberal ratings. However, the writer would be interested in knowing whether any of these bearings has been examined after extended field use and whether the lag mentioned in connection with the Winfield turbines can be ascribed to possible grooving of the races. The continued absence of friction is of course essential to the Terry design as a friction increase aggravates the lag between the exact position desired and that actually secured.

One of the interesting features of this turbine, as the author points out, is the tendency of the blades to open to their steepest angle at runaway speed, with consequent reduction in the runaway speed reached, as compared with what would be reached with a flat blade angle. Under normal conditions, the blades both of the Terry turbine and the conventional Kaplan turbine move to their steepest blade positions if the gates open wide under runaway-speed conditions.

It is a common practice among Kaplan designers to assume that the cam relation might be destroyed and the runner blades might conceivably be in the flat position with the maximum gate opening causing high runaway speed. For this to happen however a very remote chain of circumstances would have to exist. Nevertheless, it is considered sound engineering to assume that they might exist.

It appears to the writer that the mechanism of the Terry turbine might prevent the blades from taking up their steepest position during runaway, due possibly to the jamming of one of the blade links or other parts of the hub mechanism, or the malfunctioning of the pressure-reducing valve, controlling the servomotor pressure. Therefore, with the Terry turbine no less than with the Kaplan, it appears desirable to allow for the maximum possible runaway speed with the flat blade position, in view of the possible destructive results should this condition not be provided for.

Author's Closure

Several of the discussers raised questions regarding the power-efficiency characteristics of this type of turbine. The Austin dam units were field-tested in January, 1941, according to the standards of the 1938 A.S.M.E. Test Code for Hydraulic Prime Movers. The discharge was measured by the Gibson method. The results of these tests on one unit are shown in Fig. 18 of this closure, in comparison with the results of tests made at several blade positions on a $16^1/_2$-in. model under 12 ft head in the hydraulic laboratory of the author's company. It will be noted that the efficiency curve is quite flat, being above 90 per cent for about 62 per cent of the load range. The maximum efficiency obtained was 93 per cent. The average efficiency from the 20 individual test points, within the guaranteed range of from 4000 to 10,000 hp, was 91.6 per cent. The author believes that these test results are at least equal to any that have been obtained with turbines of the adjustable-blade type.

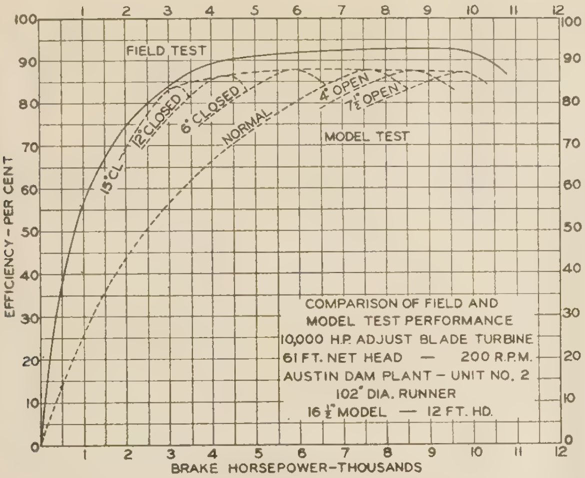

FIG. 18 COMPARISON OF FIELD AND MODEL-TEST PERFORMANCE OF 10,000-HP ADJUSTABLE-BLADE TURBINE

Mr. Adams' discussion, relating to the installation and operation of the Austin turbines is gratifying, particularly that part dealing with governing and system-frequency control.

Messrs. Nagler and Sharp raise questions in regard to the service and life of the stainless-steel roller bearings employed in the runner hub. The estimated loads, the bearing manufacturer's ratings, and other data on the several bearings of the Austin runner are given in Table 2.

TABLE 2 AUSTIN RUNNER ROLLER BEARINGS

	Outer radial	Inner radial	Thrust	
Bore of bearing, in.	9.00	5.00	8	
Outside diameter of bearing, in.	14.50	8.25	12	
Bearing length, in.	2.75	2.19	2	
Diameter and length of rollers, in.	$1^1/_8 \times 1^3/_4$	$^3/_4 \times 1^1/_2$	$^3/_4 \times 1^3/_8$	
Number of rollers	26	21	26	
Estimated load, lb	81,000	51,000	30,500	(Normal speed)
			122,000	(Runaway speed)
Rated load (2 rpm), lb	109,500	55,400	125,900	

The rated loads are based on a speed of 2 rpm, whereas, the actual service approaches a static condition under which the ratings would be somewhat increased. The ratings are also based upon a specified minimum of 52 Rockwell C hardness, whereas, the actual hardness is about 55 Rockwell C. That change in hardness increases their actual capacity from 70 to about 90 per cent of the capacity of carbon-steel bearings, an increase of about 28 per cent.

The life of the bearings for this service is not definitely known. Although the aggregate service of the five units installed is about 15 years, there has been no occasion to dismantle a runner for an examination of the internal parts. If the life of the bearings should prove less than expected, bearings of 25 to 30 per cent higher capacity are now available, interchangeable with the present bearings. That change in capacity would give a theoretical increase of life of from 50 to 75 per cent. A still greater factor of safety or longer bearing life may be obtained by the use of "quill-type" inner bearings or by employing a slightly larger diameter of runner hub. To date, there has been no indication pointing toward bearing failures.

The seventy-two bearings used in the five units in operation were all supplied by one bearing manufacturer who has cooperated in making this new application practicable. The materials used are of a commercial quality which appears to be quite satisfactory. The service required of the bearings is, in a way, somewhat crude as compared with that ordinarily required of bearings on high-speed machinery. It is the author's opinion that the bearings for this service will prove satisfactory for a much longer period, proportionately, than with most applications of antifriction bearings.

It would appear that roller bearings are inherently more suitable for adjustable-runner application than are the plain bronze-bushed bearings. Since the axial length of the roller bearings is short, the loads are uniformly applied and the unit loads may be readily computed. With the much longer plain bearings, the already high average unit pressures are greatly increased by pressure concentrations due to deflection. While seizure of the plain bearings has not been common, a number of seizures have occurred, which condition constitutes somewhat of a problem. It is believed that any problems which may arise with the roller-bearing applications can be more readily solved than in the case of plain bearings.

The waterproof grease used in the runner hubs is extremely tenacious, which, together with the slight angular movement of the bearings, should insure retention of a good lubricating film on the rollers. Reverse loads occur on the radial bearings when the wicket gates are closed with the turbine running. However, these loads are kept below the normal loads by the admission of air as described in the paper. The reverse loads do not occur as a blow, but rather change somewhat rapidly as the gates are closed from the speed-no-load to the fully closed position. The time interval is short but it is ample to prevent any action which might be termed a blow.

In general, the author agrees with the comments of Messrs. Nagler and Roberts in regard to venting turbines by air admission at certain gate openings and on shutdowns. Considerable venting of all types of reaction turbines is highly beneficial at low gate openings. Some venting of all fixed-blade turbines at intermediate loads up to nearly the point of best efficiency is also beneficial. From the standpoint of efficiency, venting at and beyond the point of best efficiency is ordinarily questionable, but a moderate amount of air is often used to alleviate the effects of blade and vortex cavitation. Venting becomes of increasing importance at values of the peripheral coefficient ϕ above its best efficiency value, as when operating under reduced heads. With adjustable-blade runners at normal values of ϕ a pressure usually exists under even the lower part of the head-cover cone over practically the entire load range, that is, down to very low gate openings, making it impractical to admit atmospheric air. A pressure lower than atmospheric may sometimes be produced by employing lips upstream of the vent openings. On future turbines, it is proposed to employ a scheme for venting the draft-tube vortex in a manner similar to that shown in Fig. 11 of the paper. The use of a small amount of compressed air has been found beneficial on certain turbines to alleviate blade or blade discharge eddy cavitation.

As Mr. Scoville points out, it is not absolutely necessary to open the blades when starting up an adjustable-blade turbine. However, exceptionally large gate openings, with resulting high thrust loads, are sometimes required to start a unit with the blades flat. The opening feature is certainly a distinct improvement. Cases have come to the attention of the author where the minimum angle of adjustable turbine blades had to be increased on account of starting conditions, particularly on low-head installations. It is also of interest to note in this connection that some of the largest Kaplan turbines in operation have been provided with special extra equipment to give the blades an initial tilt before starting. The 18,000-hp turbines at the Vargon plant in Sweden, with runners 26 ft 3 in. diam, are provided with such equipment,[17] also the 48,000-hp turbines at the Pickwick

[17] "Low Head Produces High Capacity," by George Willock, *Power*, December, 1939, pp. 74–76.

plant[18] in Tennessee with runners 24 ft 4 in. diam. The type of turbine described by the author performs this function automatically without extra equipment, due to the method employed in pivoting the blades.

Mr. Scoville states that Kaplan-type turbines are usually arranged so that the blades close in about 60 sec, and claims this to be an advantage in a plant with a relatively long penstock, the advantage being a lesser pressure rise in the penstock as a result of dropping load. Upon first thought there would appear to be a considerable difference as cited. However, referring to Mr. Scoville's Fig. 16, the time required to drop from full load to one-half load would be reduced in the case of the Kaplan turbine by nearly the same ratio as the reduction in discharge, so that the rate of reduction in discharge per second, which determines the pressure rise, would be only slightly less for the Kaplan than for the automatic adjustable-blade type. In each case it is assumed that the loading and gate timing is the same. In the case of full-load rejection, the total change in discharge is made in the same length of time. Consequently, the average pressure rise would be approximately the same in either case. An exact analysis of the conditions will reveal that, with the automatic type, the rate of reduction in discharge may be slightly higher at the higher gate openings but that the rate reduces somewhat as the gates approach their closed position, resulting in a very desirable action, similar to that produced by cushioning the closing end of the gate stroke.

In every case investigated by the author, the operating characteristics of the automatic adjustable-blade-type turbine, which affect governing, are equal or better than those generally obtained. In this connection, reference is made to the discussion by Mr. Adams in which it is pointed out that the 10,000-hp Austin turbines will be used for the frequency control of a 175,000-hp system, due partly to their excellent governing properties. The WR^2 of each Austin unit is only about 3 per cent of the total of all units of the system. The penstocks at Austin are about as long as any usually encountered with adjustable-blade turbines. From tests, the pressure rise when dropping full load in 3 sec was about 34 ft and the speed rise about 31 per cent.

The necessity for designing the generator for the highest possible runaway speed that could occur, if the blades were deliberately blocked at a relatively flat angle, is somewhat controversial. This question was brought up by Messrs. Scoville and Sharp. At full runaway speed the author's analysis shows that the forces on a blade are principally in the form of a couple, consisting of upward forces near the leading edge and downward forces, of only slightly greater magnitude, near the discharge edge. If that analysis is correct, the couple moment would be the same, irrespective of the location of the axis. That is, there would be a very strong tendency for the blades to open with either the old or new location of the blade axis. However, there are two features of the automatic type which make it inherently less likely for the blades to stick in their highest speed position: (*a*) The friction moment of the roller bearings used is only a small fraction of that of plain bearings; (*b*) a low operating fluid pressure is used and the arrangement is such that the fluid cannot become "locked in," as it could in the case of a piston-type valve, such as is employed with the high oil-pressure type of operation.

Referring to Fig. 17, Mr. Sharp seems to have confused the runaway-speed characteristics of the automatic-type turbine with its operation at normal speed under reduced heads. Consequently, his conclusions in regard to the advantages of a Kaplan turbine under subnormal heads are incorrect. High unit speeds, high values of peripheral coefficient ϕ, may be produced either by overspeed at high heads or by low-head operation at normal speed. It is only in the case of speeds above normal that the blades of the automatic turbine open to reduce the amount of overspeed. When operating at normal speed under reduced heads, the more or less constant pressure applied to the reactive piston causes the blades to assume flatter positions, with respect to gate opening, than their positions at the higher heads. This may, perhaps, be better visualized by stating that, as experimentally determined, the "reactive moment" required for each of the developed designs of the automatic turbine at variable heads and loads is approximately equal to the cube of the runner diameter, times the square of the speed, divided by a constant, $Rb = D^3N^2/K$. Thus, for variable-head operation at constant speed, the application of a constant reactive pressure tends to make the gate-blade relation vary correctly with the head as well as with the gate opening, in such a way as to maintain both efficiency and power at values approaching those theoretically possible. As a result, the turbine may operate to produce power down to a very low head, the same as is accomplished with the Kaplan type by changing cams, until the value of ϕ equals the highest value obtained with the blades nearly closed. That value may be from 2.3 to 2.5 times its normal value, or even higher, depending upon the number and camber of the blades.

Mr. Sharp also appears to have found from his model tests that a better shape of efficiency curve is obtained as a result of pivoting the blades further downstream than is necessary with the automatic type. The author does not find this to be the case. The results of the Austin tests, Fig. 18 of this closure, rather indicate that there is little increase in the leakage losses through the increased clearances at the periphery of the blades as they approach their open position. The reason for this apparent difference in characteristics may possibly be due either to a difference in camber or to a difference in the shape of the throat ring. The effect of increased clearance may be offset, in the author's opinion, by giving the blades of adjustable runners a little more camber than for fixed-blade runners. With the proportions of throat ring and blade-axis location adopted for the Austin turbines, Fig. 10 of the paper, the maximum clearances of the runner with the throat ring, for the open position of the blades, are kept to quite reasonable values. The throat, minimum diameter of the water passage, is located well below the blade axis, opposite the discharge edge of the blades when open. The clearance at that point is about 2.5 per cent of the runner diameter, i.e., about the same as is used with Kaplan turbines. The corresponding clearance at the leading edge of the blades is about 1 per cent, somewhat smaller than is customary with Kaplan turbines. The smaller clearance at the leading edge should have an advantage in decreasing the danger of a trash jam at that point.

Mr. Sharp has properly called attention to the effect of centrifugal force in giving the blades a tendency to move to their flattest position. This effect is more pronounced with the wider blades, such as are employed with the four-blade types, but the resulting closing moments about the blade axes are not large when compared with the hydraulic moments. In the case of the automatic-type turbine, it simply has the effect of reducing slightly the required reactive pressure.

Mention was also made of the blade-servomotor capacity. It is interesting to compare the capacities required with the Kaplan and automatic types. The author's analysis of several large Kaplan installations shows that the average required blade-servomotor capacity, based on piston displacement and supply pressure, may be expressed by the formula, $S = 20\, PN_s^{1/4} \div \sqrt{H}$, where S is the servomotor capacity in foot-pounds, P the brake horsepower of the turbine, N_s the specific speed, and H the operating head. For the author's type turbine

[18] "A Technical Review of the Pickwick Landing Project," Technical Monograph No. 40, Tennessee Valley Authority, March, 1939, exhibit 21.

a servomotor capacity of 3.33 $PN_s^{1/4} \div \sqrt{H}$, is quite ample, or one sixth that of the Kaplan. The Kaplan servomotor is double-acting, while that of the automatic type is single-acting, resulting in a true capacity ratio of 12 to 1, on an energy-supply basis.

The Austin turbines were originally provided with a cam-operated auxiliary-control device to vary the pressure supply to the balance piston, if that was found necessary. The units were at first operated at 20-psi reactive pressure without the use of that device. However, it was found from the preliminary tests that the use of the auxiliary control would result in a substantial improvement in efficiency at certain loadings. The official tests were made with the auxiliary-control device in operation, the actual reactive pressure varying from 14 to 35 psi. Such a device seems to be desirable, particularly for the higher head applications of the automatic type where the greatest variation in the required reactive moment was found from the model tests. The auxiliary control of the reactive pressure is rather simple in design and its use does not in any way affect the several basic advantages of the automatic adjustable-blade-type turbine.

Several important improvements in designs have been made since the paper was prepared. These include an external control valve for varying the reactive pressure, in contrast with the internal valve used at Austin. This valve is operated by an adjustable cam on the gate-operating ring and has a follow-up connection for blade movement. It acts to position the blades positively in accordance with a predetermined gate-blade relation, irrespective of the friction of the runner mechanism and irrespective of the variations in the blade hydraulic moments with load and head that were mentioned in the paper.

Design details have been prepared for a unit rated 40,000 hp 120 rpm 70 ft head to operate under heads varying from 50 to 80 ft. The latter head is at present considered to be the upper limit for the application of the automatic-type turbine.

In closing, the author wishes again to recognize the contributions of the discussers. He also wishes to thank the personnel of the Kanawha Valley Power Company and the Lower Colorado River Authority who cooperated wholeheartedly and to whom much credit is due for the successful applications of this type of turbine.

Production of Seamless Tubes by Combined Effects of Cross-Rolling and Guide Disks

By W. Trinks,[1] Pittsburgh, Pa.

From the various methods of producing seamless tubes the author has selected the Diescher elongator mill as the basis for discussion in this paper. For a better understanding of this method of tube production, the most recent Diescher mill installation at Allenport, Pa., is described in some detail. Following this the theory of cross-rolling and guide disks in the process of tube manufacture is explained.

Among the various seamless-tube-manufacturing processes, the Diescher elongator holds the center of interest because it materializes the long-unrealized dream of the Mannesmann brothers to produce a finished tube by cross-rolling.

In this connection, it will be remembered that, in 1885, the Mannesmann brothers of Remscheid, Germany, introduced their epoch-making invention whereby the production of seamless tubular blanks was made feasible by cross-rolling. This practice is universally known among English-speaking technicians as "cross-roll piercing."

Great hopes were originally expressed for the new rolling process. It was even expected by some that a finished tube could be made in one pass from a round ingot. Not only did this expectation fail of realization, but the use of cross-rolling in the desired heavy reduction of both wall thickness and diameter met the same fate. The cross-roll-piercing process did, however, most spectacularly and effectively deliver comparatively thick-walled blanks, which to be converted into finished tubes required some yet-to-be-discovered procedure. Finally, after many discouraging highly expensive, and time-consuming endeavors, a step-by-step forging process was invented by the Mannesmanns which accomplished the desired tube-forming operations.

This second procedure involves what is universally known as the "pilger mill," which received its name from the fact that the blank passed through the mill with a motion similar to that of the pilger (pilgrims) who, to demonstrate the depth of their piety, went to the shrine at Amdernach five steps forward and three steps backward.

Early Pilger or Poke Mills

In the United States, the early pilger mills were called "poke mills," which term arose from the fact that these early mills were hand-fed and required the operator to push forward or poke the mandrel with the shell on it into the mill after each roll stroke of the bell-shaped pass machined in the rolls.

Those who are familiar with the early history of the production of seamless tubes in the United States will remember that, in the piercing mill, Stiefel, a Swiss engineer employed at the British Mannesmann Works, substituted overlapping truncated conical members, provided with sidewise working faces, for the barrel-type cross-rolls of the Mannesmanns. When Mr. Stiefel emigrated to the United States, he introduced this highly creditable innovation. During his many years of fruitful life in this country, he was regarded as the dean among seamless-tube experts. Introducing slippage between the rolling faces and the billet permitted the material to flow into length more easily, whereby a reasonably thin-walled blank was produced. To be converted into a product suitable for cold-drawing, this blank required but a few passes in a plug mill. To be converted into an hot-finished product required, in addition to the plug-rolling, a reeling and then a sizing operation. Thus, methods other than cross-rolling had to be used by the Mannesmanns as well as by Stiefel for bringing the pierced blank to final cross section and length.

Almost 50 years elapsed after the Mannesmanns made hollow blanks by cross-roll piercing, before Samuel E. Diescher of Pittsburgh succeeded effectively in elongating a pierced blank into a thin-walled tube by cross-rolling. As in the case of the Mannesmanns, his mill acquired a name descriptive of the operation performed, as interpreted by men skilled in the tube art; this term was "elongator." The elongator has been briefly described in the literature, but the details of the construction, as well as the theory underlying this new method of plastic deformation, have never been published.

In reviewing the rather extensive patent structure which has thus far appeared in public print, the author has discovered that many of the later features of the process are inventions of the originator's brother, August P. Diescher. Therefore, as was the case with the Mannesmanns, here again we have the work of brothers.

Diescher Elongator Method

The Diescher elongator method requires cross-rolls arranged in a manner generally similar to those of Mannesmann. A pierced blank enters the mill on a freely floating mandrel on which it is

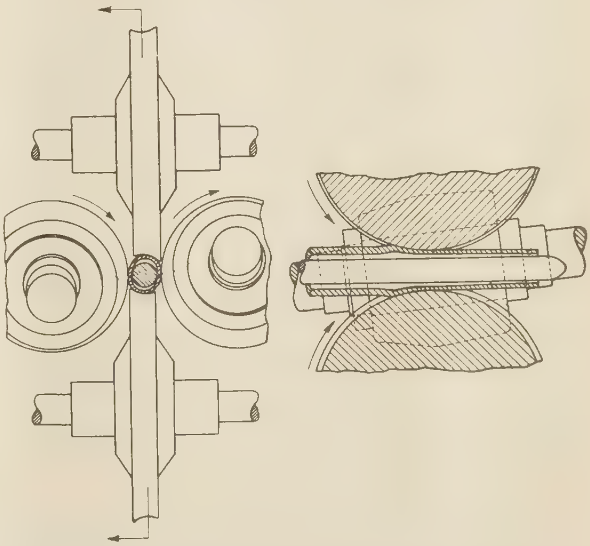

Fig. 1 Diagram of Cross-Rolling Process

[1] Professor of Mechanical Engineering, Carnegie Institute of Technology. Mem. A.S.M.E.

Contributed by the Metals Engineering Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

then cross-rolled, Fig. 1. The feature of the mandrel, moving in the forward direction during the procedure, seems to be a departure from the Mannesmann concept. However, that feature alone would not have made the Diescher procedure feasible. To explain this requires first of all a clarification of what occurs in cross-roll deformation.

The cross-rolling process is inherently an expanding process, because the contact area between roll and blank is long in the direction of tube travel, and is short in the direction of roll travel. As a consequence of the laws of plastic flow, the resistance to "bulging" is much smaller than the resistance to elongation of the product. As a matter of fact, solely by cross-rolling on a floating mandrel there could be expected at best, under the extent of deformation performed by the elongator, a tube of uncontrolled expansion and of much larger diameter than that of the pierced shell. In the Diescher mill, this expansion is prevented by the elongator disks, which by frictional contact with the blank not only prevent uncontrolled expansion, but also pull the blank forward and convert the expanding tendency of the cross-rolls into elongation of the tube. With any circumferential speed of the cross-rolls and any practicable angularity of the setting thereof, a tube issuing therefrom must travel at a much lower rate of speed than the circumferential speed of the cross-rolls. The inference is natural that the elongator disks should run at a speed but slightly higher than that at which the work material issues, so that the friction will just pull the tube blank along, without doing more work than necessary. However, that inference is incorrect. The circumferential speed of the elongator disks is required to exceed greatly even the circumferential speed of the cross-rolls.

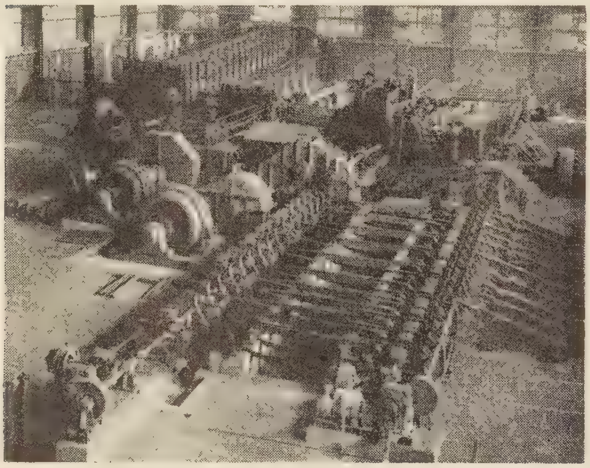

Fig. 2 Elongator Unit at Plant of Pittsburgh Steel Company, Allenport, Pa.

Moreover, there is likely to be drawn another erroneous conclusion. It might be reasoned that, since the elongator disks are intended to prevent tube expansion, they should be set to prevent any and all expansion, so that the finished tube will hug the mandrel tightly. In reality, the work material is allowed to expand away from the mandrel, at the same time, however, requiring precision control of such expansion. Both of these somewhat surprising facts merit discussion. First, however, a detailed description of the equipment will be given to aid in a better understanding of the entire subject.

Details of Latest-Type Elongator Unit

Fig. 2 shows the elongator unit in use at the Pittsburgh Steel Company's plant at Allenport, Pa. This installation is the most recent to be placed in operation in the United States. Also, the Allenport works is the only plant throughout the entire industry at which there may be seen both the Mannesmann and the Diescher practices. The plant equipment includes a large-sized Mannesmann piercing mill, working with comparatively large-diameter ingots, as contrasted with the prerolled rounds used in the Diescher mill. Conjoined therewith is also a pair of pilger mills, which embody the second notable Mannesmann principle of tube making. Further details of the Mannesmann contribution to the art are unnecessary for purposes of this paper since their significance is principally historical.

The photograph from which Fig. 2 is reproduced was taken from a position near the racks where the elongated product is accumulated for crane transportation to locations in the plant where final processings take place as, for instance, cutting to length, sinking, upsetting, cold-drawing, inspecting, and testing. In the upper left-hand part of the illustration appear the heating furnace, the piercing mill, and the transfer equipment from the piercer outlet to the elongator inlet. Of the pair of large motors, appearing nearer the foreground and toward the left-hand region of the picture, the one farther removed from the foreground drives the piercer cross-rolls and the other drives the elongator cross-rolls.

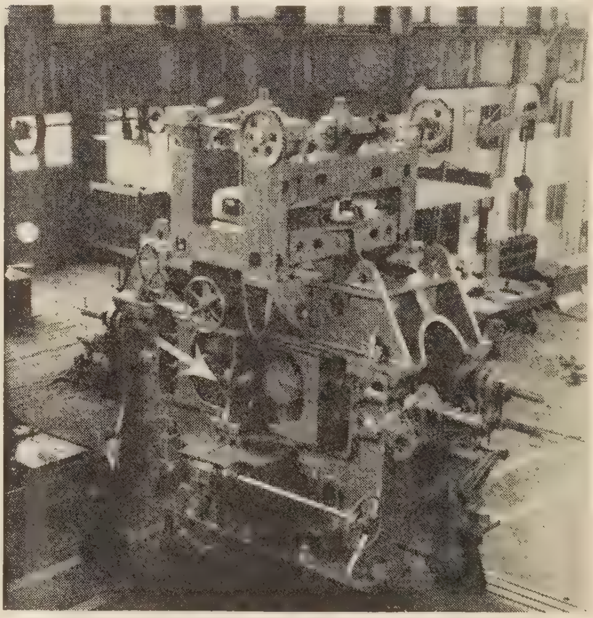

Fig. 3 Assembly of Elongator Stand

Fig. 3 is a view of the elongator stand while undergoing assembly in the machine shop prior to being set up as part of the elongator unit shown in Fig. 2. In this illustration, the main stand is shown in the reverse position to that which it occupies in Fig. 2, i.e., its entry side is seen in Fig. 3, whereas, its exit side is in view in Fig. 2.

In detail, Fig. 3 shows the main bearings at the nondriven ends of the cross-rolls; the arrangement of main screws, which for the sake of preventing any rocking of the main bearings are applied in pairs; and the screw rig for vertical adjustment of the upper guide disk. Because of the height of this rig above floor level, it is operated by hand chains and so arranged as to encircle a well located at the middle of the housing cap through which crane slings can be lowered for removing the guide disks from their shaft mountings when their renewal is required. Between the

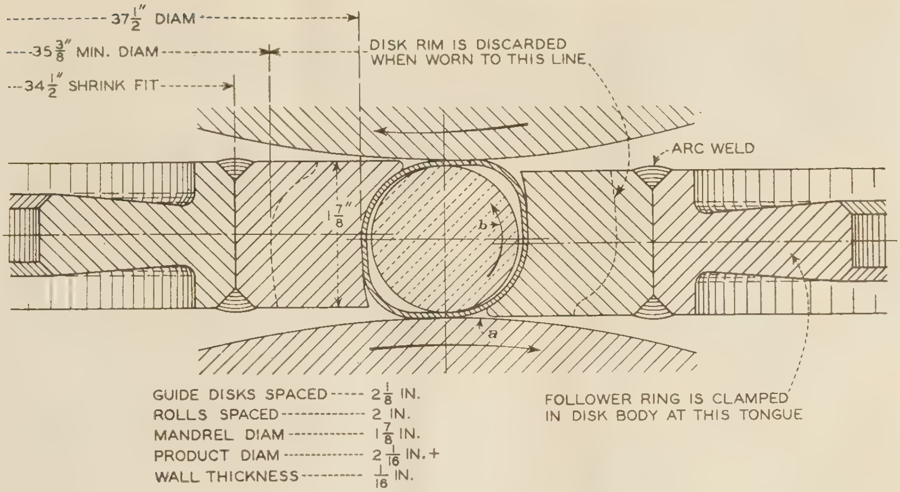

Fig. 4 Action of Rolls on Thin-Walled Tube

two pairs of main screws, Fig. 3 also shows a window giving access to the guide disks on the side in view, and entry for the guide-disk-drive spindles on the side opposite thereto and not in view.

The hand wheel, appearing centrally below the upper guide-disk-screw rig, is for clamping the upper guide-disk carrier rigidly in position, following its adjustment by the screw rig mentioned. The hand wheel appearing at the left of the one just mentioned is for adjusting the guide disk with its bearings sidewise of the vertical plane of the pass axis, the purpose of which adjustment will be explained later.

Immediately below the hand-wheel adjustment last mentioned and near the bottom of the stand is a companion adjustment which serves in the same manner as the upper one; but in this case in connection with the lower guide disk.

The hand wheel arranged at the right-hand side of the main stand and connected by a horizontal reach shaft to the worm drive, set centrally of the bottom of the housing, is for clamping the lower guide-disk carrier in a manner similar to that previously described as pertaining to the upper guide disk.

Finally, the hand wheel, on the vertical shaft centrally located in relation to the window opening at the right-hand side of the stand, is provided for the vertical adjustment of the lower guide disk. All these adjustments are duplicated on the side of the stand which is hidden from view.

The guide opening, indicated by an arrow, provides entry for the workpiece into the pass located centrally of the stand and formed by the opposed faces of the cross-rolls and of the vertically opposed faces of the guide-disk rims; on the opposite side of the stand there is another such guide which provides for the exit of the workpiece.

Comparing the size of this main stand with the size of the man standing alongside it gives an idea of its massiveness. The stand must be massive to prevent harmful give to the working elements, especially in the working of a thin-walled product accurately to size.

Referring again to the general view Fig. 2, it will be observed from the location of the crossover skids leading from the piercer outlet to the elongator entry, that the workpiece, after leaving the furnace, must travel alongside and beyond the piercer before entering it. After leaving the piercer, then traversing the crossover skids, and entering into and finally passing out of the elongator stand, it will also be noted that, in its elongating course, the workpiece travels in the opposite direction to that in which it proceeded through the piercer. This procedure was adopted because with thin-walled piercing, the shell tends to become somewhat smaller in diameter at its exit end than throughout its general course, which created a condition inviting chilling under the closer approach of the walls at this narrowed end to the elongator-mandrel surface. By causing that end to enter the guide-disk pass first, there is not the amount of time for chilling to take place that would occur if the direction of workpiece travel for both piercing and elongating was the same.

Obviously, before the pierced shell may be entered into the elongator, a mandrel bar must be strung through it. In the right-hand background of Fig. 2 may be seen a number of these bars resting on the mandrel-cooling or storage bed. As soon as a mandrel has been advanced from the bed to its position within the pierced shell, both shell and mandrel enter the elongator pass and are fed through it, traveling along between the elongator-drive spindles, thence through a passageway provided in the drive-gear housing, which can be seen in the center of the illustration, and finally forward along the runout table the drive of which appears in the left-hand foreground. By flag-switch control, the three-arm star-throwout rig, appearing immediately next this runout table, directs the finished tube with the mandrel still inside it across skids to a chain transfer and from that mechanism to another table which is parallel with the runout table but operates in the reverse direction. This latter table passes the tube and mandrel toward the rear to a pinch-roll stand which can be seen at the end of the table. At this point, the mandrel is extracted from the tube and proceeds along its course and is finally, by means of a star throwout, transferred to the mandrel storage and cooling rack.

By means of another star throwout, the tube from which the mandrel was extracted is moved sidewise into the finished-product storage rack, shown at the right, from which the product is removed in crane-load batches for further processing, as previously described.

Beyond this tubular product storage rack and midway between it and the background is shown the elongator guide-disk-drive motor.

Theory of Cross-Rolling Seamless Tubes

In a vertical plane, the cross-rolls are set at an angle of 6 deg with the direction of travel of the tube. Ordinarily and with no allowance for slip between cross-rolls and tube, the latter would travel (sin 6 deg) times the circumferential roll speed. A typical roll speed is 800 fpm, which makes the tube-delivery speed equal

to 0.105 × 800 = 84 fpm. Actually, the tube travels at a higher speed, i.e., approximately 100 fpm. Thus, the effect of the forward drag by the elongator disks becomes apparent.

The speed at which these disks operate gives rise to interesting speculation. The originator of the mill reasoned as follows: Thin-walled tubes tend to climb into the gap a, Fig. 4, and thereby

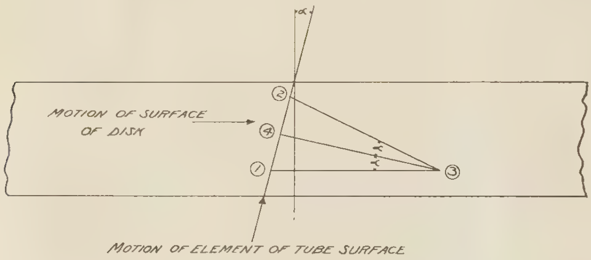

Fig. 5 Diagram of Tube Motion

become scrap, unless frictional resistance to revolving motion (arrow b) is minimized. The resistance is wholly absent if, in the diagram of motion Fig. 5, origin 1 and terminal 2 of the travel of a point on the tube surface are equally distant from point 3, which is the location of point 1 of the disk when the corresponding contact point of the pipe blank has arrived at 2. Geometrically (1) (3) is (1) (4) divided by sin α. Since equal intervals of time enter into the movements (1) (2) and (1) (3), velocities may be substituted. If, for instance, the peripheral speed is 800 fpm, the disk speed, according to this rule, should be 400/(sin 6 deg) = 400/0.105 = 3810 fpm. Actually, a lower speed, about 2300 fpm is used for the disk. At that speed, push by the cross-rolls is small when compared to the pull by the elongator disks.

To clarify this theory yet further, the following reasoning will be helpful: Referring again to Fig. 5, it will be seen that the cross-rolls move a point on the tube surface across the disk along the path (1) (4) (2). The problem is: How fast must the disk move so that the tube blank will be dragged along the path (1) (4) (2) by the disk, instead of being pushed across by the cross-rolls? The answer is the same as before.

This derivation is based upon the elementary laws of solid friction, which teach that the friction coefficient is independent of the speed. However, phenomena occur in the elongator mill which, although not yet investigated quantitatively, should at least be mentioned. Contact between hot steel, such as the tube blank, and the much cooler rolls, as well as the disks, lowers the surface temperature of the blank. Compression and deformation raise the temperature of the blank. Local friction between rolls or disks and blank also raises the temperature of the surface of the blank. This would be unimportant, but for the well-known fact that the friction coefficient between a roll and a hot-steel blank drops rapidly as the temperature of the surface of the blank rises, and vice versa.

The problem of ascertaining the surface temperature of the blank at the instant (and it is only an instant) of contact, is so filled with practical difficulties that no accurate measurements are available at the present time. One thing is certain, that control of the frictional phenomena has played an important part in the development of the elongator mill. While the instantaneous surface temperatures at the contact are not known, the more easily observed temperatures of the entering blank and tube at exit are known in a general way. For instance, it is known that the tube which emerges from the Diescher mill has a higher temperature than the blank which entered, whenever the elongation ratio substantially exceeds the value of 2 with present speeds.

Position of Disks

The speed of the disks having been disposed of, we shall now consider their position. At first, they were set with the same clearance as the cross-rolls, i.e., allowing only for the two wall thicknesses of the finished tube. Experience quickly taught that they must be set with greater clearance. It is undesirable to have the tube rolled tightly on the mandrel, for several reasons. With a tightly fitting tube, the mandrel becomes hot, the resistance to forward sliding of the tube on the mandrel during the rolling process is increased, and there is difficulty in removing the finished tube from the mandrel. In addition, the metal of thin-walled tubes tends to crowd into the gap or cleft between the cross-roll and the elongator disk, because the friction between the mandrel and a tightly fitting tube hinders elongation. Briefly expressed, the setting of the cross-rolls determines the wall thickness, while the setting of the elongator disks determines the diameter of the tube. An economic corollary is that the diameter of the mandrel may vary within wide tolerances, with the proviso that the mandrels used for one given roll setting must all be alike in diameter. Since the mandrels stay comparatively cool, they are not substantially deformed by the rolling process, but there is some wear. In consequence, no mandrel conditioning is required beyond that attainable by centerless grinding.

The problem of disk setting can be further elucidated by giving a few settings from actual practice. To roll a tube $2^1/_{16}$ in. diam and $^1/_{16}$ in. wall thickness on a $1^7/_8$-in-diam mandrel, the cross-rolls may be set apart 2 in. at the throat and the guide disks about $2^1/_8$ in. apart at their closest approach. The resulting tube would be $2^1/_{16}$ in. diam, with $^1/_{16}$ in. wall thickness, and would provide $^1/_{16}$ in. of the diam to be removed in sizing to a 2-in-diam product.

To roll a tube $2^7/_8$ in. diam × $^3/_{32}$ in. wall thickness on a $2^5/_8$-in. mandrel, the cross-rolls would be set apart $2^{13}/_{16}$ in., and the disks about $2^{15}/_{16}$ in.; with a $2^9/_{16}$-in. mandrel, the cross-rolls could be set apart $2^3/_4$ in. and the disks 3 in., although this is not the best practice.

Shape and Lateral Position of Elongator Disk

A few words should be said about shape and lateral position of the elongator disk. Since the tendency of the tube wall (particularly a thin tube wall) is to work into the gap between the cross-roll and elongator disk, it is highly desirable to reduce the width of that gap to a minimum. This feat is accomplished by making the disk one-sided, e.g., with one edge longer than the other, as shown in Fig. 4. In addition, the disk is adjustable sidewise for the purpose of obtaining the best location for each size of tube.

Several years ago, when the author first studied the Diescher elongator mill, he felt that the great relative rubbing speed between elongator disk and tube might have two undesirable effects, i.e., excessive wear of disk and excessive power consumption. In the beginning, the design and material of the disks offered a problem which, however, was soon solved by welding a rim of austenitic steel to a soft-steel center. The rim contains 25 per cent chromium and 12 per cent nickel. One set of disks has rolled as many as 20,000 tubes before having to be reconditioned. Reconditioning is required, because the disks ultimately become deformed in profile and thus become rough, whereby free circumferential flow of the tube material along the disk profile is obstructed. The tube wall then tends to work into the gap or cleft, and so-called "disk wipes" occur in the finished tube. The contact time between disk and tube is about $^1/_{150}$ sec. This time does not afford opportunity for heat penetration into the disk. Several methods are in use for fastening the disks to their shafts. One method is illustrated in Fig. 4.

Power Required for Rolling Operations

With regard to power consumption, the following data are available. The rolling of thin-walled tubes requires more power per unit of volume times elongation in unit time than the rolling of the thick-walled product.

The power required per net ton of product also varies with the size of product being made. For example, in rolling common-steel pierced blanks of 3 in. diam and 0.266 in. wall to a product of $2^5/_8$ in. diam and 0.125 in. wall at the rate of about 1.7 fps out of the elongator pass and produced at the rate of 240 pieces 20 ft long per hr, there would be required about 800 kw of power for all drives, including table drives of the elongator.

Among data regarding the consumption of power in elongating per se, and therefore dealing solely with that required for driving the cross-rolls and guide disks, the following figures should be of value: In elongating to 0.09 in. wall thickness, about 95 hp are required per cu in. of metal displaced per sec; whereas, in elongating to double that thickness about 70 hp are required per cu in. displaced per sec, the guide disks consuming about 25 per cent of the power consumed by the cross-rolls.

Closely allied with disk wear and power consumption is the cost of tooling. The smaller the diameter and the wall thickness of the product, the higher is the cost of tooling per ton of product. At present, the cost of all tooling in rolling the $2^5/_8 \times {}^1/_8$-in. wall tubing, mentioned under power consumption, is about $1.10 per net ton of product. This includes all items such as rolls, disks, mandrels, etc., and is expected ultimately to be substantially reduced.

For the sake of completeness, additional questions must be

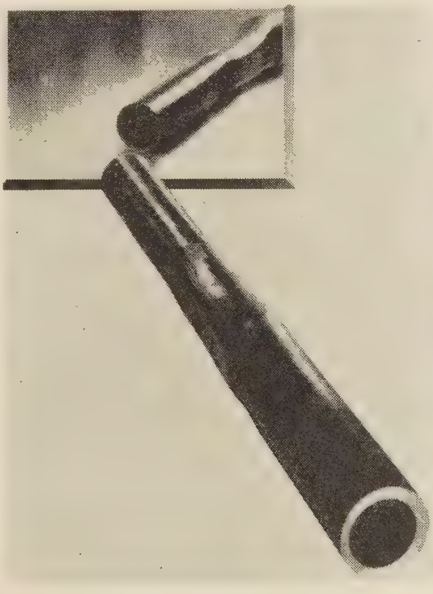

Fig. 6 Tube Partially Rolled From Eccentric Blank

discussed briefly; one is the diameter of the cross-rolls. They must be rigid enough to roll the thinnest practical tube without noticeable deflection, even if the temperatures among the blanks vary. Designers prefer to be on the safe side and make them somewhat larger than the needed neck diameter implied, rather than too small. The next question is that of greatest possible elongation. The Diescher elongator has rolled with 5 elongations, which means that the delivered tube is 5 times as long as the entering hollow blank. In practice, such extreme elongations

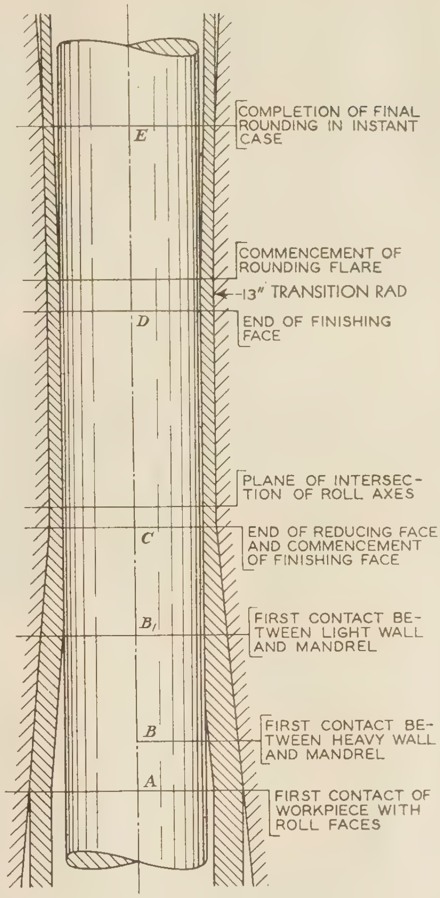

Fig. 7 Diagram of Stages in Rolling Operation

are not used. The average elongation is about 2, which means that the tube has twice the length of the hollow blank. In this manner, piercer output and elongator output are about alike, and the highest rate of combined production results.

Concentricity of Diescher Mill Tubes

The elongator mill has gained a reputation for producing concentric tubes. This statement may be discussed from two angles, viz., the importance of concentricity, and the reasons why the Diescher mill attains it. The importance of concentricity is realized by all consumers of tubing and therefore need not be stressed here. The next question is: Does the elongator produce concentric tubes and, if so, why? The fact that it produces concentric tubes may be judged from Fig. 6, which shows a tube that had been rolled part way from an eccentric blank. The latter had been bored out of center. The reason for rolling a bored blank instead of a blank produced by cross-rolling is that a uniformly eccentric blank cannot be obtained from a piercing mill because, in the latter, the piercing plug meanders, following fortuitous soft spots caused by nonuniform heating or segregation.

In turning now to reasons for the concentricity of tubes rolled on the elongator mill, it is conceivable that, when the tube blank first enters, an eccentric blank would vibrate the mandrel with high frequency, whereby resisting forces are produced which, in conjunction with the well-known fact that thick walls are more easily deformed than thin walls, tend to reduce the eccentricity of the workpiece. Furthermore, a thick wall portion introduced

into the cleft between the rolls and the mandrel emanates therefrom more pronouncedly than its companion thinner wall portion, rubs harder against the elongator disks, becomes hotter, and is more easily turned into length. By the combined effect of these actions, the mandrel becomes centered by the time the tube is traversing the long finishing pass, which in Fig. 7 reaches from C to D. From then on, the remainder of the tube (which means almost the entire length) is forced through the spaces between the rolls and a concentrically held mandrel, whereby concentricity is obtained. It goes without saying that concentricity as used in practice is a relative term, and that maximum approach to perfect concentricity requires maximum rigidity of the mill.

Conclusion

In conclusion, we may consider the economic side of the matter: Under what conditions should existing equipment be replaced by an elongator mill? Up to the present time, this question has not arisen in practice, because the four mills which are now in operation were installed as additional equipment for the purpose of making a product which heretofore could not be made, i.e., a hot-rolled, seamless, concentric tube with very smooth surfaces. Upon inquiry the author discovered that the patent owner, the Diescher Tube Mills, Inc., has a very commendable policy; new licenses seem unlikely to be solicited unless the existing mills cannot cope with the demand.

Discussion

A. B. Cox.[2] In the Diescher tube-rolling machine, as in steel-rolling processes in general, successful operation is dependent upon relative values of external and internal friction. The great importance of this principle in the field of rolling materials may therefore justify some comment on this phase of the paper.

At one point the author states that the theroetical derivation of relative speeds of cross rolls and longitudinal rolls is based on the assumption that the coefficient of external friction is independent of the speed, i.e., relative motion or slip, which previously had been taken as zero. The writer knows of no experimental data which would justify this assumption of a constant coefficient of friction for all speeds. Regardless of whether the friction is dry or lubricated, experimental data show that the coefficient of friction varies very widely with speed.[3] If the coefficient of friction between hot and cold steel is constant over any considerable range of speed, a citation to the data which show this should be of interest.

The author also refers to the laws of plastic flow, which brings up the subject of internal friction. It is well known that, in general, the resistance of a solid to deformation varies with the rate of deformation. The ease of flow, or "flowability" of the material, increases with increase in the rate of deformation; at least for low and intermediate rates of deformation. Conversely, the coefficient of viscosity (or coefficient of internal friction) decreases.[4] This is true of all materials for which data are available. The coefficient of viscosity for all materials, solid, liquid, or gaseous, follows the same law. It is only when data are taken over a relatively limited range of speed that the coefficient appears to be approximately constant in some cases. This is true even for water and for air. Hence, it is extremely probable that the coefficient of internal friction of steel, no matter how cold or how hot, varies greatly with rate of deformation.

Given experimental data for the curve of variation of speed versus coefficient of internal friction of steel at the temperature at which the metal is to be worked, and the curve of variation of the coefficient of external friction versus speed for hot or cold steel, would it not be possible to predetermine the relative speeds of the cross rolls and the elongator rolls of the Diescher machine for optimum performance with engineering certainty? It is the common impression among steel men that all mills of this general type are hard to get started in successful production on any but a product which has already been fully standardized in production. Apparently this is due to the difficulty of obtaining just the right relative tangential and longitudinal speeds. If the author can supply information which will refute this general opinion, it would be of value both to the tube manufacturer and to the manufacturer of the tube-rolling machine.

C. W. Littler.[5] From the writer's investigations in the matter, it is evident that the Diescher elongator has contributed greatly to the advancement in the art of seamless-tube manufacture. This is particularly true with reference to the ability of such a mill to maintain uniform wall thickness within very close tolerances. It is understood that elongations of 4 to 1 have been accomplished. No doubt the rotary disks add greatly to this possibility. The uniformity achieved on such a tube coming from the elongator has a beneficial effect on the reducing mill.

G. A. Pugh.[6] The Diescher elongator finishes pierced blanks into seamless tubes, eliminating intermediate operations such as pilgering, plug rolling, and reeling. The plug-rolling and reeling intermediate operations are employed by most of the tube manufacturers in this country.

The urge to eliminate the plug-rolling mill, as a means of elongating pierced hollows, is a natural one, since the plug-rolling operation is irrational and contributes most of the difficulties usually considered inherent with the manufacture of seamless tubes. For the sake of clarity, it should be understood that the Diescher mill has been used for tube sizes having a maximum diameter of about 4 in. The hot finishing of tubes of light wall, smaller than 4 in. diam, has been a difficult problem and the Diescher development along this line is noteworthy. Qualities of the product, such as accuracy of wall thickness and reduction in the percentage of eccentricity, have made the unit a commercial success.

On the other hand, large-diameter hot-finished tubes have been successfully finished by cross rolling, as for example, by the rotary rolling method employed by the National Tube Company. As one of the developments of R. C. Stiefel, mention was made of it in a paper[7] presented in 1928. The use of a second piercer, as a means of elongating pierced hollows, also has been widely employed by all of the makers of large sizes of seamless tubes.

It will be interesting to note the development in the Diescher mill as time goes on and to what degree it may be adapted for the production of large sizes of tubes. Certainly, there must come a day when the economic demand and the necessity for closer wall tolerances will force engineers to adopt methods which make such attainments possible.

C. R. Sadler.[8] This paper has been of particular interest to the writer who was closely concerned in the installation of the

[2] South Hills, Pittsburgh, Pa.

[3] "Journals and Bearings" section, L. S. Marks, Mechanical Engineers' Handbook, third edition, 1930. Stribeck data for a lubricated journal, pp. 243, 244.

[4] Refer to Bibliography on subject of "Friction," by Committee E-1, American Society for Testing Materials, 260 S. Broad Street, Philadelphia, Pa

[5] Chief Engineer, Jones & Laughlin Steel Corporation, Pittsburgh, Pa. Mem. A.S.M.E.

[6] Youngstown, Ohio. Mem. A.S.M.E.

[7] "The Manufacture of Seamless Tubes," by R. C. Stiefel and G. A. Pugh, Trans. A.S.M.E., vol. 49-50, 1927–1928, paper IS-50-7, pp. 17–22.

[8] American Munitions Division of American Type Founders, Inc., Elizabeth, N. J.

first elongator, and for several years was personally familiar with its operation. As the author states, this is the first time the theory underlying the "elongator" method of producing tubes has been published. The statements in his paper are borne out by the writer's experience.

While this paper makes mention of the concentricity of the tubes produced by this process and refers to tests made with an eccentrically drilled blank, the actual wall measurements of the blank in question were not stated. It may be of interest to know that these wall dimensions were as follows:

Before processing....................	0.120 in. minimum 0.358 in. maximum
After processing.....................	0.095 in. minimum 0.104 in. maximum

Another advantageous feature of the elongator method is the smooth interior surface of the product. Internal scratches have always been the bugbear of the seamless-tube industry, and no satisfactory method has been devised for their removal. For this reason, a method of hot rolling was sought in which scratches could not occur. Naturally, attention was turned to the idea of rolling tubes over a smooth bar rather than over a plug. Of the various methods of accomplishing this, the elongator method has proved to be the most satisfactory. The reason for this will be apparent from a study of Fig. 1 of the paper. It will be noted that there is but a slight sliding movement between the tube and the bar, and that the tube is in momentary rolling contact with the bar at only two points during the operation. Thus, there exists no tendency to produce scratches or scores in the tubes. Not only is this a very desirable feature in the production of hot-finished tubes, but where tubes are subsequently cold drawn it results in the production of superior tubes with fewer cold-draw passes.

Another feature is that, while the process is suitable for any size tube, smaller hot-finished tubes have been successfully produced by this method than were possible before its introduction. The advantage of this will be apparent to any tube manufacturer who has had experience with the difficulties inherent in hot sinking.

Author's Closure

The constructive character of the discussion which has been offered on the present paper is very much appreciated. Additional facts have been presented and they are helpful in the understanding of the process.

Mr. Cox recommends that additional research work be done on the friction between blank and rolls and also on the internal friction within the blank. Information on these problems has been accumulating for some time but as yet is not adequate for presentation.

The Flow of Saturated Water Through Throttling Orifices

By M. W. BENJAMIN[1] and J. G. MILLER,[2] DETROIT, MICH.

In this paper are presented the results of tests to determine the flow characteristics for saturated water and for various mixtures of saturated water and steam through sharp-edged thin-plate orifices. Tests show that the actual flow of saturated water through these orifices is considerably greater than would be expected from theoretical calculation based upon a change of state and that no critical back-pressure condition is evident over the range of initial pressures considered. Primarily, this investigation was intended to determine the feasibility of using throttling orifices alone or in combination with float-operated drainers for regulating the draining of condensate from feedwater heaters. The tests which form the basis of the paper have provided sufficient information to permit the derivation of practical design formulas, which have been used successfully in several instances by the authors' company. These test data apply only to throttling orifices and should not be used to design orifices for metering purposes. An Appendix to the paper shows the application of the formulas to the design of a single-stage orifice to drain the condensate from a feedwater heater, and to the design of an orifice to be used in series with a float-operated drainer.

EARLY in 1939, the authors became actively interested in the flow of boiling water through pipes in connection with the rapid erosion of elbows in heater drain piping on a 60,000-kw steam turbine. After some study, it was found that erosion was a function of the amount of flashing and the quantity of water flowing in the pipe, and interest in the subject was intensified by the indication that certain operating difficulties with float-operated condensate drainers also might be traced to the phenomena encountered with water flashing into steam in the drainer discharge pipe.

In a paper[3] by Bottomley, the suggestion was made that orifices could be used in place of float-operated traps for draining feedwater heaters. Such an application would eliminate the operating troubles with traps and, if the orifices were installed at the end of the drain line rather than at the beginning, would prevent erosion in the piping.

While the theoretical analysis of the flow of saturated water (water at saturation temperature and pressure) through orifices, which assumes a change of state in the orifice, seemed straightforward enough, the limited published test data showed the actual capacity of an orifice passing saturated water to be several times greater than its theoretical capacity. Since the available test data were not complete and because practical coefficients were needed for actual design purposes, it was decided to conduct an experimental investigation of the flow of saturated water through sharp-edged thin-plate orifices to determine whether it would be practicable to use such orifices in lieu of traps for draining feedwater heaters.

While the tests which form the basis of this paper do not cover all the variables which might have been investigated, they are sufficiently complete to provide a practical basis for design that has been applied successfully in several instances.

In this paper the term "saturated water" is used in preference to "hot" or "boiling" water as used by earlier writers, since it is considered to be more definite and possibly more accurate. It is used to denote water at saturation pressure and temperature, and refers to the condition of the water on the upstream side of the orifice.

Comparison of Theoretical and Actual Flow of Saturated Water Through Orifices

The theory of the flow of saturated water has been thoroughly covered in several published papers[4] and, therefore, will not be repeated here.

For purposes of comparison, Fig. 1 shows the theoretical and actual flow of saturated water through orifices for an initial pressure of 145 psi abs and back pressures ranging from 0 to 145 psi abs, while Fig. 2 shows the maximum theoretical and actual flow through orifices for initial pressures ranging from 14.7 to 300 psi abs and a constant back pressure of 14.7 psi abs.

Experimental Investigation of Flow of Saturated Water Through Orifices

Fulfillment of the purpose of the test required that several points be kept in mind concerning the design of the orifices and test equipment, as follows:

1 That the draining of condensate from a higher-pressure to a lower-pressure feedwater heater is primarily a throttling process.

2 That the controlling factors to be considered in designing an orifice for draining condensate from one heater to another for maximum load on a turbine are (*a*) the pressure differential between the heaters; (*b*) the initial temperature and pressure of the condensate; and (*c*) the quantity of condensate to be drained from the higher-pressure heater. The initial temperature of the condensate will be the saturation temperature corresponding to the initial pressure in all cases except those in which there is undercooling. The effect of undercooling on the flow of the condensate through an orifice is in general the same as the effect of a static head on the upstream side.

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[3] "Flow of Boiling Water Through Orifices and Pipes," by W. T. Bottomley, Trans. North-East Coast Institution of Engineers and Shipbuilders (England), vol. 53, 1936-1937. pp. 65–100.

Contributed by the Power Division, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[4] "Experimental Researches on the Flow of Steam Through Nozzles and Orifices," by A. Rateau, D. Van Nostrand & Co., New York, N. Y., 1905, supplementary chapter at end of book, pp. 62–74.

"Discharge Capacity of Traps," by A. E. Kittredge and E. S. Dougherty, *Combustion*, vol. 6, September, 1934, pp. 14–19.

"Fluid Flow Through Two Orifices in Series," by Milton C. Stuart and D. Robert Yarnall, *Mechanical Engineering*, vol. 58, 1936, pp. 479–484.

"Flow of Boiling Water Through Orifices and Pipes," by W. T. Bottomley, Trans. North-East Coast Institution of Engineers and Shipbuilders (England), vol. 53, 1936-1937, pp. 65–100.

"The Flow of Hot Water Through a Nozzle," by B. Hodkinson, *Engineering* (London), vol. 143, 1937, pp. 629–630.

3 That all of the factors given under (2) decrease with a drop in load on the turbine; and it is possible that an orifice designed for full load will be larger than needed to pass the condensate at reduced load even though the pressure differential across the orifice also is reduced. In such a case, some steam will be cascaded to the lower-pressure heater along with the condensate. This means that more steam will be bled from the higher-pressure turbine stage than is necessary for its own stage of feedwater heating, which in turn may reduce the turbine cycle efficiency somewhat, especially if the amount of steam cascaded is excessive.

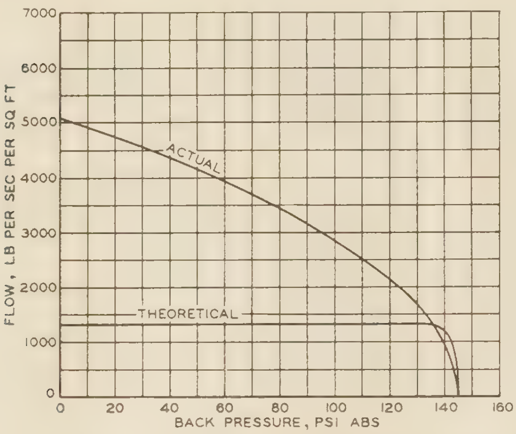

FIG. 1 THEORETICAL AND ACTUAL CHARACTERISTICS FOR FLOW OF SATURATED WATER THROUGH ORIFICES
(Initial pressure, 145 psi abs.)

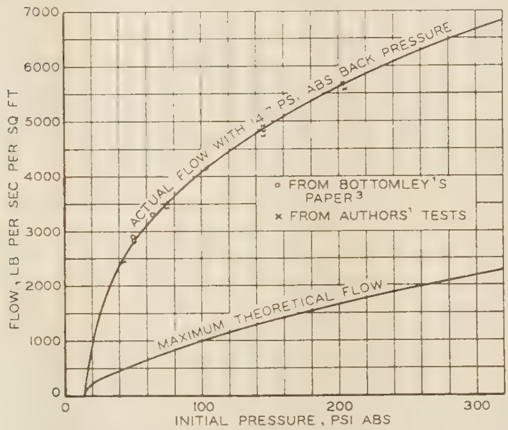

FIG. 2 ACTUAL FLOW AND MAXIMUM THEORETICAL FLOW OF SATURATED WATER THROUGH SHARP-EDGED ORIFICES
(Initial pressures ranging from 14.7 to 300 psi abs and a constant back pressure of 14.7 psi abs.)

Therefore, in order to provide the necessary test data for designing orifices to drain the condensate from feedwater heaters, the test equipment was built to permit the determination of (*a*) the flow of saturated water through orifices for various initial and back pressures; (*b*) the effect on the flow of saturated water through orifices produced by a static head above the orifice (undercooling); and (*c*) the effect on the flow of passing steam through the orifice along with the water.

Design of Orifices. While it is not an established fact, it is expected that long-continued throttling may produce wear or erosion of the orifice with consequent passage of steam at all turbine loads. To prevent loss of cycle efficiency it might be necessary to make periodic replacements and, for this reason, any heater drain-line orifice should be simple to make and easy to replace in service. If the orifice plate is made of corrosion-resisting steel, experience shows that, where no change of state occurs, there will be little if any wear of the sharp edge of the orifice; however, with orifices installed in hot-drip systems, considerable erosion due to flashing or cavitation has been noted on the downstream face of the orifice plate. It should be noted here that drain-line orifices are not intended for metering purposes.

Fig. 3 shows the design and Table 1 gives the diameters of the orifices used in this investigation. These orifices were installed in a horizontal 6-in. pipe.

TABLE 1 DIAMETERS OF ORIFICES USED IN INVESTIGATION

Orifice number	Orifice diameter, in.
1	0.247
2	0.369
3	0.364
4	0.503
5	0.614
6	0.707
7	0.879

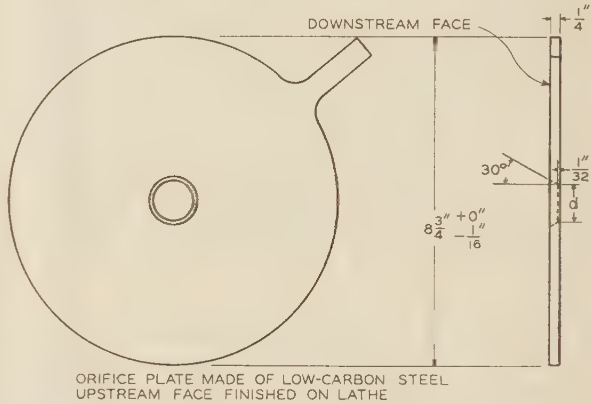

FIG. 3 DESIGN OF ORIFICES USED IN TESTS OF FLOW OF SATURATED WATER

Description of Test Equipment. Equipment used in the orifice tests consisted of the following:

1 A flash tank for regulating the pressure and supply of saturated water.

2 A set of orifices and a flanged holder for them.

3 A heat exchanger.

4 Two weigh tanks—one for the condensate and another for the heat-exchanger cooling water.

5 Temperature-measuring apparatus.

6 Pressure gages.

The schematic arrangement of the equipment and the relative location of thermocouples and of pressure gages is illustrated in Fig. 4.

Fig. 5 shows an orifice clamped in the flanged holder. The short glass filler immediately downstream from the orifice was used to permit visual observation and photography on some of the runs. During most of the tests, however, a steel filler was used.

All temperatures were measured with iron-constantan thermocouples, calibrated for 8 in. immersion and a reference junction temperature of 32 F. Calibration of the couples is believed to be accurate within ±0.5 F and, because of the depth of immersion and the precautions taken to prevent air circulation around the couples in their wells, it is believed that the measured temperatures are in error by no more than ±0.5 F. The design of the thermocouple well is shown in Fig. 6.

Operation of Equipment. The controlling conditions in operat-

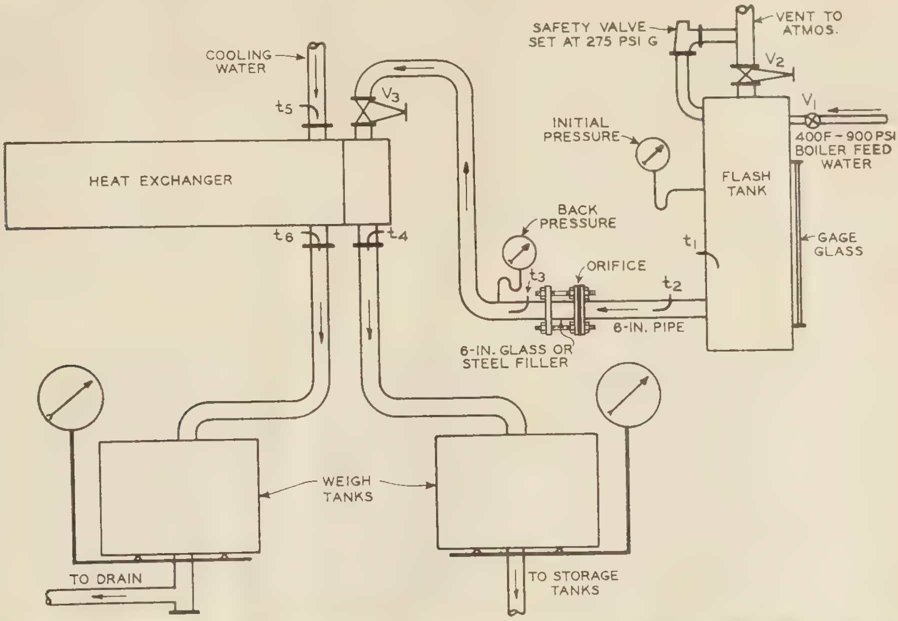

Fig. 4 Schematic Layout of Equipment for Testing the Flow of Saturated Water Through Sharp-Edged Orifices

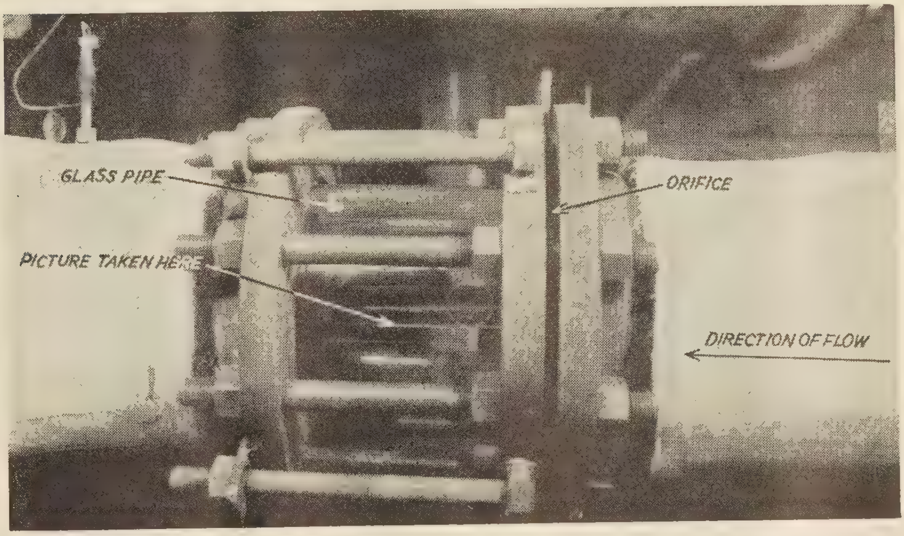

Fig. 5 Orifice and Glass-Filler Assembly

ing the test equipment were the pressure and water level in the flash tank and the back pressure on the orifice. For any given test, the required combination of these conditions was obtained by balancing the flow of 400-F water into the flash tank through valve V_1 against the flow of flashed steam through vent valve V_2 and of water or a mixture of water and steam through the orifice and back-pressure control valve V_3. Usually about 15 min were required to obtain the desired combination of pressures and water level, but once this condition was established slight adjustment of the valves was sufficient to maintain it because of the relatively constant pressure and temperature of the water supply. For the tests in which the orifice was passing both water and steam, the water level remained constant at about the center line of the horizontal 6-in. pipe and was of secondary importance. However, for the tests in which only saturated water passed through the orifice, the water level was established at from 3 to 8 in. above the center line of the orifice and was maintained at the established level within $\pm 1/2$ in. In the tests to determine the flow with a

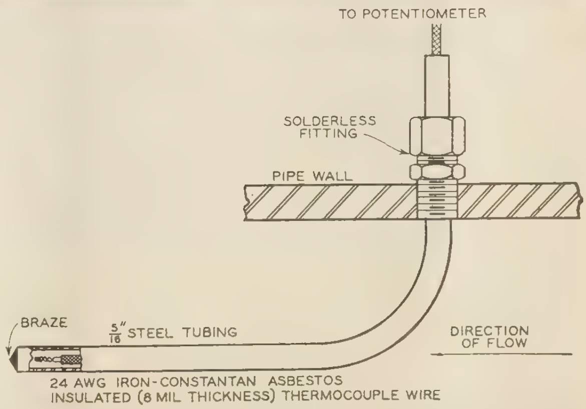

Fig. 6 Design of Thermocouple Well Used in Tests to Determine Flow of Saturated Water Through Sharp-Edged Orifices

static head above the orifice, the flash tank was filled with water under a pressure greater than the saturation pressure corresponding to the water temperature. However, in all cases, the inlet-water temperature was above the saturation temperature corresponding to the back pressure on the orifice.

Test Results

The test data give the actual flow of saturated water through sharp-edged thin-plate orifices for various initial pressures and show the effect on this flow of (*a*) varying the back pressure on the

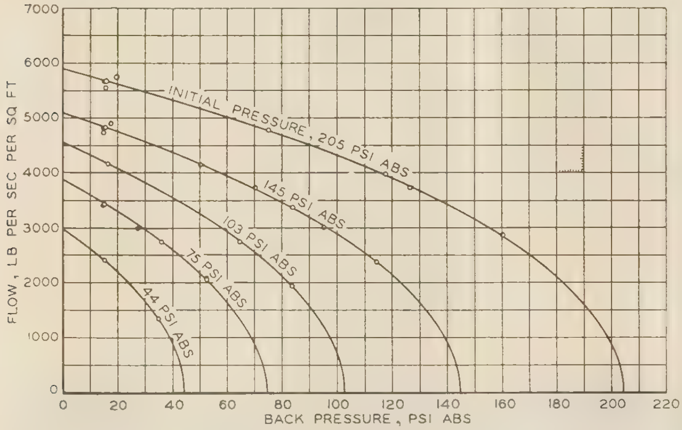

Fig. 7 Actual Flow of Saturated Water Through Sharp-Edged Orifices

orifice, (*b*) passing steam along with the water, and (*c*) a static head above or undercooling before the orifice.

The actual flow of saturated water through orifices, on the basis of pounds per second per square foot of orifice area as found on test for five initial pressures and various back pressures, is presented in Fig. 7. The results given were obtained from tests of

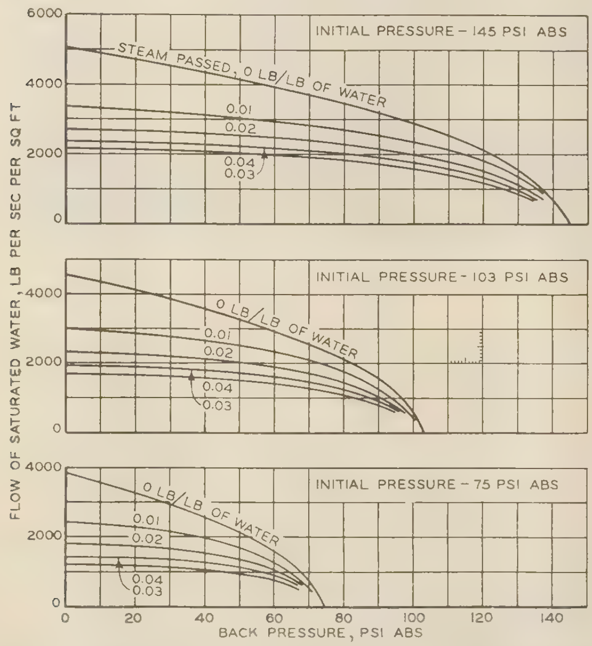

Fig. 9 Actual Flow of Saturated Water Through Sharp-Edged Orifices When Passing Various Mixtures of Steam and Water

all the various sizes of orifices listed under "Design of Orifices."

Fig. 8 shows the flow of saturated water when a mixture of steam and water is passed through an orifice. The curves are arranged to show how the initial pressure, back pressure, and relative amounts of steam, included in the mixture, affect the

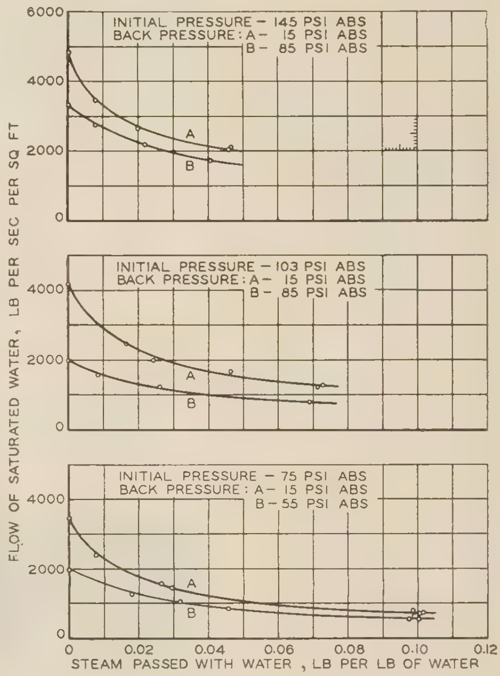

Fig. 8 Actual Flow of Saturated Water Through Sharp-Edged Orifices When Passing Mixture of Steam and Water

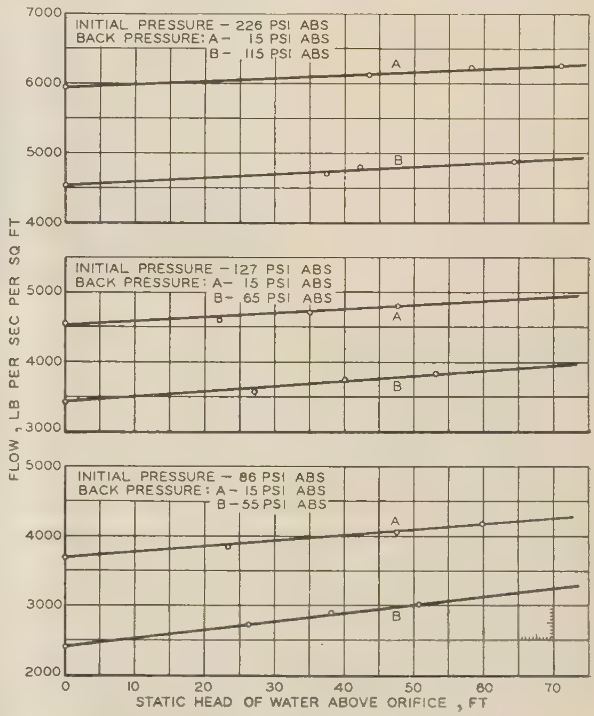

Fig. 10 Effect of Static Head on Flow of Saturated Water Through Sharp-Edged Orifices

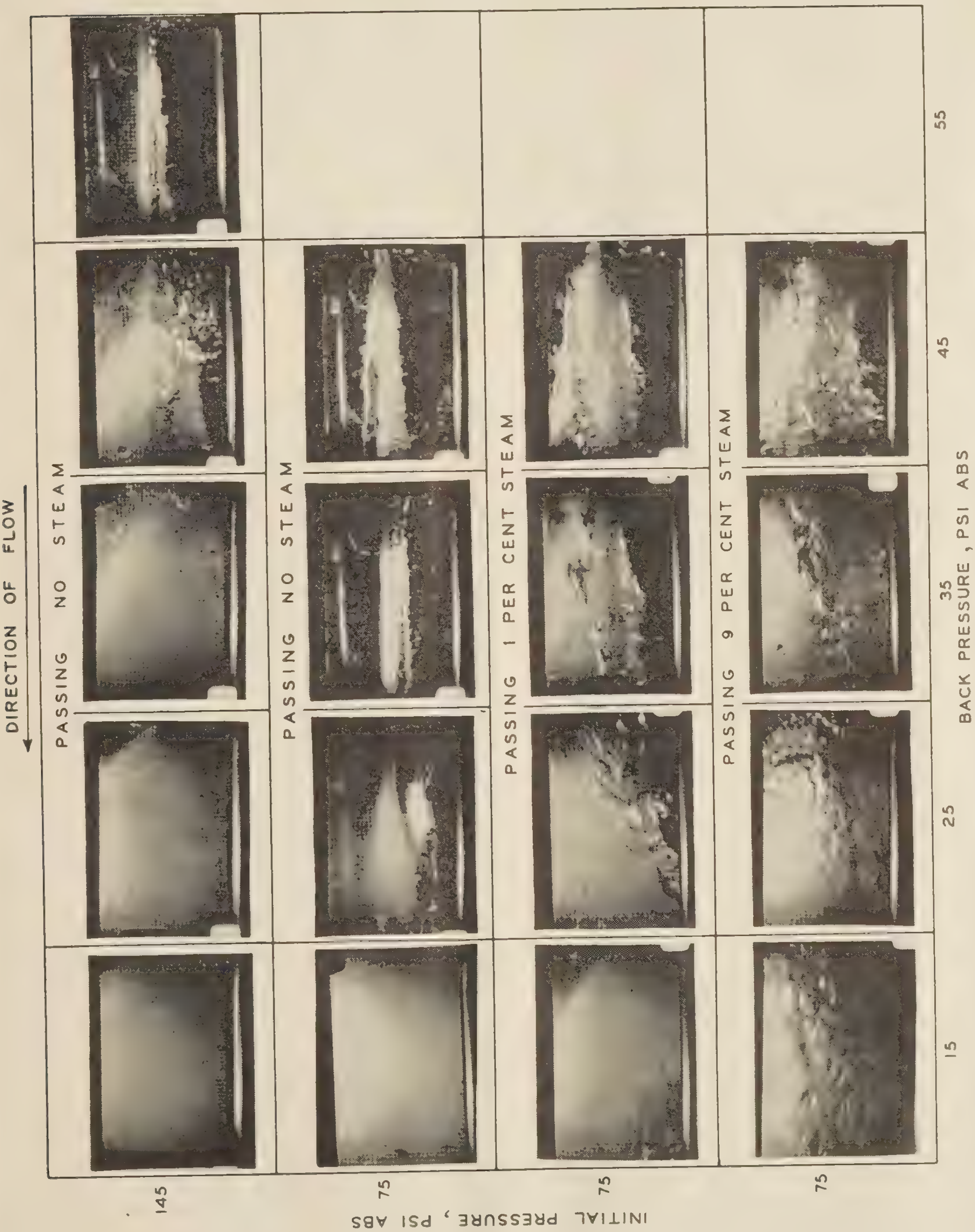

Fig. 11 Mixture of Steam and Water Leaving Downstream Face of Orifice

(Each picture represents one test, the initial pressure for which is given in the ordinate and the back pressure in the abscissa. The direction of the flow is from right to left. The screen shown in the pictures is a guard around the glass filler.)

flow. It is important to note that the amount of steam passed is given as a fraction of 1 lb per lb of water and that the total weight actually passed by the orifice is the sum of the weights of water and steam. Fig. 9, obtained by combining the data for initial pressures of 75, 103, and 145 psi abs from Fig. 7 with the data from Fig. 8, shows clearly and in convenient form the effect of passing steam with the water.

Should the rate of supply exceed the capacity of an orifice, a static head of water will build up above the orifice. Therefore, the water at the orifice center line will be under a pressure in excess of the saturation pressure corresponding to the temperature by the amount of the static head. Under this condition the fraction of water flashed into steam in passing to a region of lower pressure will be exactly the same as though the orifice were merely submerged, but the capacity of the orifice is increased somewhat as a result of the increase in pressure drop across it represented by the static head. An equivalent case is one in which the orifice is passing hot water at a normal level but undercooled to some extent below the saturated temperature. For instance, cooling the water from a saturation temperature of 332.1 F down to 317.1 F, while maintaining the original saturation pressure of 106 psi abs, is the same condition as having an equivalent static head of 20 psi (50.7 ft) above saturation pressure with water at 317.1 F. Consequently, the test data for both conditions are presented on the basis of a static head above the orifice. Fig. 10 shows how static heads of from 0 to 70 ft affect the flow at three different initial saturation pressures.

The photographs, reproduced in Fig. 11, show the flow leaving the downstream face of the orifice for two initial pressures and various back pressures. The two upper sets of pictures are from tests in which no steam was passed with the water, while the two lower sets are from tests in which steam was passed through the orifice with the water as indicated. In the two upper groups of pictures, it is interesting to note that, for the higher back pressures, the flow leaves the orifice in the form of a jet with practically no flashing evident within the length of the glass filler. As the back pressure decreases, however, flashing occurs nearer the orifice as shown by the breaking up of the jet, and the lower the back pressure the nearer to the orifice the flashing occurs until, at 15 psi abs back pressure, the flashing begins at the orifice downstream face.

Analysis of Results

As far as is known, the results presented in this paper and those given by W. T. Bottomley,[3] are the only published test data which give the actual flow of saturated water through orifices. In his tests, Bottomley could not, because of limitations in his equipment, determine the effects on the flow caused by (*a*) varying the back pressure, (*b*) passing steam with the water, and (*c*) a static head above or undercooling before the orifice. The results of some of his tests, however, agree very closely with the results found in this investigation as shown in Fig. 2. In accordance with the theory, Bottomley assumed that, even though the actual flow was several times greater than the theoretical, there must be a critical pressure in the orifice; therefore, all of his tests were run with an atmospheric back pressure (14.7 psi abs), which was considered below the actual pressure in the orifice. The data obtained in the present investigation and presented in Fig. 7 do not show the presence of a critical pressure in the orifice, and the pictures in Fig. 11 seem to bear out the conclusion that, for the range of initial pressures included in this study, no critical pressure exists in the orifice. In other words, the change of state does not occur within the orifice.

It is a well-known fact that the weight of steam which will flow through a nozzle is a maximum when the throat pressure is approximately 58 per cent of the upstream pressure. A decrease of back pressure below 58 per cent of the initial will have no effect on the flow. This, however, is not true for steam flow through a sharp-edged orifice, as is shown by the typical curves[5] in Fig. 12. Fig. 7 shows a similar orifice characteristic for saturated water, which further substantiates the conclusion that no critical pressure will exist in a sharp-edged orifice passing saturated water, even though the pressure differential across the orifice is sufficient to cause flashing at the downstream face. Whether or not a critical pressure will exist in the throat of a nozzle passing saturated water cannot be determined from the results of this investigation.

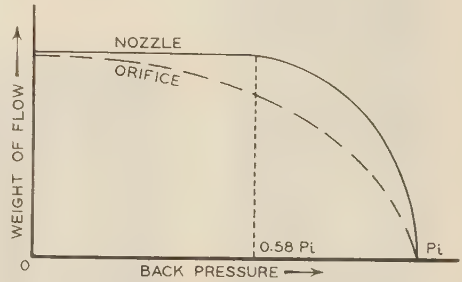

Fig. 12 Typical Curves Showing Relative Flow of Steam Through Orifices and Nozzles

An orifice, operating between fixed initial and back pressures, will pass a given amount of saturated water according to data in Fig. 7. If the amount of saturated water available is less than the orifice is capable of passing, a small amount of steam will flow through the orifice with the water. The amount of steam which will flow depends upon the initial pressure, the back pressure, and the quantity of water available. For instance, if the initial and back pressures are 145 and 40 psi abs, respectively, and the amount of saturated water to be passed is 2600 lb per sec per sq ft of orifice area, Fig. 9 shows that 0.02 lb of steam will pass through the orifice with each pound of water. From Fig. 8, it is seen that the water flow changes quite rapidly with an increase in steam flow from 0 to 0.04 lb per lb of water; however, for an increase in steam flow above 0.04 lb per lb, the change in water flow is much slower. This fact is also shown in Fig. 9 where the curves crowd together as the steam-flow fraction increases.

The curves shown in Fig. 7 are of the same general shape as the curve giving the flow of "cold" water (70 F) through orifices. which is represented by the equation

$$Q = CA\sqrt{2gh} \qquad [1]$$

where Q = the flow in cu ft per sec, A = area in sq ft, h = head in ft of flowing fluid, and C = orifice coefficient. By substituting wv for Q and $\frac{144}{\rho}(p_1 - p_2)$ for h, the equation takes the form

$$\frac{w}{A} = \rho C\sqrt{2g \times \frac{144}{\rho}(p_1 - p_2)} \qquad [2]$$

in which w = weight of flow in lb per sec, p_1 = initial pressure in psi abs, p_2 = back pressure, v = specific volume of saturated water at p_1 in cu ft per lb, and ρ = density of saturated water at p_1 in lb per cu ft. The equation in this form applies readily to the flow of saturated water through sharp-edged orifices, and values for the orifice coefficient were found to be approximately the same as those for 70 F water. As in the case of cold water, the orifice

[5] "Thermodynamics," by J. E. Emswiler, First edition, McGraw-Hill Book Company, Inc., New York, N. Y., 1921, p. 225.

"The Leakage of Steam Through Labyrinth Seals," by Adolph Egli, Trans. A.S.M.E., vol. 57, 1935, pp. 115–122.

coefficient decreases with an increase in pressure differential (head) and orifice diameter. There is also some indication that the coefficient decreases as the initial temperature (saturation pressure) increases. The data are not complete enough, however, to make it possible to determine the numerical effect of a variation in orifice diameter or initial temperature. No attempt was made to correlate the orifice coefficients with respect to the diameter ratio since it was thought to be an unwarranted refinement in the design of throttling orifices for which the diameter ratios are usually low. Fig. 13 gives average values of the orifice coefficient for different values of differential head. These are considered sufficiently accurate for many design purposes, without correcting for effect of orifice diameter, initial temperature, or diameter ratio.

Equation [2] and the orifice coefficients given in Fig. 13 may be used also to calculate the flow of saturated water through an orifice when there is a static head. In this case, however, the density of the water depends upon the temperature rather than the pressure at the orifice.

Use of Single-Stage Orifices for Draining Condensate From Feedwater Heaters

The two important functions performed by a float trap on a feedwater heater are draining the heater and maintaining the

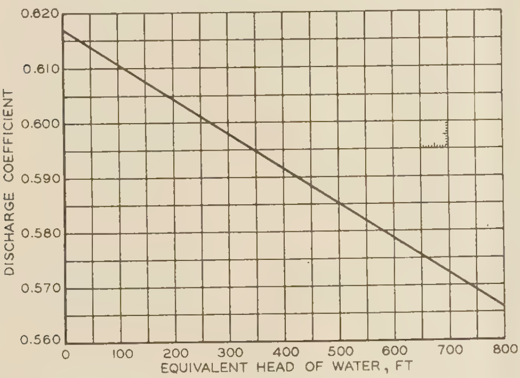

Fig. 13 Discharge Coefficient for Flow of Saturated Water Through Sharp-Edged Orifices

proper pressure differential between the heater and the drain receiver. The float-operated drainer performs these functions by means of a balanced valve, actuated by a float, which responds to changes in hot-well level determined by the quantity of drains. However, these two functions can be performed also by a fixed opening such as a pipe or an orifice and the size of the opening may be varied over a wide range without causing operating difficulty or appreciable thermal loss. For instance, the float-operated trap on a certain heater failed to function at full load on the turbine because of the relatively high pressure drop across the trap valve and, in order to keep the turbine in operation, the heater was drained for several weeks through a 3-in. by-pass around the trap. In this case, since the 3-in. line was several times as large as needed for normal flow, considerable steam cascaded to the next lower pressure heater with the drains. There was, however, no noticeable decrease in heater pressure or cycle efficiency.

The trap on this heater has since been replaced by a $^3/_4$-in-diam single-stage orifice which has been operating successfully for several months. The size of this orifice was determined for full-load condition on the turbine by use of the data given in this paper. At full load practically no steam is passed by the orifice but for two-thirds load, it is estimated that 0.015 lb of steam is passed with each pound of condensate. The method of determining the orifice diameter and the amount of steam passed at reduced loads is given in detail in the Appendix to this paper.

Whether the turbine cycle efficiency will be affected adversely by cascading some steam between heaters at partial loads depends upon the relative energy drop from the throttle to each extraction point and to the condenser, the method of returning the heater drains to the feedwater circuit, and the relative amount of partial-load operation.

Use of Orifices in Series With Traps

In cases which require the use of float-operated drainer traps to eliminate thermodynamic losses, due to cascading steam at partial loads, it may happen that an orifice installed on the downstream side of the trap will overcome certain operating difficulties with the trap. In some instances, the large pressure difference between heaters at high turbine loads may cause so much unbalance in the trap valve that the float can no longer operate the valve with the result that the heater floods. When an orifice is installed in series with a trap, the pressure drop between the heaters is divided between the trap and the orifice. By properly designing the orifice, the drop across the trap can be reduced sufficiently to permit the float to operate. A method of designing an orifice to operate in series with a trap is illustrated in Appendix.

While it is possible to use two orifices in series to drain feedwater heaters, it is doubtful whether in most cases they offer any advantages over the single-stage orifice. The design of two orifices to operate in series is much more difficult than the design of a single orifice and, in so far as the authors know, the quantity of steam passed at reduced loads on the turbine can be determined only by a cut-and-try method which is both involved and tedious.

Conclusions

When saturated water flows through a sharp-edged orifice, no flashing occurs until after the water is through the orifice and, contrary to the theory which is based on a change of state, no critical-pressure condition is evident. The quantity of saturated water that will flow through a sharp-edged orifice for given pressure conditions can be calculated with sufficient accuracy by the formula used to determine the flow of cold (70 F) water through an orifice and the discharge coefficients found for saturated water are approximately the same as those generally used for cold water. When calculating the flow of saturated water through an orifice, it is important to remember that the value of the head to use in the formula is the equivalent head in feet of water, based on the pressure drop across the orifice and the density of the saturated water.

When a mixture of saturated steam and water flows through a sharp-edged orifice with given initial and back pressures a small variation in the amount of steam in the mixture within the range of 0 to 4 per cent has a considerable effect on the total weight of mixture and, consequently, on the weight of saturated water, flowing through the orifice. For mixtures in which the quantity of steam is greater than 4 per cent, a small increase or decrease in steam content has only a slight effect on the total weight of flow.

While sharp-edged thin-plate orifices may be used to drain feedwater heaters in place of float-operated traps, it is probable that for reduced loads some steam will cascade through the orifice with the drains. No general statement can be made at this time concerning the effect on the turbine cycle efficiency of cascading small quantities of steam between heaters. Every case should be decided on its own merits. The necessary study will include consideration of the relative energy drop from the throttle to each extraction point and to the condenser, of the method of returning the heater drains to the feedwater circuit, and of the relative amount of partial-load operation.

It is important to keep in mind that the data presented in this

paper were obtained from tests of sharp-edged thin-plate orifices and, therefore, do not apply to nozzles or short tubes. These data can be used to design a throttling orifice to drain a given amount of saturated or nearly saturated water from a receiver of higher pressure to one of lower pressure and to maintain a required pressure in the former. No attempt should be made to use these data to design a metering orifice. If an orifice discharges into a low-pressure receiver through a pipe, the pressure loss in the discharge pipe must be taken into account in establishing the pressure differential across the orifice. In many cases, the pressure drop across the orifice will be only a fraction of the total drop between receivers, since a large part of the total may be required in getting the flashing mixture of water and steam through the discharge pipe. A discussion of the flow of a mixture of saturated steam and water through pipes is beyond the scope of this study and will be offered in a subsequent paper.

Acknowledgment

The authors gratefully acknowledge the encouragement and help of Messrs. P. W. Thompson, Sabin Crocker, and W. A. Carter in preparing this report and of Messrs. E. L. Liedel and B. Griffin and the technical staff at the Delray Plant in collecting the test data.

Appendix

Nomenclature

The following nomenclature is used in this Appendix:

A = area of orifice, sq ft
C = orifice coefficient of discharge
d = diameter of orifice, in.
g = acceleration due to gravity, 32 fpsps
h = static head, ft
p = pressure, psi abs
Q = flow of saturated water, cfs
t = temperature of water, F
v = specific volume of saturated water, cu ft per lb
w = flow of saturated water, lb per sec
ρ = density, lb per cu ft

Single-Stage Orifice Design to Drain Feedwater Heater

When determining the size of an orifice for a particular installation, it is more convenient to use Equation [2] than the data presented in Fig. 7, and it is important to keep in mind that the quantity of condensate to be drained from the heater will vary for a given load on the turbine as much as ±5 per cent, depending upon the variation in feedwater flow. From an operating standpoint, it is better to design the orifice too large rather than too small, and in practice it is recommended that a hand-operated by-pass be installed to provide means for passing abnormal quantities of water in case of a split or broken heater tube.

In Equation [2] $\dfrac{w}{A} = \rho \times C \sqrt{2g \times \dfrac{144}{\rho}(p_1 - p_2)}$

assume the following values corresponding to full-load operation of a 75,000-kw turbine: w = 13.6 lb per sec; p_1 = 236 psi abs (at saturation temperature); $p_1 - p_2$ = 110 psi; ρ = 53.8 lb per cu ft; head $= \dfrac{144}{\rho}(p_1 - p_2) = \dfrac{144}{53.8} \times 110 = 294$ ft; $C = 0.598$ (refer to Fig. 13).

Therefore $\dfrac{w}{A} = 53.8 \times 0.598 \sqrt{2g \times 294} = 4450$ lb per sec per sq ft and $A = \dfrac{13.6}{4450} = 0.00306$ sq ft;

$$d^2 = \frac{4 \times 144 \times 0.00306}{\pi} = 0.561; \; d = 0.75 \text{ inches}$$

For two-thirds load on the turbine p_1 = 155 psi abs; p_2 = 85 psi abs; $p_1 - p_2$ = 70 psi; and ρ = 55.2 lb per cu ft.

Head $= \dfrac{144}{55.2} \times 70 = 183$ ft; $C = 0.605$; $\dfrac{w}{A} = 55.2 \times 0.605 \sqrt{2g \times 183} = 3620$ lb per sec per sq ft; and $w = 11.1$ lb per sec.

With the initial pressure and pressure differential at this load, the $^3/_4$-in. orifice is capable of passing 11.1 lb per sec of condensate. Actually from the heat-balance data, calculated on the basis of no steam flow from the heater, only 7.8 lb per sec of condensate is available; therefore, for the existing pressure conditions some steam will pass through the orifice. Curves A in Fig. 14, which were obtained by cross-plotting the data from Fig. 9 for a constant back pressure of 85 psi abs, show that with an actual flow of condensate of 2540 lb per sec per sq ft for 155 psi abs initial

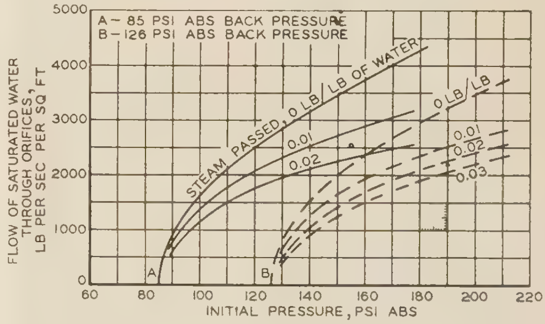

Fig. 14 Curves Illustrating Method of Determining Size of Orifices for Specified Design Conditions

pressure and 85 psi abs back pressure, approximately 0.015 lb of steam will pass through the orifice with each pound of condensate. After the approximate amount of steam passed through the orifice has been determined, a new heat balance can be made, taking into account the steam cascaded with the condensate.

By comparing the partial-load heat rate of the turbine cycle using orifices with the heat rate at the same reduced loads when only condensate is drained from the heaters, the feasibility of using orifices to drain the heaters for a particular turbine cycle can be ascertained.

Design of an Orifice to Operate in Series With a Trap

The example just given to illustrate the design of a single-stage orifice for draining a feedwater heater can also be used to show the design of an orifice to operate in series with a trap. In designing an orifice to operate with a trap, the first step is to choose a suitable intermediate pressure which in this case could be 190 psi abs. The following design data are now known: p_1 = 236 psi abs; p_i = 190 psi abs; p_2 = 126 psi abs.; and w = 13.6 lb per sec.

The drop in pressure through the trap from 236 to 190 psi abs causes part of the saturated water to flash into steam so that a mixture of 13.3 lb of water and 0.3 lb of steam or 0.0225 lb of steam per lb of water enters the orifice. Curves B in Fig. 14, which were obtained by cross-plotting and extrapolating the data from Fig. 9, show that, for an initial pressure of 190 psi abs and a back pressure of 126 psi abs, an orifice will pass 2200 lb of water per sec per sq ft plus 0.0225 lb of steam per lb of water.

or $\dfrac{w}{A} = 2200$ lb per sec per sq ft; $w = 13.3$ lb per sec

and $A = \dfrac{13.3}{2200} = 0.00605$ sq ft; $d = \sqrt{\dfrac{4 \times 144 \times 0.00605}{\pi}} = 1.06$ in.

Discussion

F. O. Ellenwood.[6] This paper is of intense interest to the writer and probably to all engineers who are at all concerned with the flow of fluids through orifices. The experimental data presented are valuable to those desiring to use orifices for draining feedwater heaters and also to those who are primarily concerned with the flow phenomena involved.

Even a hasty glance at Fig. 2 of the paper will show that the measured flow of saturated water through a sharp-edged orifice is several times the amount indicated by the so-called "theoretical curve" which is presumably Bottomley's theory. Unfortunately, this theory is not given in the paper, and the reference is not readily available.

It seems to the writer that many engineering papers are somewhat open to criticism when they simply refer to the so-called "theoretical values" without further explanation. In this particular case, the measured rates of flow are undoubtedly correct to a reasonable degree of accuracy, while the Bottomley theory, whatever it may be, is certainly far from complete.

Any theory concerning the flow of a fluid through an orifice, when that fluid is a saturated liquid at entrance, may be made simple or complicated, depending upon how complete it is. If the velocity and density at the orifice could be calculated, it would be a simple matter to determine the rate of flow for any known orifice area. As a matter of fact, however, the velocity and density at the orifice are not simply and accurately calculated. In all cases, the pressure in the orifice will be appreciably greater than that measured at a point considerably beyond the orifice. Just what portion of this total drop in pressure actually occurs in passing through a thin-plate orifice is hard to determine, but it is probably of the order of 25 or 30 per cent. If there is any transformation of liquid into vapor before passing entirely through the orifice, the complete theory then becomes further complicated due to the difficulties of calculating the change in density, the effect of the two-phase velocities, and the energy available to produce velocity in the orifice.

A. E. Kittredge.[7] In commenting on this excellent paper, the writer feels that additional emphasis be placed on the fact that the results refer to and are limited to a thin-plate orifice.

The difference observed between the results of this paper and those of Kittredge and Daugherty[8] are attributable to the difference between streamline flow and turbulent flow, respectively, as applied to this particular problem. We tested a nozzle subject to turbulent flow. The authors tested a thin-plate orifice subject to streamline flow. Complete turbulent flow represents one limit of the characteristic of the flow of saturated water and complete streamline flow represents the other limit of the characteristic of the flow of saturated water.

As stated, the difference between the results previously obtained and those now observed is a difference resulting from the characteristics of turbulent flow as opposed to the characteristics of streamline flow. In a broad sense the writer believes this to be true but there are other elements, namely, time, mass, energy, and heat-transfer rate which very likely influence the existence or nonexistence of a critical pressure in the flow of saturated water. In contrast to the flow of an almost perfectly elastic gas or vapor, allowance must be made for the fact that saturated water is initially a much denser fluid; that the ratio of the specific volume at critical pressure to the initial specific volume is much greater for saturated water than for any ordinary gas or vapor; that the flashing of water involves a complete change of state, not just a readjustment of the pressure-volume relation; that the change in state involves heat transfer from the mass surrounding a bubble to the bubble formed; and that more time and more energy per unit volume of fluid are required to redistribute the mass of saturated water from its initial volume to its greatly enlarged mixed volume after flashing, as compared to the time and energy required per unit volume to redistribute the mass of a relatively light and nearly perfectly elastic gas or vapor from its initial volume to its moderately increased volume at its critical pressure.

Turbulence and time contribute to all of these factors involved in the change of state between the initial pressure and the critical pressure. It is quite possible that, in the case of a thin-plate orifice, the completion of the change in state requires more time than elapses from the point of initial acceleration to the vena contracta of the orifice. In the authors' experiment, orifice sizes, ranging from $^1/_4$ in. diam to $^7/_8$ in. diam were used. Liquid velocities involved were of the order of 100 fps. If, in the case of a $^1/_4$-in-diam orifice, the acceleration of the liquid occurred in a distance of $^1/_4$ in. the time available for changing state during the period of acceleration and pressure reduction was approximately 0.0002 sec.

In February, 1934, in addition to the $^1/_4$-in-diam nozzle just described, we tested the application of our theory to a $^1/_2$-in-diam standard iron pipe about 10 ft long. The existence of critical pressure was determined by locating a pressure gage on the pipe at the discharge end. The results verified logical theory. This particular test setup certainly provided turbulent flow. The writer would suggest that a worth-while research would be to establish the relation between hydraulic diameter and length of pipe line or nozzle required to develop full turbulence and full flashing in accordance with thermodynamic theory.

The specific purpose of the present paper has been to determine flow rate through thin-plate orifices, but the inspiration for the investigation and the broad purpose of the paper is to determine the possibility of using orifices for the drainage of stage heaters. In this regard, it is felt that the authors have missed a remarkable opportunity by failing to investigate the characteristics of turbulent flow. For the particular purpose of draining stage heaters, a turbulent nozzle, having a capacity with saturated water substantially directly proportional to the absolute pressure, is much preferable to a thin-plate orifice, having a capacity varying as the square root of the absolute pressure. At the same time, the turbulent nozzle would have much lower capacity under saturated water and much wider range of control due to static head on the inlet side. If operating engineers are determined to eliminate interstage traps, they should attack the proposition on the basis of a turbulent nozzle located a considerable distance below the heater to be drained so that it may be subject to appreciable submergence on the inlet side. Submergence on the outlet side does not matter since flashing of the liquid will so reduce the density on the outlet side that, with reasonable pipe sizes, the pressure on the discharge side of the turbulent nozzle will never be higher than the critical pressure. Apparently then, the ideal automatic drainage arrangement for stage heaters would consist of a U-seal arrangement with the two legs of about the same diameter and the crossover connection between the two legs consisting of a small-diameter tube developing turbulent flow. The crossover tube between the two legs of the U-tube connection might be not less than one half the diameter of the respective leg.

The concluding paragraphs of the paper under discussion are rather vague regarding the applicability of the thin-plate orifice to the service of interstage draining of surface heaters but sug-

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[7] Chief Engineer, Cochrane Corporation, Philadelphia, Pa.

[8] "Discharge Capacity of Traps," by A. E. Kittredge and E. F. Daugherty, *Combustion*, vol. 6, September, 1934, pp. 14–19.

gests the possible use of a trap followed by an orifice. The writer would raise the question: If a trap is necessary why use the orifice? The trap in a full-open position is an orifice. This question is asked with particular reference to new installations and does not contemplate the possible expediency of assisting a defective trap.

S. P. Soling.[9] The authors have shown that, in the case of water, the actual flow of saturated liquid through orifices is greater than would be expected from theoretical considerations. Their graphical treatment clearly indicates how this occurs.

The same behavior has been noted for the refrigerants dichlorodifluoromethane (Freon-12) and ammonia. Orifices sized on the basis of equilibrium conditions in the orifice proved several times too large. Quantitative results are not available for comparison, as various amounts of vapors were passed with the liquid for different runs, the orifice readings being incidental to a test. No attempt was made to obtain data as comprehensive as those of the authors' who are to be congratulated on their clarification of a puzzling problem.

D. R. Yarnall.[10] In connection with this paper, it might be interesting to compare the conclusions with the trend of data obtained in the study of the flow of saturated and subcooled water through a rounded entrance orifice. This work was undertaken in the fall and winter of 1938-1939 by the research department of the writer's company and utilized a setup somewhat similar to that of the authors'. This orifice was connected to the mud-drum blowoff of a small high-pressure test boiler, using distilled feedwater.

The orifice used was 0.130 in. diam with a rounded approach of $^1/_8$ in. radius, followed by a tubular section $^1/_8$ in. long. A small pressure tap was drilled radially into the throat of the orifice to

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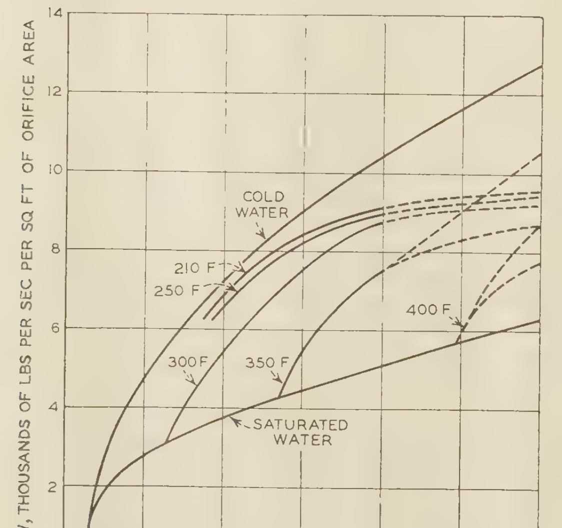

Fig. 15 Flow Curves for Saturated and Subcooled Water, Based on Actual Flow Through a Rounded Orifice

(Orifice, 0.130 in. diam, rounded to $^1/_8$ in. radius, and with throat, $^1/_8$ in. long.)

obtain some measurement of pressure conditions existing in the throat.

The results obtained were in general consistent with the trend of flow values disclosed in the paper, being greatly in excess of flow calculations based on thermodynamic equilibrium. From practical equality with cold-water flow at low heads, the ratio of the flow of saturated-water to cold-water flow reduced to approximately 0.6 at 50 psi abs; and further to approximately 0.5 at 100 psi abs, where the ratio to cold-water flow remained essentially constant up to 300 psi abs, the highest pressure reached. The actual numerical flow values obtained for saturated-water flow through the rounded orifice (in lb per sec per sq ft of orifice area), check within close limits of those given in the paper. However, when adjustment is made for differing coefficients, the flow through the rounded orifice will be considerably less in comparison.

On subcooled water, isothermal-flow curves, Fig. 15 of this discussion, showed a tendency to parallel the cold-water-capacity curve at pressures up to 200 psi abs. At higher pressures (between 200 and 300 psi abs) inconsistent results were obtained, the majority of observations indicating a marked drop in the rate of increase in flow with increased pressure. Due to capacity limitations of the apparatus and interruption of the investigation, this trend has not been conclusively substantiated to date. It would be interesting to know whether there was any indication pointing toward reduced capacity on subcooled water flowing through a sharp-edged orifice at pressures above 200 psi abs?

It is gratifying to note the growing consistency of accumulating data on the flow of saturated water and it is particularly helpful from the practical standpoint to find a simplified approach to a phase of the subject such as the authors of this paper have so clearly outlined. It is expected that work on the flow through rounded entrance orifices will be resumed, concluded, and the results submitted for publication shortly, in order that these findings may also contribute to the general subject matter.

Authors' Closure

The authors are extremely grateful for the interest shown by those who prepared discussions of their paper and feel that each discusser has added to the value of the paper by presenting his comments. The authors had hoped, however, that more interest would be shown in the possible application of these data in the design of orifices for specific uses, such as that of draining extraction feedwater heaters as described in the paper. The authors would like to emphasize again the importance of that part of their paper on the assumption that this point may have been missed by many who are interested in design work involving the flow of saturated liquids.

Regarding Professor Ellenwood's criticism of the paper because the theoretical treatment was omitted, it should be remembered that the authors were required to meet certain space limitations. The theory given in Bottomley's paper,[3] which was referred to by the authors, merely assumes thermodynamic equilibrium in the orifice. We realize, and regret, that Bottomley's paper is not widely distributed, but it is available in the Engineering Societies' Library, New York, N. Y. As usual, Professor Ellenwood's comments are welcome, and it is hoped he will be encouraged to continue the theoretical study of the phenomena of the flow of a saturated liquid through orifices and ultimately report his conclusions.

It is not clear to the authors what Mr. Kittredge means by "streamline" and "turbulent" flow in reference to this paper. Based on the usual Reynolds' number criterion, the flows reported are definitely in the turbulent region. If Mr. Kittredge's use of the term "turbulent" refers to a condition of flashing such as might occur in the downstream portion of a nozzle, or in the tail pipe immediately following an orifice, then the authors agree that

the results of the orifice tests would not apply. The paper specifically states that the orifice results do not apply to nozzles or short tubes. In view of their test results, however, the authors are at a loss to know how a flashing mixture of steam and water is to be obtained in a sharp-edged thin-plate orifice supplied only with saturated water. As shown by the results in the paper, if steam is passed through the orifice with the water, the capacity of the orifice is considerably less than if water alone is passed, however, this mixture is not the result of flashing in the orifice.

Mr. Kittredge's comments on several of the differences between saturated water and an elastic gas are pertinent to a theoretical analysis of the problem and add considerably to the background of the discussions. In addition to these theoretical factors, it probably is true that surface tension also plays an important part in retarding the flashing of a saturated liquid. This factor was pointed out by Prof. M. C. Stuart and others in oral discussion.

The authors are grateful to Mr. Kittredge for emphasizing the underlying purpose of the investigation. We feel that the practical applications of thin-plate orifices deserve serious consideration by designing and operating engineers. However, regarding the charge that "a remarkable opportunity had been missed by failing to investigate the characteristics of turbulent flow," the authors would like to say that an investigation of the flow of a flashing mixture of water and steam through pipes has been made and the results will be offered for publication later. These results were not included in the present paper because of lack of space and because the authors felt that each phase of this subject deserved the emphasis derived from a separate report.

Mr. Kittredge's suggestion of using a U-leg arrangement for draining feedwater heaters is basically sound, but the authors point out that one of the primary reasons for using orifices for draining heaters is that it is possible to install an orifice at the end of the cascade drain line and thus prevent erosion of elbows resulting from high velocities. This was mentioned in the "Introduction" of the paper as one of the important advantages of orifices over traps, and it is also one of the main advantages of the former over Mr. Kittredge's U-leg arrangement.

In answer to Mr. Kittredge's question: "If a trap is necessary why use the orifice?" it is obvious from his own qualifying statement following his question that he already knows the answer. As is well known to plant operators and trap manufacturers alike, there is nothing uncommon about the failure of a supposedly non-defective trap to function properly under some operating conditions even though it was "designed for the job." Where a trap needs assistance because the float is too small to overcome the unbalance in the trap valve, the use of an orifice in series is a simple, inexpensive expedient.

Mr. Soling's comments are most welcome in that they point out that saturated liquids other than water behave in the same general way as water when flowing through an orifice or valve into a region in which the pressure is lower than the saturation pressure.

Mr. Yarnall is to be commended for publishing some of his test data. These data supply some of the information on the characteristics of flow of saturated water through a nozzle or short tube which were lacking in the authors results and, in this respect, his discussion is a distinct contribution. The authors do not understand why the bellmouth tube should have a relatively smaller capacity than a sharp-edged orifice, after correcting for differences in the cold-water-discharge coefficients. It may be that the relatively small size of tube used in Mr. Yarnall's tests is responsible for the difference, either because of undisclosed factors associated with the small physical size or because of difficulties involved in making the laboratory determinations. The latter possibility is indicated by the inconsistent results at high pressure differences. The marked drop in rate of increase of flow with increased pressure differential may indicate a critical pressure condition in the tube. This may be what Mr. Kittredge refers to as a turbulent condition. It would seem to the authors, however, that this drop in the rate of increase of flow would have been more pronounced with saturated water than with subcooled water, which is contrary to the indications of Fig. 15 of Mr. Yarnall's discussion. In the authors' tests the reduction in discharge coefficient for increasing pressure differentials with subcooled water was essentially the same as shown in Fig. 13 for saturated water.

A recent communication from Prof. J. I. Yellott makes an interesting analogy between the so-called supersaturation in a rapidly expanding steam jet and the failure to flash in the saturated water jet. He says, "In a very rapid expansion, a substance in the liquid or vapor phase is apparently unable to change its phase rapidly enough to alter the flow characteristics of nozzles or orifices."

Train Acceleration With Steam Locomotives

By L. B. Jones,[1] Altoona, Pa.

This paper presents a study of the relation between the cylinder tractive force of high-speed steam locomotives and the energy required to accelerate trains of differing weights at various rates. Mathematical formulas for computing time and distance required for acceleration are presented in an Appendix. Consideration is given to some of the more important factors which limit cylinder tractive power.

The mathematics of acceleration of railway vehicles has been fully discussed in various textbooks and also in previous papers presented before the Society; but for convenience of ready reference the fundamental concepts are herein reviewed. Weights of locomotive and train are expressed in tons; and acceleration is expressed in terms of miles per hour per minute or per mile, to conform with the customary statistics of train schedules; in distinction to the common formulas of physics and mechanics expressed in terms of pounds, feet, and seconds. By this means, it is hoped to record data which will be helpful to operating officers as well as to designing engineers. Because the energy of acceleration varies with the square of the velocity, but only directly with the weight of the train, simple arithmetical proportion is deficient when comparing different locomotives or different weights of trains at the higher speeds, and a graphic analysis is most useful to show what actually takes place.

The force available for acceleration in the cylinders of the ordinary two-cylinder steam locomotive is expressed by the well-known formula

$$T = \frac{C^2PS}{D}$$

where T = cylinder tractive effort, lb
C = mean diameter of the cylinders, in.
P = mean effective pressure, psi
S = piston stroke, in.
D = diameter of drivers, in.

The formula contains three fixed dimensional values and only one value subject to variation with speed, i.e., the mean effective pressure. It therefore follows that, as this value is maintained or increased, the cylinder tractive force will be maintained or increased; which is the only force, on level track, available to accelerate the train. Therefore, the ability of a steam locomotive to accelerate a train rests with its mean effective pressure.

In Figs. 1 and 2 are shown cylinder-horsepower and cylinder-tractive-force versus speed curves for several locomotives, in which curve A represents a Pacific-type locomotive which has been performing satisfactorily in main-line passenger service for several years. For purpose of this study, curves B, C, D, and E represent successive improvements in the mean effective pressure of this same locomotive, but for simplification the studies of train acceleration are confined to the minimum or present locomotive A, and the maximum or improved locomotive E. The latter has been selected as the maximum locomotive for this study because, as shown in Fig. 1, the cylinder horsepower is maintained almost constant from 60 to 100 mph. While it is sufficiently in advance of current steam-locomotive practice to be called a "maximum" locomotive, it is by no means an "ultimate" locomotive because, if the mean effective pressure could be still better maintained as the speed increases, the horsepower would actually increase with the speed above 60 mph, as it now does below that speed.

If yet greater power must be obtained, a glance at the formula previously given will show that the only recourse is redesign, or increase of one or more of the dimensional values. The advantages of improving the present locomotive, as compared with design changes, involving increased weight and capital investment, are illustrated by the curves in Figs. 4 to 8, inclusive, which have been developed on the assumption that improved locomotive E has been produced from present locomotive A without any increase in weight.

For this study, three trains weighing respectively 800, 1000, and 1200 tons behind the tender have been assumed, and their gross resistances, based on the Davis formulas, are shown in Fig. 3. For simplicity, all calculations have been based on straight level track; the effect of grades, plus or minus, may be added or subtracted, and a similar correction may be made for curves. For purposes of comparing two or more locomotives, the assumption

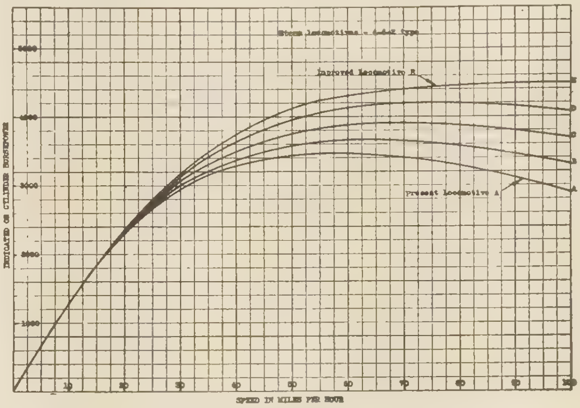

Fig. 1 Speed and Horsepower Curves

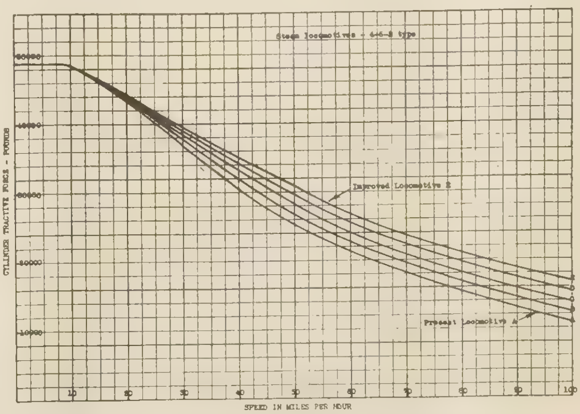

Fig. 2 Speed and Tractive Force Curves

[1] Engineer of Tests, The Pennsylvania Railroad. Mem. A.S.M.E.

Contributed by the Railroad Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

of level track will answer as well as any other condition which might be selected.

The curves involving speed, time, and distance were calculated from the tractive-force and resistance curves point by point and then verified by the mathematical formulas presented in the Appendix. In each case, the two methods checked very closely; and it is evident that acceleration curves can be constructed by the formulas which will reflect the effect of changes in the tractive-power curve on the performance of the locomotive. Since the curves are plotted for the minimum and maximum locomotives only, it is also evident that the performance of the intermediate locomotives, *B*, *C*, and *D*, can be studied from the curves by interpolation. It will be noted that the mathematical studies in the Appendix follow closely the methods of Professor Barrow (1).[2]

Fig. 4 compares present locomotive *A* with maximum locomo-

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

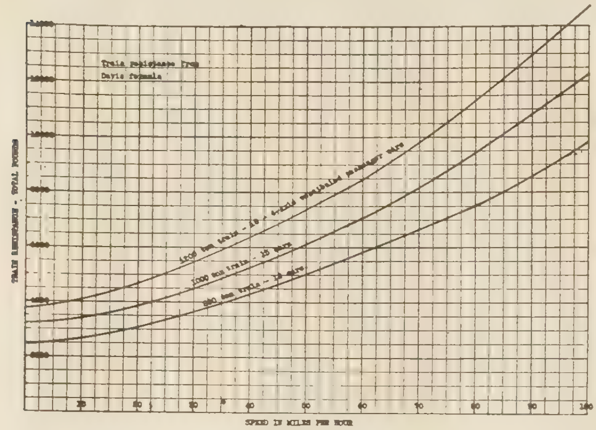

FIG. 3 SPEED AND TRAIN-RESISTANCE CURVES; LEVEL TRACK

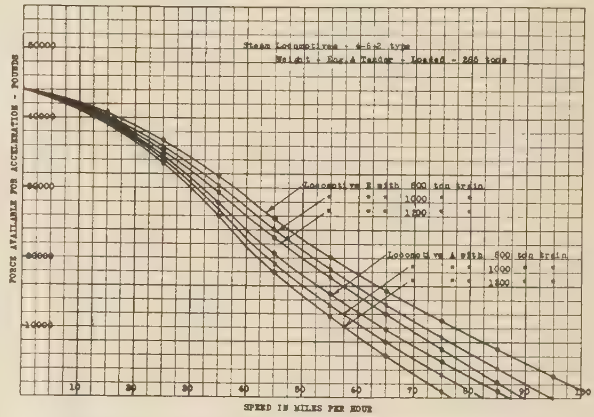

FIG. 6 SPEED AND ACCELERATING-FORCE CURVES

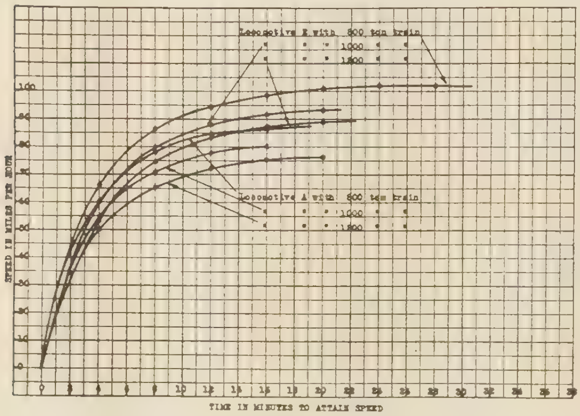

FIG. 4 TIME AND SPEED CURVES

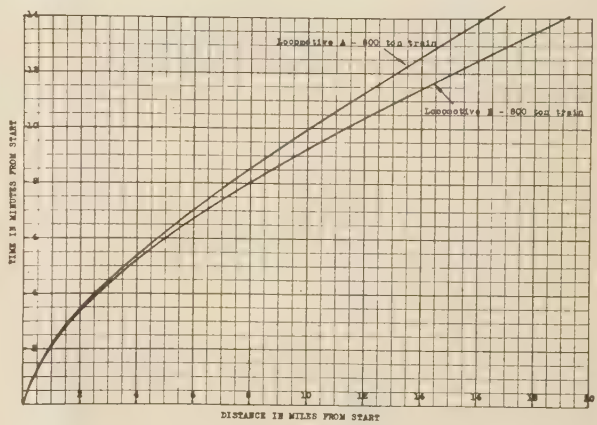

FIG. 7 TIME AND DISTANCE CURVES

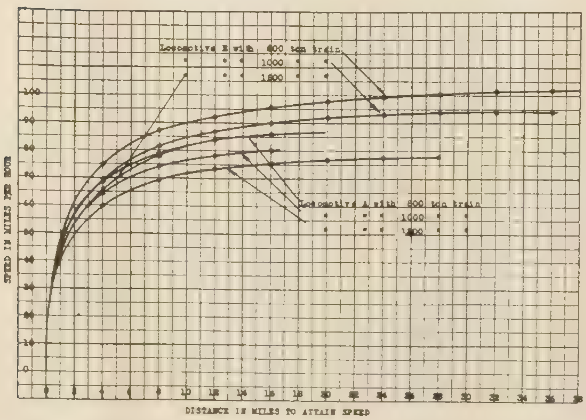

FIG. 5 DISTANCE AND SPEED CURVES

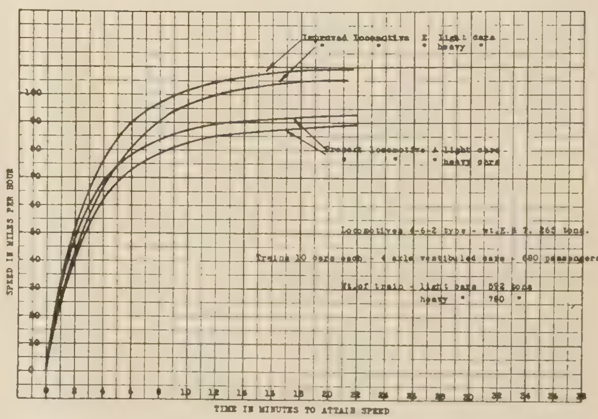

FIG. 8 LIGHT CARS VERSUS HEAVY CARS

tive E, hauling three different trains, and Fig. 5 illustrates the same comparison based on distance. It will be noted that the higher sustained horsepower of the improved locomotive results in a reduction of both time and distance required to attain a given speed.

These curves also serve to emphasize a point which must be borne in mind by operating officers, and that is the serious handicap of enforced slowdowns. Locomotive A with an 800-ton train requires approximately $2^1/_2$ min or $1^1/_2$ miles to attain 50 mph, and $6^1/_2$ additional min or $7^1/_2$ miles to attain 80 mph; so that, if the train is slowed from 80 to 50 mph, $6^1/_2$ min or $7^1/_2$ miles are required to resume the original speed. This should not be confused with elapsed time, which would include also time lost in slowing down and running at reduced speed, items not covered by this investigation.

Fig. 6 shows the force available for acceleration compared with speed. The locomotive has reached its maximum speed when the accelerating force becomes zero. To determine the maximum speed of the locomotives on a grade, it is only necessary to determine the grade resistance of the locomotive and train and draw a horizontal line at the corresponding value. The intersection of the curves with the line so drawn will show the maximum speed on the grade selected.

Fig. 7 shows a mathematical "race" between locomotives A and E. Starting from the same point, with trains of identical weight, it will be seen that, at the end of 12 min, locomotive E is 2 miles ahead of locomotive A; and the gap widens rapidly due to the more rapid acceleration of the improved locomotive.

Fig. 8 shows the effect of lightweight cars on the rate of acceleration. The weights of the two trains, with a given locomotive, are proportional to the time required to attain the same speed, and to the squares of the speeds attained in a given time. Therefore, it follows that, for a given maximum speed, the saving in schedule time by the lightweight train is confined to acceleration, and if there are no stops or speed reductions, the heavy train will require only a little more time to cover a given distance than the lightweight train. On the other hand, if there are numerous stops and slowdowns, the advantage of the lightweight train is multiplied.

Fig. 9 shows tractive-force curves for steam, electric, and Diesel locomotives of equivalent-nominal-horsepower rating. Steam locomotive E from previous studies is compared with assumed electric and Diesel locomotives, the continuous motor rating and the Diesel-engine rating being used for the electric and Diesel locomotives, respectively. It is recognized to be common practice to take advantage of the overload capacity of electric motors while accelerating, which is a distinct advantage for an electric locomotive drawing its power from a trolley; but the Diesel is limited by the capacity of its engine, and the overload possibilities of the steam locomotive are circumscribed by considerations of economy and good practice, at least in the preparation of train schedules. A direct comparison of locomotives having such different characteristics is impossible, but the curves serve to illustrate the relative capacities for accelerating trains. They also demonstrate that the steam locomotive, with moderate improvement, is capable of taking rank with the best motive-power units.

Fig. 10 illustrates an advantage of the improved locomotive with respect to the power output required for acceleration to a given speed. The kinetic energy of two trains of the same weight is the same for any speed; but the improved locomotive requires less time and distance to attain speed and, therefore, the energy required to overcome friction is less. Inasmuch as each locomotive would have to cover the same distance in actual operation, this saving during acceleration is theoretical rather than real.

Perhaps the greatest handicap of the steam locomotive is the

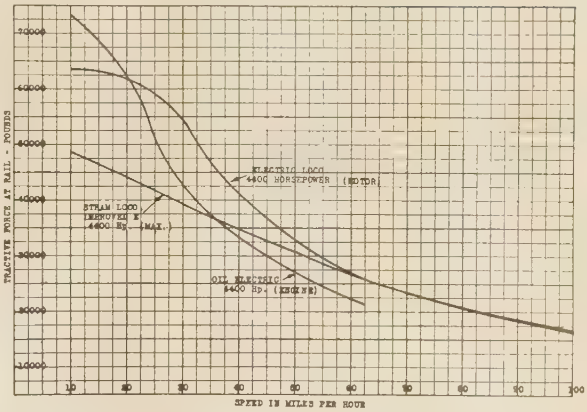

FIG. 9 TRACTIVE FORCE OF VARIOUS LOCOMOTIVES

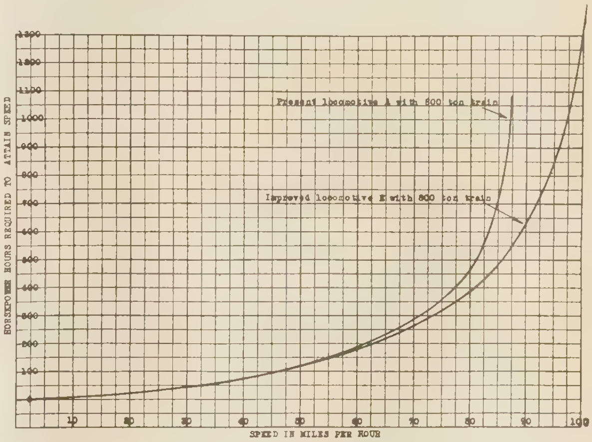

FIG. 10 SPEED AND HORSEPOWER HOURS

deep-rooted conservatism of American locomotive designers which has sentenced it to be a machine of two cylinders controlled by one valve apiece. The destructive dynamic forces which diverge from the line of power transmission, and the ineffective steam distribution have, it seems, become necessary evils, to be tolerated rather than faced. The author believes that these handicaps can be overcome, while still retaining for the steam locomotive the simplicity and flexibility which are its greatest assets. The maximum locomotive E has been assumed as a two-cylinder locomotive, conventional in all respects but valve action; and it seems probable that no satisfactory arrangement of cylinders to eliminate counterbalances and dynamic augments can be developed until a satisfactory valve action has been perfected. But regardless of the number and arrangement of the cylinders, the mean effective pressure will continue to govern the output; and the following additional assumptions have been made:

(a) Minimum pressure drop from boiler to steam chest. The superheater and pipes should afford free passage to the steam, for while steam which expands without doing work is raised in temperature, it is pressure which does the work in the cylinders.

(b) Adequate steam-chest volume. The opening of the admission valve results in equalization of pressures in the steam chest and cylinder; and at high speeds the surge of steam pressure from the pipes and header does not reach the steam chest until the valve has closed. The result is a maximum indicator-card pressure far below boiler pressure, and an average admission pressure yet lower. Meantime, the steam entering the chest at high

velocity builds up a surge pressure which may go 50 lb above boiler pressure at its peak, but drops to normal before the next valve opening. An adequate steam-chest volume is, therefore, essential to hold up the admission line on the indicator card; and also to insure a uniform velocity of steam through the superheater and pipes.

(*c*) Well-designed exhaust passages. The ideal passage would pass the steam to the nozzle at maximum velocity and minimum back pressure; but since both are impossible of attainment in the same passage, a uniform cross section of smooth proportions is desirable. Too large an exhaust passage operates as an expansion chamber which has to be choked at the nozzle to produce draft, with resulting high back pressure against the piston in the center of its stroke, where it is most damaging to the mean effective pressure.

(*d*) Large exhaust nozzle, which is only possible with an efficient front end.

(*e*) Proper steam distribution. This specification eliminates the one-piece reciprocating valve, and requires separate admission and exhaust valves so arranged that cutoff may be shortened without advancing the other valve events. Various valve arrangements which meet this requirement more or less perfectly are extensively used in Europe and we would do well to profit by their experience. Experiments now under way in this country may lead to successful results.

Appendix

Equations for Accelerating Force

When the tractive-force-speed curve of a locomotive is known, the curve for train resistance of any given train can be subtracted and the result is the accelerating-force-speed curve for the combined locomotive and train.

Let F = accelerating force for entire train, lb
a = acceleration, mphps = 1.467 fpsps
W = weight of train, including additional percentage to provide for energy of rotation, tons
V = speed, mph
L = distance traveled to reach any speed, miles

$$M = \text{mass of train} = \frac{W \times 2000}{32.2}$$

t = time to reach any speed, min
A, B, C, D, K_1, K_2, and K_3 are constants

Since
$$a = \frac{F}{M \times 1.467}$$

$$a = \frac{F \times 32.2}{2000 \times W \times 1.467}\ \text{mphps}$$

or
$$a = \frac{21.95F}{2000W} \quad \text{also,} \quad a = \frac{dv}{dt}$$

$$\frac{dv}{dt} = \frac{21.95F}{2000W} \text{ or } dt = \frac{2000W}{21.95F}dv$$

$$dt = \frac{91.1Wdv}{F}$$

The time, t to reach any speed is

$$t = \frac{91.1W}{60}\int\frac{dv}{F} \qquad [1]$$

The distance L traveled in miles to reach any speed is $dL = K_1Vdt$ where K_1, is the constant required to make the relation true.

$$K_1 = \frac{1.467}{5280} \quad \text{so} \quad dL = \frac{1.467}{5280}Vdt$$

$$dL = \frac{1.467 \times 91.1WVdv}{5280F} = 0.02533\frac{WVdv}{F}$$

or
$$L = 0.02533W\int\frac{Vdv}{F} \qquad [2]$$

The constant of integration required to make the equation true for a known condition will be called D.

Knowing the relations shown by Equations [1] and [2], it only remains to write the equation for speed V and tractive force F for the train, from which equations showing the time and distance to reach any speed can be derived.

The force of acceleration is maximum at starting. As long as the engine can be run in full gear, the accelerating force falls off from its original value by an amount which is proportional to the square of the speed. The general equation for this relationship is

$$F = F_0 - K_2V^2 \qquad [3]$$

where F_0 equals starting tractive force and K_2 = constant required for any particular curve. This condition holds for a passenger train until a speed of about 30 mph is reached, following which, changed cutoff and other conditions give quite a different curve. The curve changes from convex to concave in the 30-mph speed range (sometimes referred to as the critical range). Therefore, two equations are required for every speed-accelerating-force curve.

To simplify the actual calculations, Equation [3] was not used for the accelerating-force-speed curves but the equation was changed to

$$F = \frac{F_0}{1 + K_3V^2} \qquad [4]$$

It is obvious that the substitution of the value of F in Equations [1] and [2], as given by Equation [4], is much more easily handled than the substitution which Equation [3] gives. The substitution of Equation [4] in Equation [1] gives

$$t = \frac{91.1W}{60F_0}\int(1 + K_3V^2)dV$$

which can be very quickly solved.

The substitution of Equation [3] in Equation [1] gives

$$t = \frac{91.1W}{60}\int\frac{dV}{F_0 - K_2V^2}$$

which is much less convenient.

When constants K_2 and K_3 are chosen so that Equations [3] and [4] match at 0 and 30 mph, the intermediate points are close enough to give the correct time and distance for all speeds.

As an example, the following table shows how Equations [3] and [4] compare, for a case where the accelerating force equals 44,000 lb at starting and 36,000 lb at 30 mph. For Equation [3] the equation is

$$F = 44{,}000 - 8.88V^2, \text{ or } K_2 = 8.88$$

for Equation [4]

$$F = \frac{44{,}000}{1 + 0.0002468V^2}, \text{ or } K_3 = 0.0002468$$

Speed	Accelerating force, lb Equation [1]	Equation [2]
0	44000	44000
10	43120	43000
20	40480	40050
30	36000	36000
35	33100	33550

It is apparent that for all practical purposes the two equations are equivalent, so the one most readily usable should be chosen.

The curve for the higher speeds, which is the more important, is found by two steps. The first step is to note that if a tangent TT_1 to the tractive-force-speed curve SS_1 is drawn, the difference in ordinates of the tangent and the curve, when plotted on rectangular coordinates, forms a very good parabola, Fig. 11. The next step is to write the equation of this parabola. When the equations of the tangent and parabola are added, the result is the equation of the tractive-force curve. This curve always has the form

$$AV^2 + BV + C = F \quad \ldots\ldots\ldots\ldots [5]$$

In most cases, where a well-chosen point of tangency is used, the curve of Equation [5] fits the actual curve very closely. Almost any point of tangency gives good results.

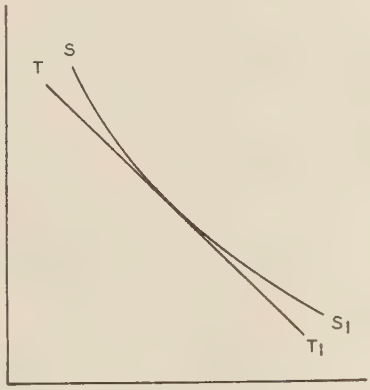

Fig. 11

Substituting the values of F in Equations [4] and [5] in Equations [1] and [2] gives

$$t = \frac{91.1W}{60F_0}\int (1 + K_3V^2)dV$$

$$t = \frac{1.5166W}{F_0}\left(V + \frac{K_3V^3}{3}\right) \quad \ldots\ldots\ldots\ldots [6]$$

which holds good to 35 mph, above which

$$t = \frac{91.1W}{60}\int \frac{dv}{AV^2 + BV + C}$$

$$t = \frac{91.1W}{60}\,\frac{2.303}{\sqrt{B^2 - 4AC}}\log\left(\frac{2AV + B - \sqrt{B^2 - 4AC}}{2AV + B + \sqrt{B^2 - 4AC}}\right) + D \ldots [7]$$

which holds good above 30 mph, and allows a short overlap with the curve of Equation [6] in the critical range.

The time t is calculated by Equation [6] for the lower speeds; then D is calculated to make the time t correct at 35 mph in Equation [7].

For distance L

$$L = 0.02533W \int \frac{Vdv\,(1 + K_3V^2)}{F_0}$$

from Equations [2] and [4]

$$L = \frac{0.02533W}{F_0}\int V(1 + K_3V^2)dv$$

$$L = \frac{0.02533W}{F_0}\left(\frac{V^2}{2} + \frac{K_3V^4}{4}\right) \quad \ldots\ldots\ldots\ldots [8]$$

which is good up to 35 mph. For high speeds

$$L = 0.02533W \int \frac{Vdv}{AV^2 + BV + C}$$

$$L = \frac{2.303 \times 0.02533W}{2A}\left(\log(AV^2 + BV + C) - \frac{B}{\sqrt{B^2 - 4AC}}\log\frac{2AV + B - \sqrt{B^2 - 4AC}}{2AV + B + \sqrt{B^2 - 4AC}}\right) + D$$

It is noted that $AV^2 + BV + C = F$ at any speed so F_v can be substituted for $AV^2 + BV + C$ in the foregoing equation which gives

$$L = \frac{2.303 \times 0.02533W}{2A}\left(\log F_v - \frac{B}{\sqrt{B^2 - 4AC}}\log\frac{2AV + B - \sqrt{B^2 - 4AC}}{2AV + B + \sqrt{B^2 - 4AC}}\right) + D \ldots [9]$$

where D makes L correct for 35 mph, according to Equation [8].

Equations [7] and [9] would not be applicable if the quantity B^2 did not exceed $4AC$, but B^2 must always exceed $4AC$, otherwise no speed could be reached where there would not be some accelerating force. The maximum speed is found when

$$F = 0 \text{ or } AV^2 + BV + C = 0$$

then

$$V = \frac{2C}{\sqrt{(B^2 - 4AC)} - B} \quad \ldots\ldots\ldots\ldots [10]$$

Inspection of Equation [10] shows that if $4AC$ were greater than B^2, the maximum speed would have no limit.

The application of Equations [7] and [9] is quite simple because the quantity $\sqrt{B^2 - 4AC}$ which is easily computed repeats itself so frequently. The quantity $(AV^2 + BV + C)$ is really F_v at any speed, and the quantities $B \pm \sqrt{B^2 - 4AC}$ for a large group of curves can be conveniently arranged in a table. Then to obtain Equations [7] and [9], it is simply a matter of substituting, and calculating the particular constants of integration, which give the correct time and distance at 35 mph, as obtained from the more elementary Equations [6] and [8].

Ten cases were solved along the foregoing lines. They were for trains hauled by the present passenger locomotive A and the improved passenger locomotive E, using 800-, 1000- and 1200-ton trains, a loaded 10-car passenger train made up of heavyweight cars, and a loaded 10-car passenger train made up of lightweight cars.

The following table shows the make-up of the trains:

Condition	No. of cars	Weight of cars, tons	Gross weight of train, tons	(W) weight, allowing for kinetic energy of wheels and axles, tons
1	12	800	1065	1098
2	15	1000	1265	1303
3	18	1200	1465	1507
4	10	592	858	891
5	10	780	1045	1077

The equations for the various curves may be listed as follows:
Case 1, condition No. 1; present locomotive *A*:

Below 35 mph

$$F = \frac{44000}{1 + 0.00045V^2}$$

$$t = 0.03785V + 0.00000568V^3$$

$$L = 0.000316V^2 + 0.000000071V^4$$

Above 30 mph

$$F = 2.9V^2 - 869V + 54000$$

$$t = 10.66 \log \frac{1228 - 5.8V}{510 - 5.8V} - 4.014$$

$$L = 11.02 \log F_v + 26.68 \log \frac{1228 - 5.8V}{510 - 5.8V} - 62.34$$

Case 2 Condition No. 2; present locomotive *A*:

Below 35 mph

$$F = \frac{44000}{1 + 0.000455V^2}$$

$$t = 0.045V + 0.00000684V^3$$

$$L = 0.000375V^2 + 0.000000085V^4$$

Above 30 mph

$$F = 4.21V^2 - 1069.5V + 59530$$

$$t = 12.07 \log \frac{1447 - 8.42V}{692 - 8.42V} - 3.68$$

$$L = 9.017 \log F_v + 25.55 \log \frac{1447 - 8.42V}{692 - 8.42V} - 51.22$$

Case 3 Condition No. 1; improved locomotive *E*:

Below 35 mph

$$F = \frac{44000}{1 + 0.00031V^2}$$

$$t = 0.03785V + 0.0000039V^3$$

$$L = 0.000316V^2 + 0.000000049V^4$$

Above 30 mph

$$F = 3V^2 - 887V + 59350$$

$$t = 14.06 \log \frac{1160 - 6V}{614 - 6V} - 3.72$$

$$L = 10.663 \log F_v + 34.64 \log \frac{1160 - 6V}{614 - 6V} - 60.44$$

Case 4 Condition No. 2; improved locomotive *E*:

Below 35 mph

$$F = \frac{44000}{1 + 0.00036V^2}$$

$$t = 0.045V + 0.0000054V^3$$

$$L = 0.000375V^2 + 0.0000000675V^4$$

Above 30 mph

$$F = 2.26V^2 - 804V + 55770$$

$$t = 12.07 \log \frac{1181.4 - 4.52V}{426.6 - 4.52V} - 5.19$$

$$L = 16.8 \log F_v + 35.79 \log \frac{1181.4 - 4.52V}{426.6 - 4.52V} - 95.47$$

Case 5 Condition No. 3; present locomotive *A*:

Below 35 mph

$$F = \frac{44000}{1 + 0.00052V^2}$$

$$t = 0.052V + 0.000009V^3$$

$$L = 0.000435V^2 + 0.000000112V^4$$

Above 30 mph

$$F = 4.883V^2 - 1142V + 59400$$

$$t = 13.89 \log \frac{1521.5 - 9.766V}{762.5 - 9.766V} - 4.02$$

$$L = 8.992 \log F_v + 27.09 \log \frac{1521.5 - 9.766V}{762.5 - 9.766V} - 51.04$$

Case 6 Condition No. 3; improved locomotive *E*:

Below 35 mph

$$F = \frac{44000}{1 + 0.000417V^2}$$

$$t = 0.052V + 0.0000072V^3$$

$$L = 0.000435V^2 + 0.00000009V^4$$

Above 30 mph

$$F = 0.5332V^2 - 607.7V + 49854$$

$$t = 10.28 \log \frac{1120.5 - 1.0664V}{94.9 - 1.0664V} - 11$$

$$L = 82.37 \log F_v + 97.62 \log \frac{1120.5 - 1.0664V}{94.9 - 1.0664V} - 491.57$$

Case 7 Condition No. 4; present locomotive *A*:

Below 35 mph

$$F = \frac{44000}{1 + 0.0004V^2}$$

$$t = 0.0305V + 0.00000407V^3$$

$$L = 0.000254V^2 + 0.000000051V^4$$

Above 30 mph

$$F = 2.52V^2 - 795.8V + 51860$$

$$t = 9.305 \log \frac{1128.2 - 5.04V}{463.4 - 5.04V} - 3.59$$

$$L = 10.23 \log F_v + 24.4 \log \frac{1128.2 - 5.04V}{463.4 - 5.04V} - 57.71$$

Case 8 Condition No. 5; present locomotive *A*:

Below 35 mph

$$F = \frac{44000}{1 + 0.000395V^2}$$

$$t = 0.0372V + 0.0000049V^3$$

$$L = 0.000323V^2 + 0.000000061V^4$$

Above 30 mph

$$F = 3V^2 - 870V + 54000$$

$$t = 11.42 \log \frac{1200 - 6V}{540 - 6V} - 3.94$$

$$L = 10.468 \log F_v + 27.60 \log \frac{1200 - 6V}{540 - 6V} - 59.09$$

Case 9 Condition No. 4; improved locomotive *E:*

Below 35 mph

$$F = \frac{44000}{1 + 0.00031V^2}$$

$$t = 0.0305V + 0.00000316V^3$$

$$L = 0.000254V^2 + 0.00000004V^4$$

Above 30 mph

$$F = 1.76V^2 - 684V + 53800$$

$$t = 10.36 \log \frac{982.5 - 3.52V}{385.5 - 3.52V} - 4.14$$

$$L = 14.65 \log F_v + 33.57 \log \frac{982.5 - 3.52V}{385.5 - 3.52V} - 82.89$$

Case 10 Condition No. 5; improved locomotive *E:*

Below 35 mph

$$F = \frac{44000}{1 + 0.000318V^2}$$

$$t = 0.0372V + 0.00000395V^3$$

$$L = 0.000323V^2 + 0.000000049V^4$$

Above 30 mph

$$F = 2.4375V^2 - 792V + 57274$$

$$t = 14.35 \log \frac{1054.5 - 4.875V}{529.5 - 4.875V} - 4.11$$

$$L = 12.88 \log F_v + 38.87 \log \frac{1054.5 - 4.875V}{529.5 - 4.875V} - 72.40$$

BIBLIOGRAPHY

1 "Problems in Locomotive Acceleration," by A. C. Barrow, *Civil Engineering*, vol. 4, 1934, pp. 202–204.

2 "Steam Locomotive Design," by E. A. Phillipson, Locomotive Publishing Company, London, England, 1936.

3 "Principles of Locomotive Operation and Train Control," by A. J. Wood, McGraw-Hill Book Company, Inc., New York, N. Y., 1915.

4 "Air Resistance of High-Speed Trains," by O. G. Tietjens and K. C. Ripley, Trans. A.S.M.E., vol. 54, 1932, paper APM-54-23, pp. 235–246; also *Railway Mechanical Engineer*, vol. 106, 1932, pp. 18–22 and 61–63.

5 "The Resistance of Express Trains," by C. F. D. Marshall, published by *The Railway Engineer*, London, England, 1925.

6 "Aerodynamics of the Railway Train," by Alexander Klemin, *Railway Mechanical Engineer*, vol. 108, 1934, pp. 282–286, 312–316 and 357–360.

7 "Air Resistance of Railroad Equipment," by A. I. Lipetz, Trans. A.S.M.E., vol. 59, 1937, pp. 617–640.

8 "A.A.R. Passenger Locomotive Tests," Office Mechanical Engineer, Association of American Railroads, Chicago, Ill., February, 1939.

9 "Resistance of Lightweight Passenger Trains," by A. I. Totten, *Railway Age*, vol. 103, 1937, pp. 75–80.

10 "Atmospheric Resistance and Its Relations to the Speed of Railway Trains," by F. U. Adams, Rand McNally & Company, Chicago, Ill., 1892.

11 "Train Resistance and Power of Locomotives—Data and Tables," American Locomotive Company, New York, N. Y., 1906.

12 "Tractive Power and Hauling Capacity of Steam Locomotives," by John E. Muhlfeld, Boell and Freeman, New York, N. Y., 1924.

13 "Locomotive Tractive Effort in Relation to Speed and Steam Supply," by E. G. Young and C. P. Pei, Trans. A.S.M.E., vol. 57, 1935, pp. 335–341.

14 "Train Acceleration and Braking," by R. C. Jones, Trans. A.S.M.E., vol. 63, 1941, pp. 147–155.

15 "Main-Line Oil-Electric Locomotive for Roumania," Bulletin, International Railway Congress Association, vol. 20, December, 1938, pp. 1230–1244.

Discussion

H. B. Oatley.[3] This paper presents, in easily understandable form, some of the essentials in the consideration of increased steam-locomotive power at the higher speeds which are today necessary. The thesis set forth is one in which the writer can heartily concur. There is only one point to which he would like to direct attention. It is the stress laid upon enlarged steam areas throughout the path of the steam from the dome to the exhaust-nozzle tip.

Chapelon, in his comprehensive analysis[4] of the steam locomotive, the essential part of which has been so ably presented in English by Lawford H. Fry,[5] has indicated the increased capacity resulting from careful attention to the question of more ample steam path. In the path of the steam as here considered, the fact must be kept in mind that all of the passages, with the exception of the superheater units, are of metal having a lower temperature than that of the steam carried. The superheater units, however, in part surrounded by gases at temperatures of upward of 1500 F, must be recognized as severely stressed parts. The steam velocity through these units is an important factor in preventing rapidly destructive conditions and it is essential that adequate steam velocity must be provided.

In the early days of the superheater, when materials less heat-resistant were available, steam velocities with the single-loop superheater were many times noticeably low; and the rapid deterioration of superheater units soon led to modified designs which provided higher steam velocities and better protection to the metal of the units. Today, with improved alloy steels, it is the practice to design for lower steam velocities than would otherwise be permissible, but it must be recognized that, even with these improved materials, too low steam velocities will result in unduly short life of this portion of the steam path. Like most important features in an engineering structure, there must be a compromise and all factors must be fairly evaluated. On one hand there is the desirability of the maximum output; on the other hand, the initial cost and maintenance, as well as the losses which may be incurred in repairs, must be considered.

L. K. Sillcox.[6] The author has presented, in intelligent and consecutive form, an example of the problem that is recurrent in both operating and mechanical departments of railways, i.e., the estimation of locomotive performance in terms of cars, tonnage, and speed. It is understood that the locomotives compared, designated A and E, are identical in principal dimensions and that the cylinder-horsepower performance of locomotive E has been achieved by securing a higher mean effective pressure in the cylinders by reducing pressure drop between superheater header and the cylinders, and by so modifying the valve events that an increase in negative work is not a function of short cutoffs at high speed.

The writer would expect that locomotive E indicates some hypothetical performance which locomotive A can never be re-

[3] Vice-President, The Superheater Company, New York, N. Y. Mem. A.S.M.E.

[4] "La Locomotive à Vapeur," by André Chapelon, published by J. B. Bailliere et Fils, Paris, 1938.

[5] "The Evolution of the Locomotive in France," by Lawford H. Fry, *Railway Mechanical Engineer*, vol. 112, 1938, pp. 473–475; vol. 113, 1939, pp. 1–5.

[6] First Vice-President, The New York Air Brake Company, Watertown, N. Y. Mem. A.S.M.E.

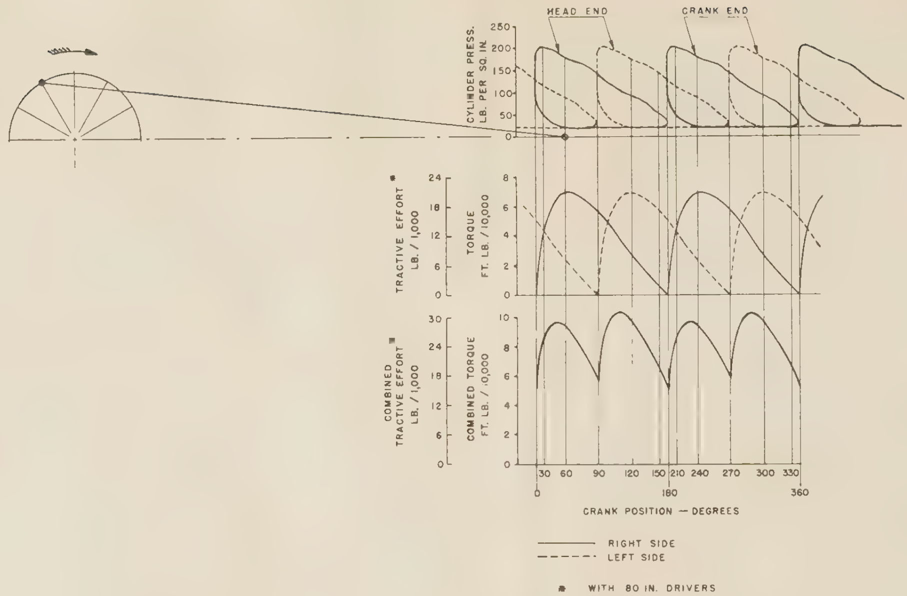

Fig. 12 Composite Indicator Card for Both Sides of Locomotive

constructed to produce economically. For instance, if the boiler will supply steam to deliver 4500 cylinder horsepower at 100 mph with a high degree of efficiency, its very size must be a burden to locomotive *A* which can do no better than a little more than 3400 hp at 60 mph. If maximum tractive effort is the same in either case, the same weight on drivers would serve either locomotive. The large boiler of locomotive *E* would demand more weight on idle axles.

Tractive effort expressed as

$$T = \frac{C^2PS}{D}$$

represents in simple form the average torque of a two-cylinder steam locomotive. Deduct total train resistance and it is a measure of accelerating capacity but is not an accurate and precise expression, because an equivalent mean effective pressure cannot be utilized to utmost advantage, bearing in mind the variation in instantaneous values of cylinder pressure during admission and expansion of steam, the effective lever arm of the couple which turns the main driving wheel on that side, and the manner in which the forces developed by two engines, operating 90 deg out of phase, combine. Increased mean effective pressure, however it may be obtained without proportionate increase in steam consumption or wider variation in extreme pressures, is desirable. The actual shape of the composite indicator card for both sides of the locomotive as it affects uniformity of torque also is important as disclosed by Fig. 12 of this discussion.

Since driving-wheel torque must be variable, there is a question as to the possible use which may be made of the peaks in accelerating force. At low speeds, surges are felt and at each surge some energy is wasted through movement of friction-draft-gear elements. This dissipation determines the number of cars through which the surge is discernible. If only the spring elements of the draft gears are affected a substantial part of the stored energy may be recovered. Maximum and minimum tractive efforts, developed from indicator cards, have been measured at 125 per cent and 76 per cent of the average values derived from the combined efforts of both engines at starting in full gear and at 30 mph, 73 per cent cut off. Obviously, no increase in mean effective pressure, secured by increasing instantaneous high pressures at crank positions which now are associated with the highest maximum torques, would be effective. At the same time, minimum torque occurs when the crankpins are at or near the center and quarter positions. At these points, any reduction in negative work, represented by compression and preadmission, is highly beneficial in the direction of useful and uniform tractive effort. Thus, a valve gear, which will provide independent timing of separate events, will be of considerable value in smoothing out the torque curve, quite independently of its effect upon mean effective pressure.

Author's Closure

Mr. Oatley has rightly called attention to the necessity for proper balance in all things, particularly in the steam locomotive. It is fully recognized that freer steam and gas passages should not be realized at the expense of either superheat or tube maintenance. It is rather the author's contention that our enthusiasm for evaporative surface and superheat has combined with increasing demands on the capacity of the locomotive as a whole in such a way as to swing the balance away from adequate areas. Steam and gas passages which were sufficient twenty or thirty years ago are too restrictive today, and our conceptions of proper balance must therefore submit to some overhauling.

Mr. Sillcox has pertinently pointed out that increased mean effective pressure without increase in boiler pressure can be brought about by eliminating negative work. Bearing in mind that the greatest improvement is realizable at the higher speeds, a large increase also comes from improved admission-valve action and later exhaust opening, which combine to increase the positive work available from the same amount of steam.

When this improvement in the cylinder cycle is supplemented by a higher average steam-chest pressure and a lower average exhaust back pressure brought about by refinements in other details of the machine, locomotive *E* is no longer a hypothetical case but becomes an attainable reality. The same boiler will serve because the steam delivered to the cylinders is the same in both cases.

Discharge Coefficients of Long-Radius Flow Nozzles When Used With Pipe-Wall Pressure Taps

By H. S. BEAN,[1] S. R. BEITLER,[2] and R. E. SPRENKLE[3]

For the last six years, the Special Research Committee on Fluid Meters has been conducting a research on flow nozzles. In previous papers relating to the program, the Committee's plans for carrying out the research were outlined (1),[4] and some of the results obtained have been presented (2, 3, 4). In the present paper, the results of determinations of coefficients of long-radius nozzles when used with pipe-wall taps[5] are given, as based on a combination of separate analyses by the three authors. Later papers will extend the results to other conditions than those considered in this paper.

Introduction

As was pointed out in one of the earlier papers (1), there is no one laboratory with facilities for making the required tests on flow nozzles over the entire range of conditions which it was desired to cover. Furthermore, by distributing the tests among several laboratories, more or less overlapping would occur, which would tend to furnish information on the agreement between the results of tests by the different laboratories on the same nozzle; in some cases using the same sections of piping. In addition, such data would provide a basis for estimating the tolerance to be assigned to the coefficient values for use in commercial metering of fluids. Since it is impossible to determine, from the smoothed results as here given, the portions contributed by the different laboratories, it is appropriate to list them and to indicate the range of conditions covered by their tests, so that due recognition may be given to their several contributions. This is done in Table 1.

Method of Analysis

For the purpose of the present paper, the essential result from each test is the relation between the discharge coefficient and the Reynolds number. The discharge coefficient is defined (5) by

$$w = 0.525\,\frac{CD_2^2}{\sqrt{1-\beta^4}}\sqrt{(p_1-p_2)\rho_1}\;\ldots\ldots\ldots\ldots[1]$$

in which w = actual rate of flow, lb per sec
C = coefficient of discharge
β = diameter ratio, D_2/D_1
D_1 = inside diameter of approach section of pipe, in.
D_2 = diameter of nozzle throat, in.
p_1 = absolute static pressure on inlet side of nozzle, psi
p_2 = absolute static pressure on outlet side of nozzle, psi
ρ_1 = density of fluid based on inlet pressure and temperature, lb per cu ft

The Reynolds number R_d, applying to the nozzle throat diameter, is given by

$$R_d = \frac{V_2D_2\rho_1}{12\mu_1} \quad\text{or}\quad R_d = \frac{48\,w}{\pi D_2\mu_1}\;\ldots\ldots\ldots\ldots[2]$$

while R_D, applying to the pipe diameter, is given by

$$R_D = \frac{V_1D_1\rho_1}{12\mu_1} \quad\text{or}\quad R_D = \frac{48\,w}{\pi D_1\mu_1}\;\ldots\ldots\ldots\ldots[3]$$

V_1 = average fluid velocity in the pipe, fps

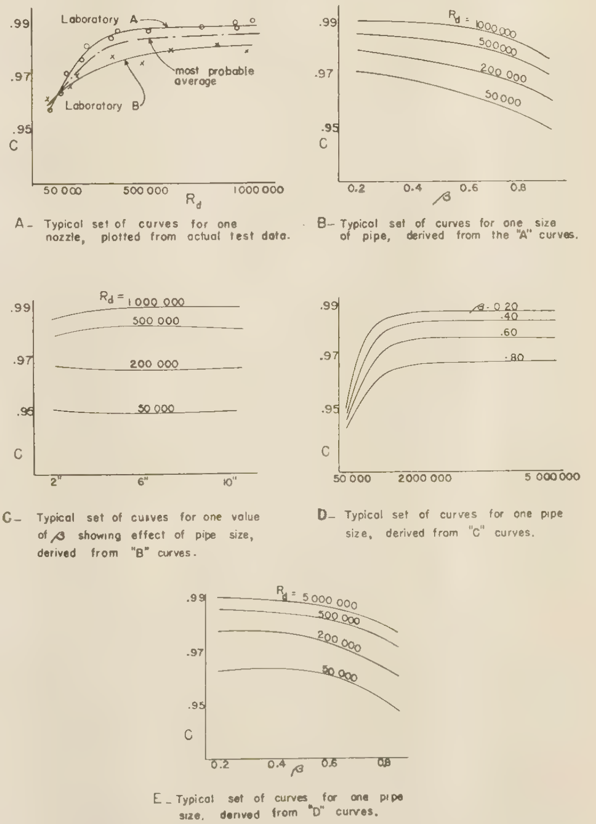

Fig. 1 Schematic Illustration of Graphical Method Used in Analyzing Flow-Nozzle Test Data

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[2] Associate Professor of Mechanical Engineering, The Ohio State University, Columbus, Ohio. Mem. A.S.M.E.

[3] Hydraulic Engineer, Bailey Meter Company, Cleveland, Ohio.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

[5] As here used, the term "pipe-wall taps" applies to pressure taps connected to the pipe wall with the inlet-pressure tap located 1 pipe diam ahead of the nozzle-inlet face, and the outlet pressure tap about $^1/_2$ pipe diam following the nozzle-inlet face, but in no case beyond the outlet end of the nozzle.

Contributed by the Special Research Committee on Fluid Meters and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

V_2 = average fluid velocity in nozzle throat, fps
μ_1 = absolute viscosity of fluid at inlet pressure and temperature, lb per ft per sec

From Equations [2] and [3] it will be seen that

$$R_D = \beta R_d \qquad [4]$$

a relation which will be referred to again.

The first step in the analysis of the results was to plot all the values of C determined for any one nozzle, against the corresponding values of the Reynolds number. A smooth curve was then drawn to represent the locus of the test points as closely as possible, as illustrated in A, Fig. 1. When more than one laboratory calibrated a given nozzle, the results from each laboratory were plotted separately, or with distinguishing marks. Smooth curves were then drawn to represent the results from each laboratory. Finally, a single curve was drawn to represent what the analyst believed to be the most probable average of all the data taken on that one nozzle. The result of this step was that an average curve was obtained for each nozzle tested.

Next, from the average curves for all nozzles tested in any one size of pipe, values of C were read off corresponding to some even value of the Reynolds number and these were plotted against the diameter ratio. Smooth curves were then drawn to connect the points for like values of the Reynolds number as in B, Fig. 1. This process was repeated for each different size of pipe in which nozzles were tested. As a matter of detail, it may be mentioned that it was usually necessary to cross-plot back and forth several times before sufficiently smooth curves were obtained in both the A and B plots.

For an even value of the diameter ratio, values of C at a given Reynolds, number were read from each group of B curves and plotted against pipe diameter (or the reciprocal of the pipe diameter). In this manner groups of curves were obtained for each selected value of the diameter ratio, as shown in C, Fig. 1. These curves show the change in the coefficient due to pipe size. While this effect was small and rather irregular, it was deemed desirable to give it some consideration in this analysis.

From these sets of curves, giving the effects of pipe size, values of the discharge coefficient were read for a single pipe size and plotted against Reynolds' number. Smooth curves were readily drawn through the points for the same diameter ratio, and thus for each pipe size was formed a family of curves giving coefficient against Reynolds' number for even diameter ratios, as illustrated in D, Fig. 1. Each of these families of curves was extrapolated to a Reynolds number R_d of 5,000,000, in order to meet the conditions in many present-day plants, particularly power plants where very high steam temperatures and velocities are used.

It will be noted that the curves in group D are similar to those of group A, but are much smoother and more evenly spaced. The analysis could have been ended here, and this set of D curves used to present the final results. However, it was the authors' belief that most users of flow nozzles would find it more convenient to have available curves in which the coefficient is plotted against diameter ratio. Accordingly, the final step was to make cross-plots from the D curves, thus giving curves of C for constant Reynolds' numbers against diameter ratio, as shown in E, Fig. 1, a separate set of curves being made for each pipe size.

As already mentioned, each of the authors made independent analyses of the data, following the general procedure outlined.

TABLE 1

Laboratories in Which Flow Nozzle Tests Were Made and Range of Conditions Covered

Laboratory	Size of Pipe inches	Number Nozzles Tested	Range of Diameter Ratios		Fluid	Range of Reynolds Numbers R_D min	R_D max	R_d min	R_d max	Reference Measurement by	Tests Made or Supervised by
Bailey Meter Co	3	13	.16 to	.80	water	6 600	870 000	22 000	1 090 000	volumetric tank	R. E. Sprenkle
Cornell University, Hydraulic Laboratory	8	1	.46		water	18 000	294 000	39 700	633 000	volumetric tank	Prof. E. W. Schoder,
	24	3	.24	.56	"	23 600	1 012 000	97 600	2 357 000	" "	A. N. Vanderlip
Cornell University, Sibley School of M.E.	3	4	.25	.60	water	3 000	400 000	12 700	670 000	weigh tank	Prof.s F. G. Switzer
	4	8	.35	.85	"	9 000	596 000	25 400	781 000	" "	& W. C. Andrae
General Electric Co. Turbine Research Dept.	2	8	.50	.84	steam	117 400	1 377 000	243 000	1 690 000	weighed condensate	B. O. Buckland, C. J. Walker
Ingersoll-Rand Co. Testing Dept.	4	2	.25	.50	air	6 000	59 500	24 000	119 000	impact tube	R. W. Johnson,
	16	9	.24	.36	"	51 000	386 000	200 000	1 700 000	traverse	R. E. Hunn
Massachusetts Institute of Technology, Dept. of M.E.	2	5	.125	.50	steam	4 400	128 600	34 900	257 200	weighed condensate	Prof. J. H. Keenan, W. J. Lindsey
National Bureau of Standards	2	20	.125	.84	water	155	620 000	1 290	768 000	weigh tank	H. S. Bean, F. C. Morey,
	3	17	.16	.86	"	426	628 000	2 900	734 000	" "	H. G. Wagner, H. P. Bean,
	4	16	.15	.85	"	500	403 000	3 260	480 600	" "	B. O'Connor
	8	10	.37	.86	"	10 700	783 000	28 400	986 000	volumetric tank	
Ohio State University, Dept. of M. E.	3	13	.16	.86	water	1 300	755 000	4 200	1 068 000	weigh tank	Prof. S. R. Beitler
	3 XH	6	.16	.86	steam	43 000	969 000	160 000	1 500 000	weighed condensate	
	6 XH	5	.54	.79	water	45 300	1 367 000	84 000	1 737 000	weigh tank	
	8 XH	6	.69	.82	"	57 700	1 952 000	82 000	2 672 000	volumetric tank	
	8	10	.37	.86	"	38 000	1 756 000	102 000	2 064 000	" "	
	10 XH	3	.82		"	226 000	1 626 000	277 000	2 110 000	" "	
	12	1	.64		"	112 000	1 418 000	175 000	2 205 000	" "	
University of California Dept. of M. E.	3	3	.4	.8	water	26 000	692 000	65 000	865 000	volumetric tank	Prof.s M. P. O'Brien
	4	8	.35	.85	"	78 700	754 000	127 000	868 000	" "	& R. G. Folsom
	8	9	.37	.56	"	62 000	549 000	160 000	984 700	" "	
University of Oklahoma, School of M. E.	2	22	.125	.84	oil	60	60 100	493	61 200	weigh tank	Dean W. H. Carson,
	3	19	.16	.86	"	48	49 500	290	69 700	" "	Prof. E. E Ambrosius
	4	14	.15	.85	"	6	37 800	40	50 700	" "	
University of Pennsylvania, Dept. of C. E.	4	5	.37	.85	water	4 000	852 000	10 800	990 000	weigh tank	Prof. W. S. Pardoe
	6	5	.25	.58	"	4 000	590 000	12 000	1 020 000	" "	
	8	2	.46	.83	"	7 600	1 650 000	15 000	2 000 000	" "	

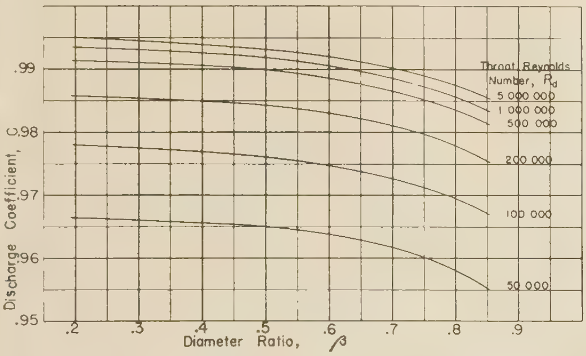

FIG. 2 DISCHARGE COEFFICIENTS OF LONG-RADIUS NOZZLES IN 2-IN. PIPE

(When inlet-pressure connection is 1 pipe diam preceding and outlet-pressure connection is $^1/_2$ pipe diam following plane of nozzle inlet.)

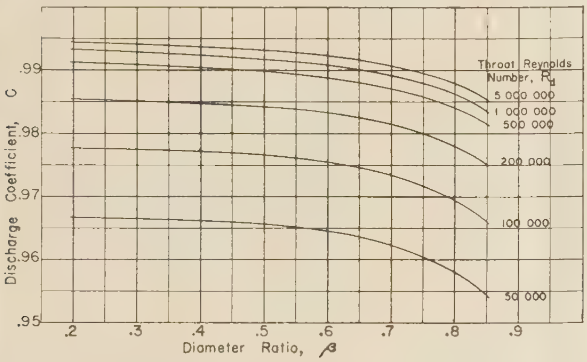

FIG. 3 DISCHARGE COEFFICIENTS OF LONG-RADIUS NOZZLES IN 3-IN. PIPE

(Refer to note Fig. 2.)

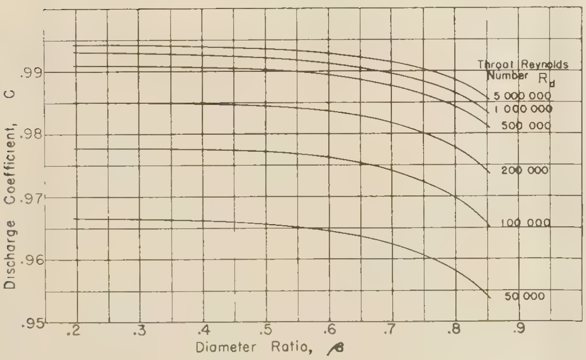

FIG. 4 DISCHARGE COEFFICIENTS OF LONG-RADIUS NOZZLES IN 4-IN. PIPE

(Refer to note Fig. 2.)

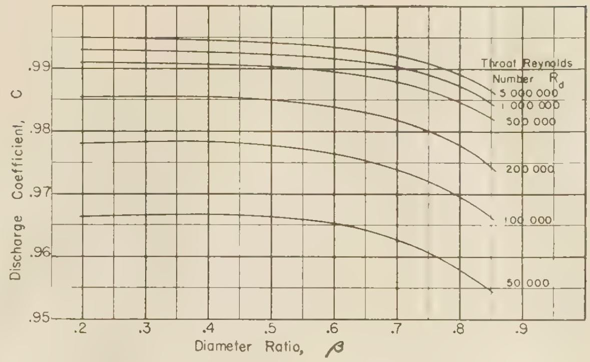

FIG. 5 DISCHARGE COEFFICIENTS OF LONG-RADIUS NOZZLES IN 6-IN. PIPE

(Refer to note Fig. 2.)

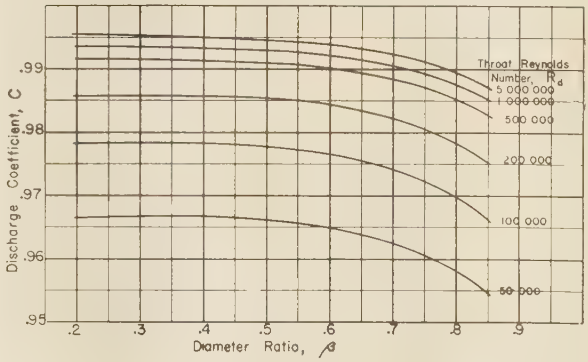

FIG. 6 DISCHARGE COEFFICIENTS OF LONG-RADIUS NOZZLES IN 8-IN. PIPE

(Refer to note Fig. 2.)

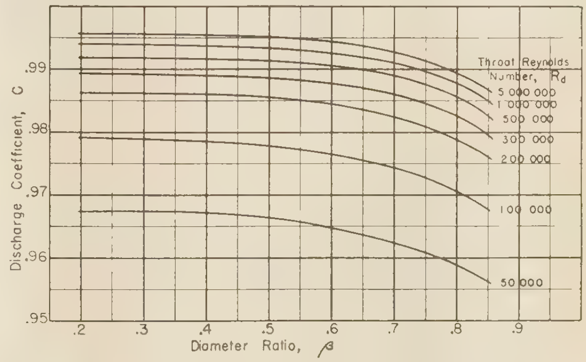

FIG. 7 DISCHARGE COEFFICIENTS OF LONG-RADIUS NOZZLES IN 10-IN. PIPE

(Refer to note Fig. 2.)

There was, however, more or less variation in the details of performing the several steps. For example, in plotting the original test data in the first step, either R_D or R_d may be used as the abscissa coordinate. Also, it was sometimes more convenient to use the logarithm of the Reynolds number, or to plot on semilogarithmic coordinate paper. Whichever Reynolds' number was used in the first step was used ordinarily throughout the succeeding steps. However, for the final comparison, only one was used, in this case R_d. This means that, when R_D was used in the first four steps, it was necessary to apply Equation [4] to the curves in D, Fig. 1, before cross-plotting to obtain the final set of E curves. This shifted each curve in D to the right, but by different amounts, the lower the diameter ratio the greater the amount of shift. It is evident that this alters the shape of the curves which are obtained in the final step by cross-plotting the D group of curves.

Furthermore, each author used his own estimation of the weight to be placed on the results from the different laboratories, particularly when two or more laboratories calibrated the same nozzle. These three individual sets of curves were then compared, line by line and the arithmetical mean taken as the basis for the curves herein presented.

Coefficients for Long-Radius Flow Nozzles, Tolerances, and Range of Application

The average curves obtained from the three analyses are shown in Figs. 2, 3, 4, 5, 6, and 7, applying to flow nozzles in 2-, 3-, 4-, 6-, 8-, and 10-in. pipes, respectively. These curves constitute the Committee's recommendation on the values of discharge coefficients for the long-radius or elliptical type of flow nozzle when the inlet-pressure connection is located 1 pipe diam preceding the inlet face of the nozzle, and the outlet-pressure connection is $^1/_2$ pipe diam following the inlet face of the nozzle.

The average difference between the results reported here and any one of the three individual analyses is about ± 0.2 per cent, while the maximum difference is about 0.5 per cent. However, between the results reported by different laboratories on tests of the same nozzle, the differences were as much as 1 per cent to 1.5 per cent, even when the same sections of pipe had been used. As a rule, the differences were larger at the low Reynolds numbers than at the high. Therefore, the authors suggest that, in the use of these nozzle coefficients, a tolerance (i.e., the probable range uncertainty) of ± 0.75 per cent be allowed at Reynolds' numbers R_d of 500,000 and over, with 3-in. pipe and larger. At lower values of R_d and with 2-in. pipe, the tolerance should be ± 1 per cent.

While the coefficient curves for all six pipe sizes have been extended up to a diameter ratio of 0.85, it is recommended that the use of diameter ratios in excess of 0.8 should be avoided wherever possible. The reasons for this recommendation are (*a*) the test data for diameter ratios over 0.8 were less extensive and more irregular than for the lower values of the ratio; (*b*) the slope of the curves is increasing rapidly in the 0.8 to 0.85 region, and the errors due to any uncertainty in the calculation of the diameter ratio or to reading the coefficient value from the curves will be greater than at the lower values of β.

A careful comparison of the curves for use with the different pipe sizes will show there is very little change in the value of the coefficient with pipe size, particularly with the larger pipes. Any further change in coefficient values with pipe size above a 10-in. pipe will probably be very slight. Therefore, it is suggested that for nozzles in pipes larger than 10 in. the coefficient curves for 10-in. pipe may be used without introducing any appreciable error.

TABLE 2 CONTRIBUTORS TO THE FLOW-NOZZLE RESEARCH PROGRAM

American Gas and Electric Company
Anaconda Copper Mining Company
Bailey Meter Company
Bethlehem Steel Company
Burlington Lines
Central Illinois Light Company
Central Railroad of New Jersey
Chicago, Milwaukee, St. Paul and Pacific Railroad Company
Connecticut Light and Power Company
Consolidated Edison Company of New York, Inc.
De Laval Steam Turbine Company
Detroit Edison Company
Duquesne Light Company
Ebasco Services, Incorporated
Economy Pumping Machinery Company
The Engineering Foundation
Foxboro Company
Gulf Oil Corporation of Pennsylvania
Gulf Research and Development Corporation
Humble Oil and Refining Company
M. W. Kellogg Company
New England Power Association
New England Power Service Company
Norfolk and Western Railway Company
Pacific Pump Works
Philadelphia Electric Company
Potomac Edison Company
Public Service Electric and Gas Company
Socony-Vacuum Oil Company
Standard Brands, Incorporated
Standard Development, Incorporated
Sun Oil Company
United States Steel Corporation
United States Works Progress Administration
Washington Gas Light Company
West Penn Power Company
Union Electric Light and Power Company
Utilities Coordinated Research, Incorporated
Westinghouse Electric and Manufacturing Company
Worthington Pump and Machinery Corporation
Youngstown Sheet and Tube Company

Acknowledgments

The conduct of any research project, and particularly so extensive a program as this has been, requires ample financial support. Too often the efforts to provide this support pass unnoticed, and that such may not occur in this case, we are pleased to record here that the carrying on of this program was made possible by the efforts of E. C. M. Stahl as chairman of the fund-raising subcommittee. Assisting him on this subcommittee were R. K. Blanchard, W. A. Carter, Paul Diserens, and L. K. Spink. For their repeated and generous response to the requests from this subcommittee particular acknowledgment is due The Engineering Foundation and the former Utilities Coordinated Research, Inc. The names of all the contributors of both funds and materials (other than laboratory testing facilities) are given in Table 2.

BIBLIOGRAPHY

1 "Research on Flow Nozzles," by H. S. Bean, *Mechanical Engineering*, vol. 59, 1937, pp. 500–502.

2 "Determining Flow Nozzle Contours," by F. C. Morey, *Instruments*, vol. 10, 1937, pp. 157–160.

3 "Some Results From Research on Flow Nozzles," by H. S. Bean and S. R. Beitler, Trans. A.S.M.E., vol. 60, 1938, pp. 235–244.

4 "Nozzle Coefficients for Free and Submerged Discharge," by R. G. Folsom, Trans. A.S.M.E., vol. 61, 1939, pp. 233–238.

5 "Fluid Meters, Their Theory and Application," Fourth edition, The American Society of Mechanical Engineers, 1937, Research Report, par. 156.

Discussion

W. A. Carter.[6] In connection with the graph showing the large spread of discharge coefficient of a certain flow-nozzle installation plotted against the Reynolds number, which appears in the authors' closure to this paper, it is noted that the installation involved corner taps.

The writer believes the authors will agree that such installations are much more susceptible to erratic performance than are those having the pressure taps located 1 pipe diam upstream and $^1/_2$ pipe diam downstream from the nozzle inlet, which latter is a type of installation with which the paper is concerned.

Corner taps are known to be dependent upon the width of the pressure-slot opening if the I.S.A. nozzle design is employed. A small variation in the thickness of the gaskets may seriously affect the nozzle coefficient.

W. W. Johnson.[7] Inasmuch as the data given in this paper supply authoritative values of nozzle-discharge coefficients over a wide region, where heretofore only meager information has been published, the paper will be greatly appreciated by engineers having flow problems to handle.

In order to clarify certain questions which have arisen will the authors supply answers to the following:

Are the coefficients given in the paper for low-ratio or high-ratio nozzles, as defined in "Instruments and Apparatus," part 5, chapter 4,[8] or both?

What rules were followed in regard to wall thickness of the test

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[7] Research Engineer, Turbine Engineering Department, General Electric Company River Works, Lynn, Mass. Mem. A.S.M.E.

[8] Information on Instruments and Apparatus, Part 5, Chapter 4, on Flow Measurment by Means of Standardized Nozzles and Orifice Plates, published by The American Society of Mechanical Engineers, 29 W. 39th St., New York, N. Y.

nozzles, and what would be the effect on the coefficients if relatively heavy walled nozzles were used?

Why have the coefficients not been given for throat Reynolds' numbers less than 50,000 and what, in general, was the result of the tests at the lower range of Reynolds' number?

W. V. KING.[9] Two high-ratio flow nozzles for unit No. 7 at Waterside Station No. 2 of the Consolidated Edison Company have been calibrated at Ohio State University by S. R. Beitler. The results of these calibration tests present an interesting discussion to this paper because they tend to substantiate the opinion of the authors that it is difficult to predict the coefficient of flow nozzles with diameter ratios above 0.8 to any reasonable degree of accuracy.

Both of these nozzles were designed for 700,000 lb steam flow per hr at 1300 psi pressure and 925 F temperature. To limit the differential head to 212 in. of water at rated steam flow, it was necessary to use flow nozzles with a diameter ratio of 0.8366.

The Reynolds number on these nozzles, corresponding to rated steam flow, is 10,000,000, equivalent to nearly 3 times the maximum Reynolds number obtained on test with water. Therefore it was necessary to extrapolate the calibration tests by a method which appears to give the most accurate determination of coefficient at design conditions. This basis of extrapolation is the relation that the logarithm of the actual water flow plotted against the logarithm of differential head is a straight line. The method of extrapolation is explained as follows:

1 All test points were corrected to a constant temperature; the differential head by a factor equivalent to the ratio of water densities, and the actual water flow by a factor equivalent to the square root of the water-density ratio. To minimize the magnitude of these corrections, the variation of water temperatures during test was confined to 5 deg for nozzle No. 59535 and to 3 deg for nozzle No. 59534.

2 The equation of a straight line expressing the relation between the logarithms of actual flow and differential head is log $Q = A + B \log h$, where A is the intercept and B is the slope of that line. The equation of the best line through the test points can be calculated most accurately by the method of least squares, applying the relations

[9] Assistant Engineer, Consolidated Edison Company, Inc., New York, N. Y. Mem. A.S.M.E.

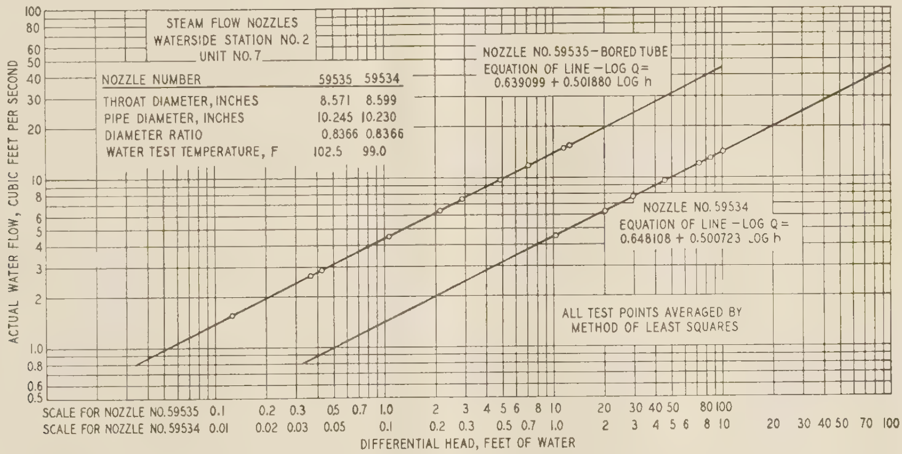

FIG. 8 LOGARITHMIC RELATION BETWEEN DIFFERENTIAL HEAD AND ACTUAL WATER FLOW

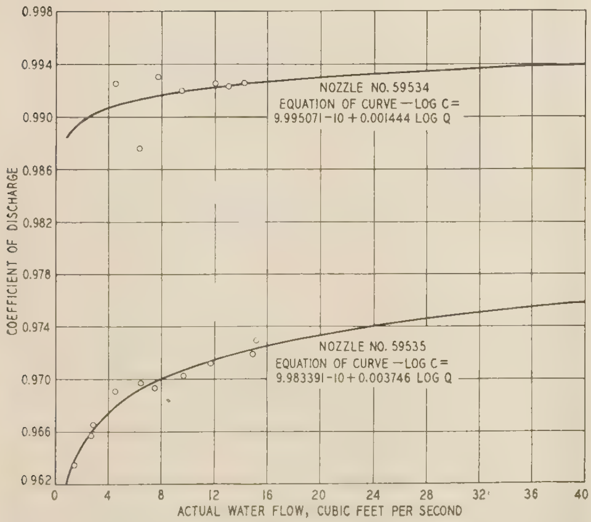

FIG. 9 COEFFICIENT OF DISCHARGE IN RELATION TO ACTUAL WATER FLOW

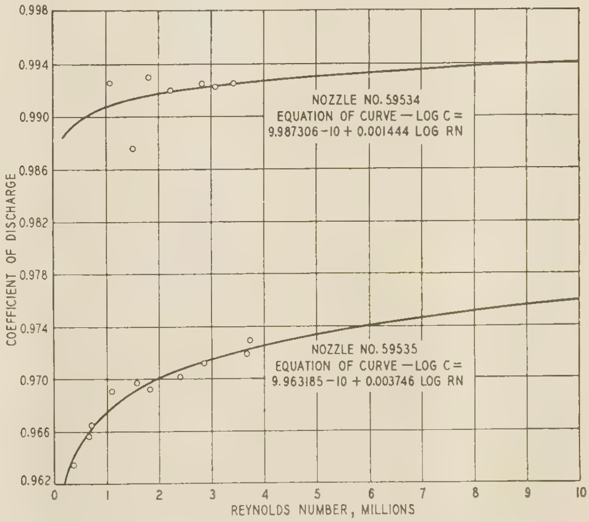

FIG. 10 COEFFICIENT OF DISCHARGE IN RELATION TO REYNOLDS' NUMBER

$$\Sigma B \log h + NA = \Sigma \log Q$$

$$\Sigma A \log h + \Sigma B (\log h)^2 = \Sigma \log Q \log h$$

where N is the number of test points.

3 Since the coefficient of discharge is equivalent to the actual water flow divided by the theoretical flow ($Q = K\sqrt{h}$), it is a simple mathematical procedure to determine the logarithm of the coefficient in terms of the logarithms of differential head, actual water flow, and Reynolds' number which are all straight lines.

The final calibration curves on these two nozzles are presented, first, to indicate the apparent accuracy of extrapolation and then to show the difference in discharge coefficient between two nozzles with the same diameter ratio of 0.8366.

The coefficient curves for nozzle No. 59535 installed in a bored tube do not vary from any test point by more than 0.1 per cent and the curve is the best average of the test points. Unfortunately, the coefficient curves for nozzle No. 59534, installed in an unbored tube, pass through four test points, fall under two test points by approximately 0.15 per cent, and lie above one test point by approximately 0.35 per cent. With the exception of one test point, the calibration-test accuracy falls within 0.15 per cent of the faired curves.

A comparison of these coefficient curves indicates that nozzle No. 59534 has a coefficient approximately 2 per cent greater than nozzle No. 59535, and also that the coefficient for nozzle No. 59534 is relatively flat compared to the coefficient for nozzle No. 59535.

Experience with high-ratio flow nozzles has prompted the following comments on paper:

1 The plotting of a smooth curve, representing the locus of test points, by an arithmetical average may be sufficiently accurate when the spread of points from the faired curve is not excessive and when the test covers the entire range of Reynolds' numbers. Whenever an extensive extrapolation is required, it is preferable to use the method of least squares in averaging the test points.

2 The straight-line relation between logarithms of flow and head discloses that a plot of the logarithm of coefficient against logarithms of differential head, actual water flow, and Reynolds' number also have a straight-line relation. Therefore, any direct plot of coefficient against Reynolds' number might introduce inaccuracy, provided the coefficient curve has a steep slope and test points cover a great range of Reynolds' number.

The authors are to be commended on presenting these extensive test data on flow nozzles in such a concise manner. Due to the tremendous amount of work necessary in preparing this paper, the methods used by the authors in extrapolating and averaging coefficient curves appear to be the most suitable that could be applied under the circumstances. This discussion has been presented to introduce additional data on the higher-diameter-ratio nozzles and to show what appears to be a more desirable method of extrapolating test data on coefficient of discharge to procure the greatest accuracy possible.

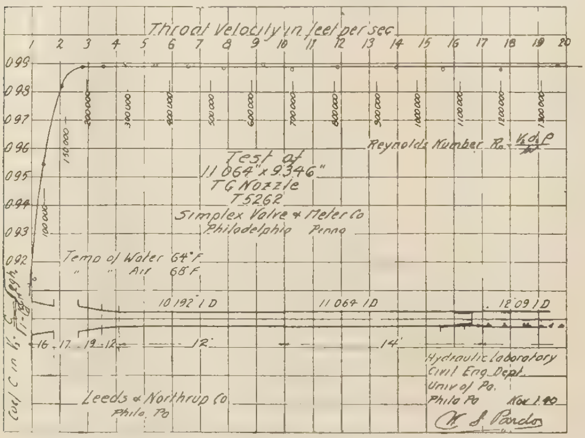

Fig. 11 Test of 11.064 × 9.346-In. TG Nozzle
(Simplex Valve & Meter Company, Philadelphia, Pa.)

W. S. Pardoe.[10] From an inspection of the graph giving the coefficient curves of high-ratio flow nozzles, with pipe taps, which appears in the closure to this paper, it is suggested that the authors might do well to consider the use of throat taps for high-ratio nozzles.

Fig. 11 of this discussion shows the test of an 11.064 × 9.346-in. flow nozzle with throat taps. The coefficient curve is quite normal and is flat from Reynolds' number 200,000 up.

Authors' Closure

Mr. Carter refers to Fig. 12 of this closure which shows the results of tests on large-diameter-ratio nozzles when corner taps were used. The purpose of this illustration is to emphasize the difficulty of correlating the results when there is so much scatter-

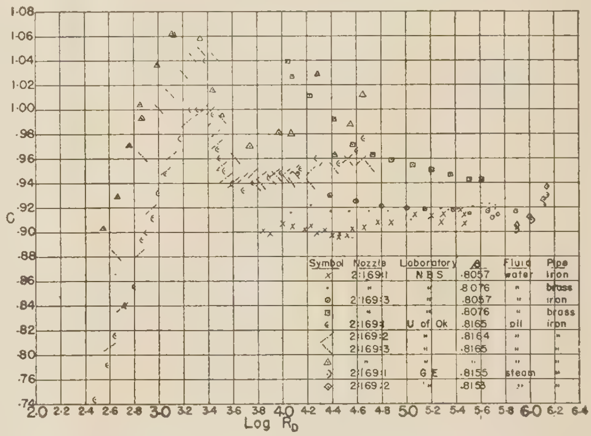

Fig. 12 Results of Tests With Corner Taps on Three Nozzles of Nearly the Same Throat Diameter

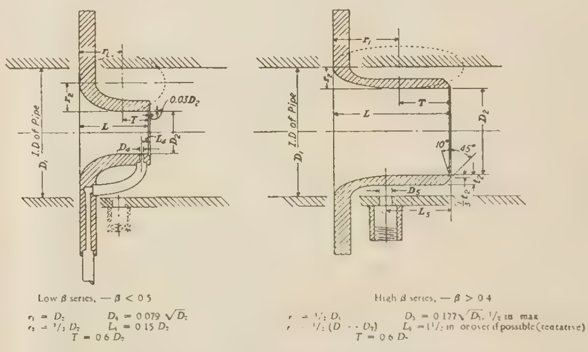

Fig. 13 Proportions of Long Radius Flow Nozzles Adopted by the Subcommittee on Flow-Nozzle Research for Use in Its Research Program
(The use of a pipe-wall tap instead of a nozzle-throat tap in the low β series is optional.)

ing. It is for the reason that there is more scattering with corner taps, as mentioned by Mr. Carter, that this particular plot is re-

[10] Professor of Hydraulic Engineering, University of Pennsylvania, Towne Scientific School, Philadelphia, Pa.

produced. It may be added that the location of the pressure taps, as between corner taps and pipe-wall taps, is more important than the nozzle-entrance forms, as represented by I.S.A. and long radius.

In reply to Dr. Johnson's questions, as shown by the curves in Figs. 2 to 7, inclusive, of the paper, the coefficients given cover diameter ratios all the way from 0.2 to over 0.8, so that they apply to both low-ratio and high-ratio nozzles, as these terms were originally defined by the committee. The meanings of the terms "low-ratio" and "high-ratio" are clearly defined and illustrated in Fig. 13[11] of this closure. It should be noted that the nozzle proportions given in Fig. 20 of Dr. Johnson's reference[8] do not correspond exactly to those given in Fig. 12 just referred to. The most important difference is in the length of the parallel portion of the throat. Except for a few nozzles loaned to the committee, the length of the throat section of all nozzles did not exceed the $0.6D_2$ shown in Fig. 13 of this closure. Of course, with the I.S.A. nozzles that were used, this throat section is even shorter.

It has been known for some time[12] that a long throat section will result in lowering the value of the discharge coefficients slightly. This will be particularly true as the throat diameter is decreased. Therefore, it cannot be expected that the coefficient values given in the present paper will apply exactly to small-diameter nozzles if made with as long a throat section as is called for by Dr. Johnson's reference.[8] It is unfortunate that such a discrepancy should exist between values given in papers by two committees of the Society. The authors do not know the source of Fig. 20 of that report.[8] However, as one of the authors is a member of the subcommittee which prepared the former report, he accepts his share of the responsibility for the difference between the two figures.

As to the wall thickness of the nozzle throat there was no general rule. For the larger-diameter-ratio nozzles, there was the requirement that there be some clearance (over $^1/_{16}$ in. diam) between the outside surface of the nozzle throat and the inner surface of the pipe. The purpose of this clearance is to provide fluid passage to the outlet-pressure tap when this is "under" the nozzle throat. This made it necessary to have the throat-wall thickness about $^1/_8$ in. for some of the larger nozzles for 4-in. and smaller pipe. For most of the nozzles the throat-wall thickness ranged from about $^3/_{16}$ in. for 2-in. pipe to $^5/_8$ in. for 8- and 16-in. pipe. So far as the authors know, the principal advantage of a thick wall is to diminish the possibility of the nozzle being damaged or deformed in handling.

The authors believe that the range of Reynolds' numbers covered in the present paper probably meets 75 per cent or more of the cases in commercial use. Moreover, there is much less scattering of the test data in the higher Reynolds' number range so that it was easier to obtain agreement over that region. Therefore, it seemed advisable to present this much of the results so users could have the benefit of it. As shown by Table 1 of the paper, some data were obtained for throat Reynolds' numbers of less than 100 and these results will be presented in a later paper. However, the spread of coefficients at low Reynolds' numbers is much greater than for the region here considered, so that they will be subject to a much larger tolerance than given for these results.

As to Professor Pardoe's suggestion, some eight to twelve nozzles provided with throat taps were used in the research program, although none was much if any over 0.5-diam ratio. Two of the nozzles were equipped with four pressure holes at 90 deg. With both of these, the results were more or less different with each pressure hole. While such a limited number of tests is insufficient as a basis for a final conclusion, it does strengthen the authors' belief that placing a pressure hole in a nozzle throat, where the fluid velocity past the hole is the highest, adds to the difficulties of obtaining reproduceable conditions. Moreover, except where the nozzle is of the solid-block type, the use of a throat tap adds considerably to the cost of construction.

Mr. King has shown a method which may be followed where it becomes necessary to extrapolate far beyond the range of the test data. The authors agree that, if practicable, for such cases, it is always better to use a method of plotting which results in straight lines.

However, it should be pointed out that there are but scant published data to support the assumption that the actual flow is:

$Q = Kh^n$, where n is a value different from the theoretical value of 0.5, which is the assumption upon which this straight line is drawn. It is also apparent that, if the curve is to be used for extrapolation, the head scale is being extrapolated 9 times its highest reading if the quantity and Reynolds' number curves are being extrapolated to 3 times their value.

These facts seem to indicate that Mr. King's suggested method requires more proof as to its accuracy before it can be accepted as the best method of extrapolation of these data.

Referring to the difference in coefficients between Mr. King's nozzles No. 59534 and No. 59535, the authors believe this illustrates the desirability, not only of keeping the ratio of throat diameter to pipe diameter at or below 80 per cent whenever possible, but also of boring the pipe so as to make the pipe surface concentric with the nozzle throat and as smooth and free from local irregularities as possible. Naturally, the effect of such surface roughness and eccentricity is greatly minimized with nozzles of small diameter ratio, but quite likely to be magnified as the diameter ratio increases.

In comparing the shape of the coefficient curves of these two nozzles in Fig. 10 of Mr. King's discussion, it must be remembered that twice as much data were taken on the bored-pipe nozzle No. 59535 as with the unbored-pipe nozzle No. 59534; and that, had a slight amount of data been taken with nozzle No. 59534, the shape of its coefficient curve would likely have been much the same as the curve for the bored-pipe nozzle.

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[11] Reproduced from paper, "Research on Flow Nozzles," by H. S. Bean, *Mechanical Engineering*, vol. 59, 1937, Fig. 1, p. 501.

[12] "Measurement of Flow of Air and Gas With Nozzles," by S. A. Moss, Trans. A.S.M.E., vol. 50, 1928, paper APM-3, pp. 1–10; discussion by H. S. Bean, pp. 13–15.

Remarks on the Analogy Between Heat Transfer and Momentum Transfer

(B)

By L. M. K. BOELTER,[1] R. C. MARTINELLI,[2] and FINN JONASSEN[2]

This paper presents an extension of Von Kármán's analysis of heat transfer to fluids in closed conduits, based on the analogy between heat transfer and momentum transfer. For a particular ideal system, an expression for the temperature distribution in a fluid in turbulent motion being heated or cooled inside of a circular pipe is derived and a relation is obtained between Nusselt's modulus and the pipe-friction factor for "isothermal" heat transfer. This equation is extended to apply in cases in which the physical properties of the fluid vary across the section of the pipe. The apparent variation of the dimensionless distance parameter y^+ with ratio of wall viscosity to laminar sublayer viscosity and Reynolds' number is obtained. A comparison between the equation developed in the paper and the Nusselt empirical equation, including the constants evaluated by Dittus and Boelter, is made.

Nomenclature

The following nomenclature is used in the paper:

a = constant in empirical equation for viscosity
A = area perpendicular to heat flow, ft²
A_0 = inside surface area of pipe, ft²
b = exponent in empirical equation for viscosity
c_p = unit heat capacity of fluid, Btu/lb deg F
C = constant
D = diameter of pipe, ft
f = friction factor for flow in pipe, dimensionless
f_c = unit thermal conductance between pipe and fluid, Btu per (hr) (sq ft) (deg F)
g = gravitational constant, ft/hr sec
k = thermal conductivity of fluid, Btu per (hr) (ft²) (deg F/ft)
l = Prandtl mixing length, ft
ln = natural logarithm
m = exponent of Re in empirical heat-transfer Equation [2]; also subscript indicating mean temperature
n = exponent of Pr in Equation [2]; also subscript denoting location of thermal resistance
q = radial rate of heat transfer, Btu per hr
q_0 = radial rate of heat transfer through pipe wall, Btu/hr
r = distance from center of pipe to any point, ft
r_0 = radius of pipe, ft
t = temperature at any point y, deg F
t' = fluctuating component of temperature, deg F
t_1 = temperature at outer edge of laminar sublayer, deg F
t_w = temperature of pipe wall, deg F
u = average axial velocity of flow at any point y, fps
u_1 = velocity at edge of laminar sublayer, fps
u' = fluctuating component of axial velocity, fps
u_{max} = average axial velocity at center of pipe, fps
u_{mean} = mean velocity of flow based on rate of discharge and pipe area, fps
u^+ = dimensionless "velocity" parameter $= \dfrac{u}{\sqrt{\dfrac{\tau_0}{\rho}}}$
v' = fluctuating component of velocity perpendicular to axis of pipe, fps
y = distance from wall, ft
y_1 = distance from wall to outer edge of laminar sublayer, ft
y_2 = distance from wall to outer edge of buffer layer, ft
y^+ = dimensionless "distance" parameter $= \sqrt{\dfrac{\tau_0}{\rho}}\, y/\nu$
y_1^+ = dimensionless parameter fixing dimensions of laminar sublayer $= \sqrt{\dfrac{\tau_0}{\rho}}\, y_1/\nu$
y_2^+ = dimensionless parameter fixing dimensions of buffer layer $= \sqrt{\dfrac{\tau_0}{\rho}}\, y_2/\nu$
Δt = difference between temperature of pipe wall and any point y, deg F
Δt_{max} = difference between temperature of pipe wall and center of pipe, deg F
Δt_{mean} = difference between temperature of pipe wall and average (mixed) temperature of fluid, deg F
ϵ = eddy diffusivity, ft²/sec
γ = weight density of fluid, lb/cu ft
κ = Kármán constant = 0.4
μ = viscosity of fluid, lb-sec/ft²
μ_1 = viscosity of fluid at temperature t_1, lb-sec/ft²
μ_f = mean fluid viscosity in laminar sublayer, lb-sec/ft²
μ_w = viscosity of the fluid at the temperature t_w, lb-sec/ft²
ν = kinematic viscosity of fluid, ft²/sec
ν_1 = kinematic viscosity of fluid at temperature t_1, ft²/sec
ξ = function which fixes velocity at edge of laminar sublayer
ρ = density of fluid, (lb sec²)/ft⁴
τ = unit shear at any point y, lb/ft²
τ_0 = unit shear at wall, lb/ft²
ϕ = function
Nu = Nusselt's modulus $= \dfrac{f_c D}{k}$
Nu_1 = Nusselt's modulus of laminar sublayer
Nu_2 = Nusselt's modulus of buffer layer
Nu_3 = Nusselt's modulus of turbulent core
Nu_n = Nusselt's modulus of any of three fluid layers
Pr = Prandtl's modulus $= \mu c_p g/k$
Pr_1 = Prandtl's modulus at temperature t_1
Pr_m = Prandtl's modulus at mean (mixed) fluid temperature
Re = Reynolds' modulus $= \dfrac{u_{mean} D\rho}{\mu}$
Re_m = Reynolds' modulus at mean (mixed) fluid temperature

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Introduction

Application of the theory of similarity to the differential equations of heat transfer and fluid flow yields the result that for heat transfer from a fluid to a solid boundary in similar systems a relation

$$Nu = \phi(Re, Pr) \qquad [1]$$

will exist. The function ϕ may be formulated from experimental data and, for purposes of design, an empirical expression of the form

$$Nu = CRe^{m}Pr^{n} \qquad [2]$$

is normally employed.

Reynolds (1),[3] and later Prandtl (2), G. I. Taylor (3), and von Kármán (4) have attempted to deduce the form of the function ϕ analytically for turbulent flow by comparing the exchange of momentum and of heat. Reynolds dealt with a purely turbulent system in which exact similarity between temperature and velocity fields was postulated (i.e., $Pr = 1$). Prandtl and G. I. Taylor extended the analogy to other values of the Prandtl modulus Pr by introducing a concept which included a laminar sublayer and a turbulent core. Von Kármán defined a buffer layer (of particular characteristics) which was located between the laminar layer and the turbulent core.

An attempt is made by the authors of this paper to provide an extension of the ideal system of von Kármán. In this ideal system, the resistances to heat transfer from a solid boundary to a fluid consist of the following:

1 A laminar sublayer in which viscous forces predominate. The heat is transferred through this sublayer by conduction only.

2 A "buffer" layer in which both viscous and eddy forces are important and through which heat is transferred by both thermal conduction and eddy diffusion.

3 A core of fluid in which eddy forces predominate. The heat is transferred in the core by eddy motion only.

The rate of heat transfer per unit area and the unit shear at any point in the system may be expressed as follows:

	Unit shear[4]	Rate of heat transfer per unit area
Laminar sublayer	$\frac{\tau}{\rho} = \nu \frac{du}{dy}$	$\frac{q}{A} = -k\frac{dt}{dy}$
Buffer layer	$\frac{\tau}{\rho} = (\nu + \epsilon)\frac{du}{dy}$	$\frac{q}{Ac_p\gamma} = -\left(\frac{\nu}{Pr} + \epsilon\right)\frac{dt}{dy}$
Turbulent core	$\frac{\tau}{\rho} = \epsilon\frac{du}{dy}$	$\frac{q}{Ac_p\gamma} = -\epsilon\frac{dt}{dy}$

[3] Numbers in parentheses refer to the Bibliography at end of paper.
[4] See Note 1 of Appendix.

For $Pr = 1$ the temperature and the velocity fields are similar.

The evaluation of the unit shear and the rate of heat transfer require a knowledge of the velocity gradient $\left(\frac{du}{dy}\right)$ at every point in the system.

"Isothermal" Heat Transfer in a Circular Pipe

For circular pipes, the experimentally determined velocity distribution may be correlated (7) by plotting the parameter

$$u^+ = \frac{u}{\sqrt{\frac{\tau_0}{\rho}}} \qquad \text{against} \qquad y^+ = \frac{\sqrt{\frac{\tau_0}{\rho}}\, y}{\nu}$$

One curve is obtained for all magnitudes of Reynolds' modulus, Re. This plot is shown in Fig. 1.

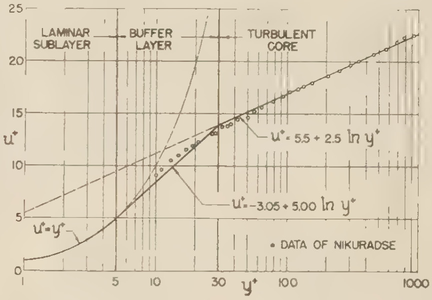

Fig. 1 Velocity Distribution Across a Pipe, Represented by Means of Generalized Coordinates

The following equations express the velocity distribution in the three fluid layers:

Laminar sublayer	$(0 \leq y^+ \leq 5)$	$u^+ = y^+$
Buffer layer	$(5 \leq y^+ \leq 30)$	$u^+ = -3.05 + 5.00\, ln\, y^+$
Turbulent core	$(y^+ \geq 30)$	$u^+ = 5.5 + 2.5\, ln\, y^+$

These equations allow the determination of $\left(\frac{du}{dy}\right)$ at any point in the flow system. The thermal resistance of each of the fluid layers may be calculated from the equations relating u^+ and y^+.

The necessary operations are given in Table 1. The following specifications of the ideal system are to be added to those listed:

1 All fluid physical properties are independent of temperature; i.e., "isothermal" heat transfer is postulated.

TABLE 1 DEFINITION OF IDEAL SYSTEM—ISOTHERMAL HEAT TRANSFER IN A CIRCULAR PIPE

	EXPERIMENTAL VELOCITY DISTRIBUTION	SHEAR	SHEAR EQUATION	EDDY DIFFUSIVITY	RATE OF HEAT FLOW	$\frac{q}{A}$	TEMPERATURE DROP	INTEGRATED FORM OF Δt	THERMAL RESISTANCES IN DIMENSIONLESS FORM
LAMINAR SUBLAYER	$u^+ = y^+$, $0 \leq y^+ \leq 5$	$\tau = \tau_0$	$\frac{\tau_0}{\rho} = \nu\frac{du}{dy}$	$\epsilon = 0$	$\frac{q}{A} = -k\frac{dt}{dy}$	$\frac{q_0}{A_0}$	$\Delta t = \frac{q_0}{A_0}\int_0^{y_1}\frac{dy}{k}$	$\Delta t = \frac{q_0}{A_0}\frac{y_1}{k}$	$\frac{1}{NU_1} = \frac{y_1^+}{RE}\sqrt{\frac{8}{f}}$
BUFFER LAYER	$u^+ = -3.05 + 5.00 \ln y^+$, $5 \leq y^+ \leq 30$	$\tau = \tau_0$	$\frac{\tau_0}{\rho} = (\nu + \epsilon)\frac{du}{dy}$	$\epsilon = \left(\frac{\frac{\tau_0}{\rho}}{\frac{du}{dy}} - \nu\right)$	$\frac{q}{Ac_p\gamma} = -\left(\frac{\nu}{PR} + \epsilon\right)\frac{dt}{dy}$	$\frac{q_0}{A_0}$	$\Delta t = \frac{q_0}{Ac_p\gamma}\int_{y_1}^{y_2}\frac{dy}{\left(\frac{\nu}{PR} + \epsilon\right)}$	$\Delta t = \frac{q_0}{Ac_p\gamma}\frac{y_1^+}{\sqrt{\frac{\tau_0}{\rho}}}\ln\left(1 + PR\left\{\frac{y_2^+}{y_1^+} - 1\right\}\right)$	$\frac{1}{NU_2} = \frac{y_1^+\sqrt{\frac{8}{f}}\ln\left(1 + PR\left\{\frac{y_2^+}{y_1^+} - 1\right\}\right)}{RE\ PR}$
TURBULENT CORE	$u^+ = 5.5 + 2.5 \ln y^+$, $y^+ \geq 30$	$\tau = \tau_0\left(1 - \frac{y}{r_0}\right)$	$\frac{\tau}{\rho} = \frac{\tau_0}{\rho}\left(1 - \frac{y}{r_0}\right) = \epsilon\frac{du}{dy}$	$\epsilon = l^2\frac{du}{dy}$, $l = \kappa y\sqrt{1 - y/r_0}$	$\frac{q}{Ac_p\gamma} = -\epsilon\frac{dt}{dy}$	$\frac{q_0}{A_0}\left(1 - \frac{y}{r_0}\right)$	$\Delta t = \frac{q_0}{Ac_p\gamma}\int_{y_2}^{r_0}\frac{dy}{\epsilon}$	$\Delta t = \frac{q_0}{2.5Ac_p\gamma\sqrt{\frac{\tau_0}{\rho}}\kappa}\ln\frac{r_0}{y_2}$	$\frac{1}{NU_3} = \frac{2.5}{RE\,PR}\sqrt{\frac{8}{f}}\ln\frac{RE\sqrt{\frac{f}{8}}}{2y_2^+}$

A = AREA PERPENDICULAR TO HEAT FLOW AT ANY y
f = FRICTION FACTOR
k = THERMAL CONDUCTIVITY OF THE FLUID
l = PRANDTL MIXING LENGTH
q = RATE OF HEAT TRANSFER (RADIAL) AT ANY y
r_0 = RADIUS OF PIPE.
t = TEMPERATURE OF FLUID AT ANY POINT y
u = VELOCITY AT ANY POINT y
y = DISTANCE FROM THE WALL
y_1 = DISTANCE FROM WALL TO EDGE OF LAMINAR SUBLAYER
y_2 = DISTANCE FROM WALL TO EDGE OF BUFFER LAYER
c_p = UNIT HEAT CAPACITY OF FLUID
q_0 = RATE OF HEAT TRANSFER (RADIAL) AT $y = 0$
$A_0 = 2\pi r_0 L$ = SURFACE AREA OF PIPE
L = LENGTH OF PIPE

ϵ = EDDY DIFFUSIVITY
κ = KÁRMÁN CONSTANT = 0.40
ν = KINEMATIC VISCOSITY OF FLUID
ρ = DENSITY OF FLUID
τ = SHEAR AT ANY POINT y
τ_0 = SHEAR AT WALL
γ = UNIT WEIGHT OF FLUID

PR = PRANDTL'S MODULUS

$u^+ = \frac{u}{\sqrt{\frac{\tau_0}{\rho}}}$

$y^+ = \frac{\sqrt{\frac{\tau_0}{\rho}}\,y}{\nu}$

$y_1^+ = \frac{\sqrt{\frac{\tau_0}{\rho}}\,y_1}{\nu}$

$y_2^+ = \frac{\sqrt{\frac{\tau_0}{\rho}}\,y_2}{\nu}$

NU = NUSSELT'S MODULUS

2 The shear is constant in both the laminar sublayer and buffer layer, Fig. 2.

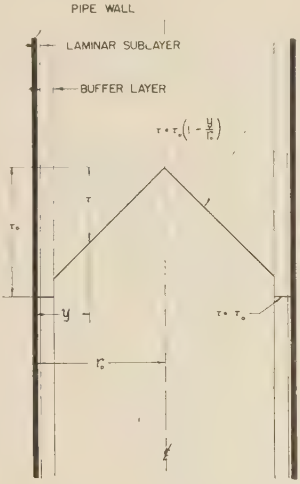

FIG. 2 UNIT-SHEAR DISTRIBUTION ACROSS PIPE
(As employed in ideal system presented in this paper.)

3 The rate of radial heat flow per unit area is constant except in the turbulent region where it varies linearly to zero at the center. The postulate may be shown to be very nearly the case by performing a heat balance on a differential annulus of fluid.

4 The mixing length l, in the turbulent region is defined (8) by the equation

$$l = \kappa y \sqrt{1 - y/r_0}$$

Experimental evidence reveals that this equation does not represent the facts. However, Prandtl (9), in effect,[5] by making the same idealization, adequately represented the velocity distribution in a pipe.

5 The velocity gradient $\left(\frac{du}{dy}\right)$ is not zero at the center of the pipe. This is a well-known weakness of all present-day logarithmic expressions for velocity distribution.

From Table 1 the total resistance to heat transfer may be obtained

$$\frac{1}{Nu} = \sum_1^3 \frac{1}{Nu_n}$$

Using values of $y_1^+ = 5$ and $y_2^+ = 30$, which are reasonable from an inspection of the data in Fig. 1, the final result is

$$\frac{Nu}{Re\,Pr} = \frac{\sqrt{\frac{f}{8}}}{5\left[Pr + ln\,(1 + 5Pr) + 0.5\,ln\,\frac{Re}{60}\sqrt{\frac{f}{8}}\right]} \quad \ldots [3]$$

This expression for $\frac{Nu}{Re\,Pr}$ is based on the temperature difference between the pipe wall and the center of the pipe. Experimental results are always based on an average fluid temperature. Thus, to compare Equation [3] with experimental values, the right side should be divided by the ratio

$$\frac{\Delta t_{\text{mean}}}{\Delta t_{\text{max}}} \quad \ldots\ldots\ldots\ldots [4]$$

The temperature-difference ratio expressed in Equation [4] may be determined analytically from Equation [3].

The thermal resistance from the wall to any point y, located in the turbulent core, is given by

[5] See Note 2 of Appendix.

$$\frac{5}{Re\,Pr}\sqrt{\frac{8}{f}}\left(Pr + ln\,(1 + 5\,Pr) + 0.5\,ln\,\frac{Re}{60}\frac{y}{r_0}\sqrt{\frac{f}{8}}\right) \ldots [5]$$

The resistance to the center of the pipe is

$$\frac{5}{Re\,Pr}\sqrt{\frac{8}{f}}\left(Pr + ln\,(1 + 5\,Pr) + 0.5\,ln\,\frac{Re}{60}\sqrt{\frac{f}{8}}\right) \ldots [6]$$

Thus the temperature distribution in the turbulent core is given by

$$\frac{\Delta t}{\Delta t_{\text{max}}} = \left(\frac{Pr + ln\,(1 + 5\,Pr) + 0.5\,ln\,\frac{Re}{60}\frac{y}{r_0}\sqrt{\frac{f}{8}}}{Pr + ln\,(1 + 5\,Pr) + 0.5\,ln\,\frac{Re}{60}\sqrt{\frac{f}{8}}}\right) \ldots [7]$$

A plot of the temperature distribution for various magnitudes of the Prandtl modulus Pr at Reynolds' modulus Re equal to 10,000 is shown in Fig. 3.

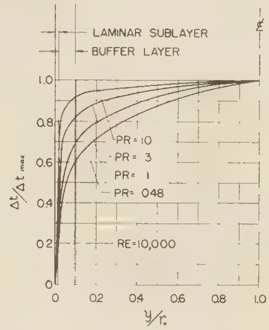

FIG. 3 PREDICTED TEMPERATURE DISTRIBUTION ACROSS PIPE SECTION FOR VARIOUS MAGNITUDES OF PRANDTL MODULUS AT REYNOLDS' NUMBER = 10,000
(Prediction is based upon Equation [7].)

The velocity distribution (10) is given by the equation

$$\frac{u}{u_{\text{max}}} = \left(\frac{5.5 + 2.5\,ln\,\frac{Re}{2}\frac{y}{r_0}\sqrt{\frac{f}{8}}}{5.5 + 2.5\,ln\,\frac{Re}{2}\sqrt{\frac{f}{8}}}\right) \ldots\ldots\ldots [8]$$

At a value of $Pr = 1$ the temperature distribution should be identical with the velocity distribution. Setting $Pr = 1$ in Equation [7]

$$\frac{\Delta t}{\Delta t_{\text{max}}} = \left(\frac{5.45 + 2.5\,ln\,\frac{Re}{2}\frac{y}{r_0}\sqrt{\frac{f}{8}}}{5.45 + 2.5\,ln\,\frac{Re}{2}\sqrt{\frac{f}{8}}}\right) \ldots\ldots\ldots [9]$$

which agrees favorably with Equation [8].

The mean temperature may be obtained graphically from the velocity and temperature distributions. Thus for an incompressible fluid

$$\frac{\Delta t_{\text{mean}}}{\Delta t_{\text{max}}} = \frac{\int_0^{r_0} \frac{u}{u_{\text{max}}} \times \frac{\Delta t}{\Delta t_{\text{max}}}\,rdr}{\int_0^{r_0} \frac{u}{u_{\text{max}}} \times rdr} \ldots\ldots\ldots [10]$$

Curves of this expression for several magnitudes of Re are represented in Fig. 4. Inclusion of the effect of compressibility within the range of magnitudes employed by the experimenters quoted in this paper yielded results which did not differ appreciably from

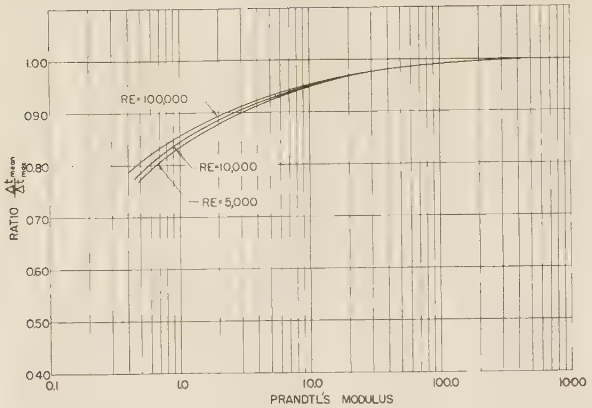

FIG. 4 RATIO OF MEAN TO MAXIMUM TEMPERATURE DIFFERENCES AS A FUNCTION OF PRANDTL MODULUS FOR REYNOLDS' NUMBERS = 5000, 10,000, AND 100,000

(Refer to Equation [10].)

those of Equation [10]. Thus, for comparison with experimental data, Equation [3] becomes

$$\frac{Nu}{Re\,Pr} = \frac{\sqrt{\frac{f}{8}}\,\frac{\Delta t_{\max}}{\Delta t_{\text{mean}}}}{5\left[Pr + ln\,(1 + 5\,Pr) + 0.5\,ln\,\frac{Re}{60}\sqrt{\frac{f}{8}}\right]} \quad [11]$$

Plots of this expression for various magnitudes of Re are shown

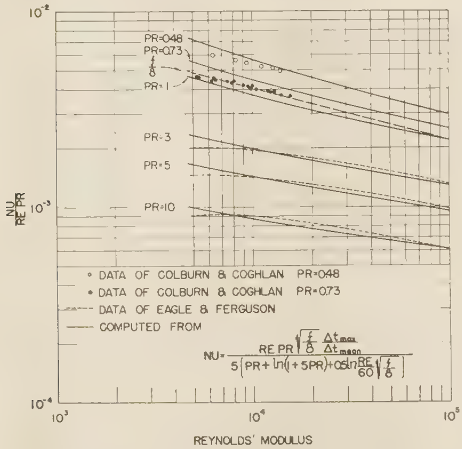

FIG. 5 EXPERIMENTAL AND PREDICTED MAGNITUDES OF $Nu/(Re\,Pr)$ AS FUNCTION OF REYNOLDS' MODULUS FOR DIFFERENT MAGNITUDES OF PRANDTL MODULUS

(Predicted magnitudes apply to ideal isothermal heat-transfer system.)

in Fig. 5, as well as the data of Eagle and Ferguson (11) and Colburn and Coghlan (12). The correlation is seen to be good except at low values of Re. This discrepancy may be due in part to the fact that, since thermal calming sections were not used, the temperature distribution considered in this analysis was not attained until the fluid traversed a considerable distance in the test section. This distance would be shorter the higher the magnitude of the unit thermal conductance. Thus better correlation at high Reynolds' numbers may be expected.

The lower limit of $\frac{Nu}{Re\,Pr}$ is that given by Equation [3], in which case the temperature is, in effect, considered constant across the pipe section. Calculation reveals that the experimental points of Colburn and Coghlan fall between this lower limit and that given by Equation [11].

Equation [11] indicates that $\frac{Nu}{Re\,Pr}$ is directly proportional to $\sqrt{\frac{f}{8}}$ with another term involving the friction factor in the denominator.

For $Pr = 1$, Equation [11] becomes

$$\frac{Nu}{Re\,Pr} = \frac{\sqrt{\frac{f}{8}}\,\frac{\Delta t_{\max}}{\Delta t_{\text{mean}}}}{5\left[1 + ln\,6 + 0.5\,ln\,\frac{Re}{60}\sqrt{\frac{f}{8}}\right]} \quad [12]$$

As shown in Table 2, the coefficient of $\sqrt{\frac{f}{8}}$ is almost numerically equal to $\sqrt{\frac{f}{8}}$ so that the following relation results

$$\frac{Nu}{Re\,Pr} \cong \frac{f}{8} \quad [13]$$

This result was obtained by Reynolds (1) for the ideal system in which the heat transfer was accomplished by eddy diffusion only.

TABLE 2 COMPARISON OF MAGNITUDES INVOLVED IN EQUATION [12]

Re	$\sqrt{\frac{8}{f}}$	$\frac{5\left(1 + ln\,6 + 0.5\,ln\,\frac{Re}{60}\sqrt{\frac{f}{8}}\right)}{\Delta t_{\max}/\Delta t_{\text{mean}}}$
5000	14.3	15.0
10000	15.8	16.6
100000	21.4	22.0

NONISOTHERMAL HEAT TRANSFER IN A CIRCULAR PIPE

The previous analysis is applicable only to "isothermal" heat transfer, since variations of viscosity and other fluid properties with temperature were not considered. The data of Colburn and Coghlan and Eagle and Ferguson are comparable to these conditions for, in the former, variations of viscosity were small and, in the latter, all results were extrapolated to conditions of zero heat transfer. On the other hand, the analysis of heat transfer in fluids which possess a highly variable viscosity requires the definition of another ideal system. In particular, the boundary of the laminar sublayer for nonisothermal flow requires investigation. The dimensionless variable, y_1^+, which fixes the outer boundary of the laminar sublayer will be redefined as a point function (exclusive of the unit-shear term which is defined as constant). Thus

$$y_1^+ = \frac{\sqrt{\frac{\tau_0}{\rho}}\,y_1}{\nu_1} \quad [14]$$

where ν_1 is the kinematic viscosity of the fluid at the edge of the laminar sublayer and τ_0 is the shear at the wall under nonisothermal conditions. Evaluation of the thermal resistance of the laminar sublayer yields the expression

$$\frac{1}{Nu_1} = \frac{y_1^+\sqrt{\frac{8}{f}}\,Pr_1}{Re_m Pr_m} \quad [15]$$

Without introducing too great an error, the remainder of the fluid system may be defined as possessing a constant viscosity which is fixed by the mean fluid temperature. The expression for $\frac{Nu}{Re\,Pr}$ then becomes

$$\frac{Nu}{Re_m Pr_m} = \frac{\sqrt{\frac{f}{8}}\,\frac{\Delta t_{\max}}{\Delta t_{\text{mean}}}}{y_1^+\left[Pr_1 + ln\left(1 + Pr_m\left\{\frac{30}{y_1^+} - 1\right\}\right) + \frac{2.5}{y_1^+}\, ln \frac{Re_m}{60}\sqrt{\frac{f}{8}}\right]} \quad [16]$$

Examination reveals that Equation [11] is a special case of Equation [16]. A large variation of viscosity across the laminar sublayer should cause variations in y_1^+, which fixes its boundary. To determine this variation, the data of Morris and Whitman (13) were substituted into Equation [16]. Excellent correlation resulted, y_1^+ being a function of Reynolds' modulus, and the ratio $\frac{\mu_w}{\mu_1}$ where μ_w = the viscosity of the fluid at the wall temperature and μ_1 = the viscosity of the fluid at the temperature existing at the edge of the laminar sublayer.[6]

The results of the computations are shown in Fig. 6, and are tabulated in Table 3. Using the curves of Fig. 6 and the experimental conditions of Colburn and Coghlan, Eagle and Ferguson, Morris and Whitman, and J. F. D. Smith (14), magnitudes of Nu were computed. The magnitude of the predicted Nusselt modulus employing Equation [16] was plotted against the experimental magnitudes in Fig. 7.

TABLE 3 MAGNITUDES OF y_1^+

$\frac{\mu_w}{\mu_1}$	Re_m 3000	Re_m 5000	Re_m 10000	Re_m 20000	Re_m 40000	Re_m 100000
0.10	..	..	..	..	..	..
0.20	2.30	..	..	..	..	..
0.30	3.12	2.34	..	..	..	..
0.40	3.70	3.07	2.40	..	..	..
0.50	4.10	3.60	3.09	2.60	2.00	..
0.60	4.43	4.03	3.62	3.34	2.90	2.50
0.70	4.68	4.40	4.10	3.90	3.62	3.40
0.80	4.82	4.70	4.50	4.40	4.25	4.10
0.90	4.95	4.90	4.80	4.75	4.70	4.65
1.00	5.00	5.00	5.00	5.00	5.00	5.00
1.50	4.95	4.80	4.60	4.35	4.00	3.75
2.00	4.89	4.52	4.20	3.75	3.00	2.50
3.00	4.75	4.18	3.60	2.70	..	..
4.00	4.65	3.90	3.15	..	..	..
5.00	4.57	3.70	..	..	..	..
6.00	4.50	..	..	..	..	..
7.00	..	..	..	..	..	..

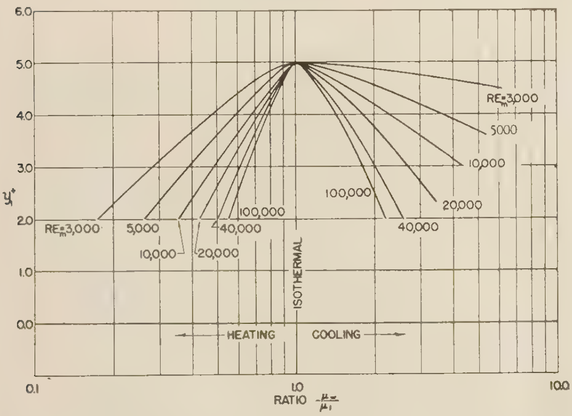

FIG. 6 MAGNITUDE OF y_1^+ AT INTERFACE BETWEEN LAMINAR SUBLAYER AND BUFFER LAYER AS A FUNCTION OF RATIO OF WALL TO INTERFACE VISCOSITIES, REYNOLDS' NUMBER AS PARAMETER

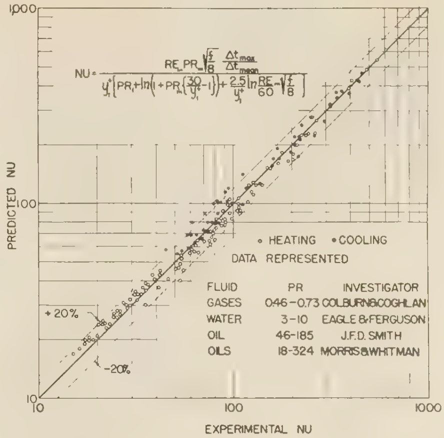

FIG. 7 PREDICTED MAGNITUDE OF NUSSELT MODULUS PLOTTED AS FUNCTION OF EXPERIMENTAL MAGNITUDES OF NUSSELT MODULUS, FOR HEATING AND COOLING, AND FOR RANGE OF PRANDTL MODULUS FROM 0.46 TO 324
(Refer to Equation [16].)

The friction factors used in the foregoing correlation were obtained from the isothermal friction-factor curve[7] at a Reynolds number computed using the mean laminar sublayer viscosity μ_f instead of the viscosity at the mean fluid temperature μ_m. This viscosity (17) may be found from the relation

$$\mu_f = \frac{1}{y_1}\int_0^{y_1} \mu dy \ldots\ldots\ldots\ldots\ldots\ldots [17]$$

If $\mu = \frac{a}{t^b}$ (an empirical expression of wide application)—then

$$\mu_f = \mu_1 \left\{ \frac{\left(\frac{\mu_1}{\mu_w}\right)^{\frac{1-b}{b}} - 1}{(1 - b)\left[\left(\frac{\mu_1}{\mu_w}\right)^{\frac{1}{b}} - 1\right]} \right\} \ldots\ldots\ldots [18]$$

A plot of this expression is shown in Fig. 8.

A comparison of the predicted value of $\sqrt{\frac{f}{8}}$ as compared with the measured values of J. F. D. Smith is shown in Fig. 9. The check is usually within 6 per cent which is sufficiently accurate for

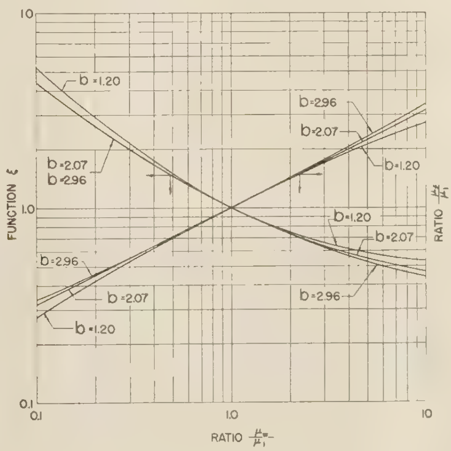

FIG. 8 FUNCTION ξ PLOTTED IN TERMS OF RATIO OF WALL TO INTERFACE VISCOSITIES FOR VARIOUS MAGNITUDES OF VISCOSITY EXPONENT b; RATIO OF MEAN LAMINAR SUBLAYER VISCOSITY TO INTERFACE VISCOSITY AS A FUNCTION OF RATIO OF WALL TO INTERFACE VISCOSITIES

[6] See Note 3 of Appendix.
[7] See Note 4 of Appendix.

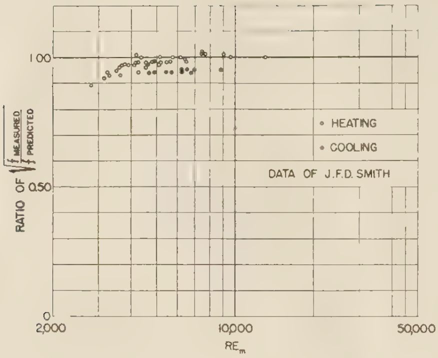

Fig. 9 Ratio of the Square Root of the Measured to the Predicted Friction Factor for Nonisothermal Flow as a Function of Reynolds' Modulus

the correlation shown in Fig. 7. Rohonczi (16) suggests a method of presenting the nonisothermal friction factor in terms of Reynolds' modulus evaluated at the wall temperature. This method results in over-correcting the friction factor.

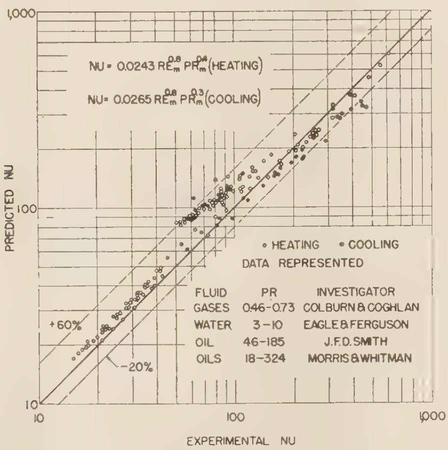

Fig. 10 Predicted Versus Experimental Magnitudes of Nusselt Modulus
(Predictions in terms of empirical Equations [19] and [20].)

A practical limit to the correlation with the available experimental data exists. This limitation is illustrated as follows:

Investigators	Run	Re_m	Pr_m	$\frac{\mu_w}{\mu_1}$	Nu
J. F. D. Smith	13 (cooling)	5410	103	4.0	71.1
Morris and Whitman	F 16	5460	99	3.4	98.0

Thus for practically identical experimental conditions the two investigators reported a difference in Nusselt's modulus of 40 per cent. Some of this difference is explained by the fact that Morris and Whitman used commercial pipe, while J. F. D. Smith used smooth pipe. Even after making allowance for this fact the values of Nu differ by about 30 per cent.

In order to compare the results from Equation [16] with the empirical equations (15).

$$Nu = 0.0243 Re_m^{0.8} Pr_m^{0.4} \text{ (heating)} \qquad [19]$$

$$Nu = 0.0265 Re_m^{0.8} Pr_m^{0.3} \text{ (cooling)} \qquad [20]$$

the same experimental values used in Fig. 7 are plotted against the values computed by Equations [19] and [20], and are shown in Fig. 10. In Equations [19] and [20] the fluid properties are evaluated at the arithmetic mean fluid temperature.

Inspection of Equation [16] indicates that Nu should be practically independent of Pr for large magnitudes of the Prandtl modulus. However, variations of y_1^+ accompany the variations in viscosity which always occur in the experimental determination of Nu. These variations in y_1^+ give the apparent influence of Pr expressed by Equations [19] and [20].

The velocity existing at the edge of the laminar sublayer for nonisothermal conditions may be readily estimated. Since

$$\frac{\tau_0}{\rho} = \nu \frac{du}{dy}$$

and

$$u_1 = \int_0^{y_1} \frac{\tau_0 \, dy}{\mu}$$

which upon integration with

$$\mu = \frac{a}{t^b} \qquad [21]$$

becomes

$$\frac{u_1}{\sqrt{\frac{\tau_0}{\rho}}} = y_1^+ \left\{ \frac{1 - \left(\frac{\mu_1}{\mu_w}\right)^{\frac{b+1}{b}}}{(1+b)\left[1 - \left(\frac{\mu_1}{\mu_w}\right)^{\frac{1}{b}}\right]} \right\} \qquad [22]$$

or

$$\frac{u_1}{\sqrt{\frac{\tau_0}{\rho}}} = y_1^+ \xi$$

A plot of the function ξ is given in Fig. 8.

Conclusions

1 The thermal resistances involved in heat transfer from a boundary to a turbulently moving fluid have been segregated. An expression which correlates existing heat-transfer data for flow through tubes is proposed for the system in which the fluid properties are not functions of temperature.

2 The introduction of the concept of a variable y_1^+, the utilization of an average viscosity rather than the viscosity at an average temperature, and the estimation of a nonisothermal friction factor make possible the prediction of unit thermal conductances to within ±20 per cent for Reynolds' numbers up to 100,000 and magnitudes of the Prandtl modulus between 0.5 and 325.

3 An ideal system may be defined which will more accurately predict $Nu/(Re\ Pr)$ but the algebraic evaluation will be tedious. Among other operations, the variation of Nu with length (due to the changes in properties and of temperature distribution and the variation of q/A with radius) should be considered. Additional accurate experimental heat-transfer data will aid the analyst.

4 Finally, a more precise formulation of the laws governing the velocity distribution (18) will allow the definition of a more satisfactory ideal system for heat transfer.

Appendix

Note 1

Reynolds (5) conceived the flow at a point in a two-dimensional system to be composed of an average velocity u upon which are superimposed fluctuating components u', v'. The unit shear due to the momentum transferred by these fluctuating components becomes

$$\tau = -\rho \overline{u'v'}$$

where the expression $\overline{u'v'}$ represents the time average of the instantaneous fluctuations u' and v'. Prandtl (6) assumed that the fluctuating velocities would be equal to the product of a "mixing" length l and the velocity gradient at the point. Thus the magnitude of the shear becomes

$$|\tau| = \rho l^2 \left(\frac{du}{dy}\right)^2$$

By analogy to the expression for shear in viscous flow

$$\tau = \rho\nu \frac{du}{dy}$$

an eddy "kinematic viscosity" ϵ may be defined as

$$\epsilon = l^2 \frac{du}{dy}$$

and

$$\tau = \rho\epsilon \frac{du}{dy}$$

Similar reasoning in the case of the transfer of heat yields the rate of heat transfer per unit area in turbulent flow as

$$\frac{q}{A} = c\gamma\rho v't'$$

where v' and t' are the fluctuating components of velocity and temperature, respectively. Then if

$$t' = l\frac{dt}{dy}$$

$$v' = l\frac{du}{dy}$$

the rate of heat transfer per unit area in turbulent flow becomes

$$q/A = c_p\gamma\epsilon \frac{dt}{dy}$$

The identity of the eddy diffusivity ϵ for heat transfer and momentum transfer is known as Reynolds' analogy.

Note 2

Prandtl actually postulated a constant shear across the pipe and a mixing length $= \kappa y$. Thus

$$\frac{\tau}{\rho} = \frac{\tau_0}{\rho} = \epsilon\frac{du}{dy}$$

and

$$\epsilon = l^2 \frac{du}{dy}$$

$$\frac{\tau_0}{\rho} = \kappa^2 y^2 \left(\frac{du}{dy}\right)^2$$

from which

$$u = \frac{\sqrt{\tau_0/\rho}}{\kappa}\, ln\, y + C$$

Exactly the same result is obtained if

$$\tau = \tau_0\left(1 - \frac{y}{r_0}\right)$$

$$l = \kappa y\sqrt{1 - y/r_0}$$

for then

$$\frac{\tau}{\rho} = \frac{\tau_0}{\rho}\left(1 - \frac{y}{r_0}\right) = \kappa^2y^2\left(1 - \frac{y}{r_0}\right)\left(\frac{du}{dy}\right)^2$$

whence

$$\frac{\tau_0}{\rho} = \kappa^2 y^2\left(\frac{du}{dy}\right)^2$$

Note 3

The temperature at the edge of the laminar sublayer was estimated by computing the thermal resistance of the laminar sublayer, buffer layer, and core. Then as a first approximation

$$\frac{t_w - t_1}{\Delta t_{\max}} = \frac{Pr_m}{\left[Pr_m + ln\,(1 + 5Pr_m) + 0.5\, ln\, \frac{Re_m}{60}\sqrt{\frac{f}{8}}\right]}$$

Knowing t_1 an estimate of y_1^+ can be made.

Then more precisely

$$\frac{t_w - t_1}{\Delta t_{\max}} = \frac{Pr_1}{\left[Pr_1 + ln\left(1 + Pr_m\left\{\frac{30}{y_1^+} - 1\right\}\right) + \frac{2.5}{y_1^+}\, ln\, \frac{Re_m}{60}\sqrt{\frac{f}{8}}\right]}$$

Note 4

For predictions corresponding to the experimental data of Colburn and Coghlan, and Eagle and Ferguson, the friction data of Stanton and Pannell for smooth pipe were used.

For the J. F. D. Smith comparison, his own experimental magnitudes of f were used and for the Morris and Whitman comparison the following isothermal friction-factor curve (rough pipe) was utilized:

Re_m	$\sqrt{\frac{8}{f}}$
2000	12.0
5000	13.7
10000	14.9
30000	16.6
100000	18.3

Acknowledgment

The authors wish to express their thanks to Mr. John Longwell who performed many of the computations in this paper, and to the National Youth Administration for assistance given.

BIBLIOGRAPHY

1 "On the Extent and Action of the Heating Surface for Steam Boilers," by Osborne Reynolds, Proc. Manchester Literary and Philosophical Society, vol. 14, 1874, pp. 7–12; Collected Papers, vol. 1, pp. 81–85.

2 "Bermerkung über den Wärmeübertragung im Rohr," by Ludwig Prandtl, *Physikalische Zeitschrift*, vol. 29, 1928, pp. 487–489.

3 "Conditions at the Surface of a Hot Body Exposed to the Wind," by G. I. Taylor, Great Britain Advisory Committee for Aeronautics, Reports and Memoranda, No. 272, vol. 2, 1916–1917, pp. 423–429.

4 "The Analogy Between Fluid Friction and Heat Transfer," by T. von Kármán, Trans. A.S.M.E., vol. 61, 1939, pp. 705–710.

5 "On the Dynamical Theory of Incompressible Viscous Fluids and the Determination of the Criterion," by Osborne Reynolds, Royal Society Philosophical Transactions, vol. 186A, 1895, pt. 1, pp. 123–164; Collected Papers, vol. 2, pp. 535–577.

6 "Über die ausgebildete Turbulenz," by Ludwig Prandtl, Proc Second International Congress for Applied Mechanics, Zurich, 1926, pp. 62–74.

7 "Gesetzmässigkeiten der turbulenten Strömung in glatten Rohren," by J. Nikuradse, *V.D.I.* Forschungsheft 356, 1932, 36 pp.

8 "The Mechanics of Turbulent Flow," by Boris A. Bakhmeteff, Princeton University Press, 1936, p. 72.

9 "Neuere Ergebnisse der Turbulenzforschung," by Ludwig Prandtl, *Zeitschrift des Vereines deutscher Ingenieure*, vol. 77, 1933, pp. 105–113.
10 Reference (8), p. 80.
11 "On the Coefficient of Heat Transfer From the Internal Surface of Tube Walls," by A. Eagle and R. M. Ferguson, Proc. of the Royal Society, vol. 127A, 1930, pp. 540–566.
12 "Heat Transfer to Hydrogen-Nitrogen Mixtures Inside Tubes," by A. P. Colburn and C. A. Coghlan. Paper in preprint form for presentation at A.S.M.E. Annual Meeting, New York, N. Y., December 2–6, 1940.
13 "Heat Transfer for Oils and Water in Pipes," by F. H. Morris and Walter G. Whitman, *Industrial and Engineering Chemistry*, vol. 20, 1928, pp. 234–240.
14 "Heat Transfer and Pressure Drop Data for an Oil in a Copper Tube," by J. F. Downie Smith, Trans. American Institute of Chemical Engineers, vol. 31, 1934, pp. 83–111.
15 "Heat Transfer in Automobile Radiators of the Tubular Type," by F. W. Dittus and L. M. K. Boelter, University of California Publications in Engineering, vol. 2, no. 13, 1930, pp. 443–461.
16 "Druckabfall und Wärmeübertragung bei turbulenter Strömung in glatten Rohren mit Berücksichtigung der nichtisothermen Strömung," by Georg Rohonczi, Doctorate Dissertation E. T. H. at Zürich.
17 "Die Grundgesetze der Wärmeübertragung," by H. Gröber and S. Erk, p. 174, Julius Springer, 1933; also Nusselt, W. Gesundh. Ing., vol. 38, p. 42, 1915.
18 "Modern Developments in Fluid Dynamics," edited by S. Goldstein, vols. 1 and 2, Oxford University Press, 1938, Chapters 5, 7, 8, 14, and 15.

Discussion

R. H. Norris.[8] The results achieved in this paper would be of greater practical value, if the authors could choose one or more relatively simple equations as practical approximations to their complicated correlation. Discrepancies up to 60 per cent are shown to be produced by the simple Equations [19] and [20], heretofore in common use.

This paper is of course concerned with the relation between heat transfer and fluid friction, in terms of the parameter $c\mu/k$, only for conditions of turbulent flow. For viscous flow, however, there is also a relation between heat transfer and fluid friction, the existence of which may, heretofore, have been obscured by the derivation of the theoretical results for heat transfer independently from those for friction, and the arbitrary choice of the arithmetic mean for the temperature difference.

In the region of viscous flow, when the logarithmic mean is used for the temperature difference, the variable denoted as $(Nu)/(Re)(Pr)$ can be plotted versus Re, as shown in Fig. 3, of a recently published paper.[9] When the ratio of pipe length L to the diameter D becomes sufficiently great, the value of $(Nu)/(Re)(Pr)$ becomes asymptotic to a straight line of slope —1, the same slope that the curve of friction factor has in the viscous region; thus $\frac{(Nu)}{(Re)(Pr)} = \frac{3.66}{(Re)(Pr)}$ for $(c\rho u_{\text{mean}}D^2)/(kL) < 4$, and $f = 64\ (Re)^{-1}$ so that $\frac{(Nu)}{(Re)(Pr)} = \frac{0.458(f/8)}{(Pr)}$ for $(c\rho u_{\text{mean}}D^2)/(kL) < 4$.

For larger values of $(c\rho u_{\text{mean}}D^2)/(kL)$, with laminar flow, the slope becomes flatter, but this can be considered as a thermal "entrance effect," resulting from the sudden change in surface temperature at the start of the heated section of the duct. The viscous-flow–friction-factor curve would likewise flatten if the pipe length were sufficiently decreased and the calming section omitted.

It should be noted that f, as defined by the authors, is 4 times the value sometimes given in the definitions of f by other authors.

E. S. Smith.[10] The writer wishes to ask the authors just what they would expect to see happen in the throat of a venturi tube in which the flow is at a sufficiently high Reynolds number that turbulence would be high in a long straight pipe of the same diameter and with the same fluid moving at the same velocity? The eddies should be powerfully suppressed in the venturi throat and the shear would be high near the wall of the throat since the velocity distribution is so nearly uniform. This question is raised in a paper by W. S. Pardoe,[11] in which his results can be explained by having the temperature of the water next to the wall under such conditions substantially equal to that of the ambient air. Now this does not seem to the writer to be reasonable, although he does not profess to be an expert on heat transfer. It would seem that there should be some gradient or drop of temperature from the air to the wall, assuming that the air temperature is higher.

Th. von Kármán.[12] This paper is a valuable contribution to the solution of the problem of heat transfer. The authors give an excellent presentation of the similarity between heat and momentum transfer following the lines indicated in the writer's paper[13] on the same subject. The new expression "buffer layer" for the transition layer between perfectly laminar and perfectly turbulent regions, introduced by the writer in order to explain the discrepancies between experiments and the theories of Prandtl and G. I. Taylor, is quite appropriate. The investigation is definitely carried further beyond the results given in the former paper[13] by a more detailed analysis of the turbulent region and especially by consideration of the temperature changes in the cross section of the pipe. It is gratifying that, by a more exact analysis, the agreement between theory and experiments appears even closer than the writer has found.

Authors' Closure

For low magnitudes of the Prandtl modulus, the empirical expressions, Equations [19] and [20], or their equivalent, such as Colburn's j function (19),[14] approximate the analytical relation with some accuracy. Fig. 11 of this closure illustrates the analytical equation plotted in terms of Colburn's j function and Fig. 12 reveals a comparison of Colburn's empirical equation, Reynolds', Prandtl's, and von Kármán's analogies, and Equation [11] of this paper. Below magnitudes of Prandtl's modulus equal to 10 the average slope of the analytical curve is approximately —$^2/_3$. However, at high magnitudes of Pr the slope of the analytical curve becomes —1, which indicates that, for low rates of heat transfer (isothermal) and high magnitudes of Prandtl's modulus, Nusselt's modulus will be independent of Prandtl's modulus. Thus, the exponent of the Prandtl modulus in any empirical equations, such as Equation [19] or [20], should be a function of Pr and the rate of heat transfer. It appears to be fortuitous that a constant exponent yields a satisfactory correlation.

Several other modes of empirical correlation, using viscosities at temperatures other than the mixed mean temperature have

[8] General Engineering Laboratory, General Electric Company, Schenectady, N. Y. Jun. A.S.M.E.

[9] "Laminar-Flow Heat-Transfer Coefficients for Ducts," by R. H. Norris and D. D. Streid, Trans. A.S.M.E., vol. 62, August, 1940, pp. 525–533.

[10] C. J. Tagliabue Mfg. Co., Brooklyn, N. Y. Mem. A.S.M.E.

[11] "Effect of Ambient Temperatures on the Coefficients of Venturi Meters," by W. S. Pardoe, Trans. A.S.M.E., vol. 63, 1941, Figs. 5 and 10, p. 458.

[12] Director, Daniel Guggenheim Aeronautical Laboratory, California Institute of Technology, Pasadena, Calif. Mem. A.S.M.E.

[13] "The Analogy Between Fluid Friction and Heat Transfer," by Th. von Kármán, Trans. A.S.M.E., vol. 61, 1939, pp. 705–710.

[14] Numbers (19) to (23) refer to Bibliography at the end of this closure; all other numbers in parentheses refer to the Bibliography at the end of the paper.

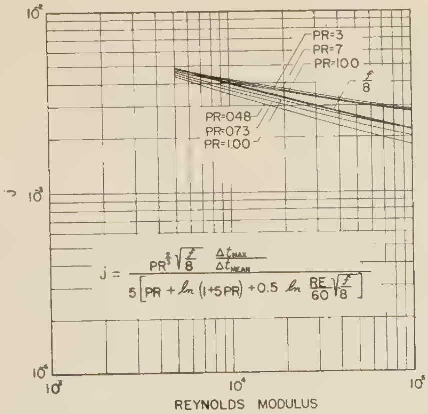

FIG. 11 PLOT OF COLBURN'S j FUNCTION $[Nu/(Re\,Pr)]Pr^{2/3}$ CALCULATED FROM EQUATION [11] FOR MAGNITUDES OF PRANDTL'S MODULUS BETWEEN 0.48 AND 10

(The friction factor over eight ($f/8$) is also presented as a function of Reynolds' modulus.)

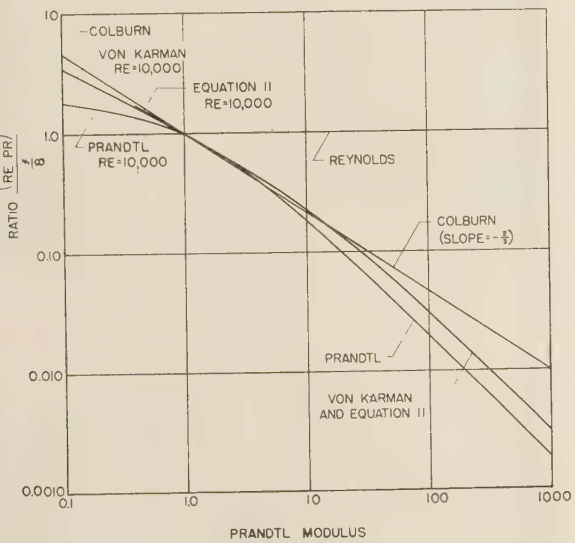

FIG. 12 COMPARISON OF REYNOLDS', PRANDTL'S, COLBURN'S, AND VON KÁRMÁN'S EQUATIONS FOR VARIOUS MAGNITUDES OF PRANDTL'S MODULUS FROM 0.1 TO 1000 FOR REYNOLDS' MODULUS = 10,000

(The magnitudes as computed from Equation [11] are also presented.)

been proposed (14, 17). Fair success in bringing experimental data together was attained by these methods so that Fig. 10 of the paper may leave the impression of overestimating the inaccuracy of empirical correlations. However, the same fundamental limitations apply to these later correlations as to those presented in the paper. Equations [19] and [20], or similar forms, are in current use in industrial design.

An important phenomenon which will cause scattering of data, particularly at low values of Prandtl's modulus, is the entrance effect mentioned in the body of the paper.

A uniform temperature distribution exists at the entrance to a heating section, which will change to a distribution such as is shown in Fig. 3 of the paper if the heating section is of sufficient length. If, however, the test section is short, it is possible that the equilibrium-temperature distribution will not be reached. Application of Equation [11] to entrance sections will yield magnitudes of Nusselt's modulus which are too low (for instance, $\frac{\Delta t_{mean}}{\Delta t_{max}} = 1$ for uniform temperature distribution at entrance) because it applies only where the velocity and temperature distributions have been established and steady-state conditions obtain. Due to the transition conditions existing in the thermal quieting sections, the magnitudes of Nusselt's modulus have been found to exceed those which obtain after the temperature distribution has been established (20).

If the linear variation in unit shear in the buffer layer is utilized, rather than the constant wall-unit shear, the predicted magnitudes of $Nu/Re\,Pr$ for low magnitudes of Reynolds' and Prandtl's moduli fall below those shown in Fig. 5 of the paper, and approach closely the experimental points of Colburn and Coghlan (21).

Referring to E. S. Smith's comments, regarding nonisothermal flow in venturi meters, a difference in temperature between the ambient air and the fluid within the pipe will cause heat transfer to (or from) the fluid. Heat transfer will occur throughout the entrance length as well as in the venturi proper. This temperature difference will cause free convection in the entrance length, and will also change the viscosity of the fluid in the laminar sublayer in contrast to the magnitude corresponding to the mixed mean fluid temperature. These phenomena may influence the discharge coefficient of the venturi and account for the fact that the nonisothermal discharge-coefficient–Reynolds-number curve crosses the isothermal curve.

Dr. Theo. von Kármán's comments are indeed gratifying. Each correction as it is applied yields a prediction which more nearly approaches the best experimental results. The segregation of the resistances, due to each of the three fluid layers, has aided the authors in making further refinements to the isothermal-heat-transfer theory. The attendant success served as a stimulus toward the analysis presented (yet incomplete) for nonisothermal flow.

GENERAL COMMENTS

Four excellent articles (22, 23, 24, 25) on this subject have come recently to the attention of the authors. Mattioli (23) also utilizes y^+ as a point function. He considers the transfer of vorticity and momentum simultaneously.

The similarity of $Nu/Re\,Pr = f_c/Gc_p$, where G is the unit mass-flow rate, to the number of transfer units (N.T.U.) deserves attention in view of the move to assign a name to the former. In the tube-to-fluid exchanger, the N.T.U. $= (Nu/Re\,Pr)S/A$, where S is the tube area through which heat flows and A is the cross-sectional area of the pipe.

BIBLIOGRAPHY

19 "Mass Transfer (Absorption) Coefficients," by T. H. Chilton and A. P. Colburn, *Industrial and Engineering Chemistry*, vol. 26, 1934, pp. 1183–1187.

20 "Die Wärmeübertragung," by M. ten Bosch, Third edition, Julius Springer, Berlin, 1936, p. 121.

21 T. Shimazaki, private communication.

22 "Mass Transfer and Friction in Turbulent Flow," by T. K. Sherwood, Trans. American Institute of Chemical Engineers, Dec. 25, 1940, pp. 817–840.

23 "Theorie der Wärmeübertragung in glatten und rauhen Rohren," by G. D. Mattioli, *Forschung auf dem Gebiete des Ingenieurwesens*, Ausgabe A, Band 11, no. 4, July-Aug., 1940, pp. 149–158.

24 "Über die Gesetzmaszigkeiten der Wärme- und der Stoffübertragung auf Grund der Strömungsvorganges im Rohr," by E. Hofmann, *Forschung auf dem Gebiete des Ingenieurwesens*, Ausgabe A, Band 11, no. 4, July-Aug., 1940, pp. 159–169.

25 "Die Wärmeübertragung in turbulenten Reikungschichten," by H. Reichardt, *Zeitschrift für Angewandte Mathematik und Mechanik*, Band 20, no. 6, December, 1940, pp. 297–328.

Effect of Ambient Temperatures on the Coefficients of Venturi Meters

By W. S. PARDOE,[1] PHILADELPHIA, PA.

PRIOR to 1930, the writer attempted to plot the coefficients of a large number of venturi tubes against Reynolds' number and found the results rather unsatisfactory. This would, of course, be easily explained on the basis of proportional roughness and, as there seemed to be no definition of this function, the results were considered most unsatisfactory.

During that period, on three occasions venturi tubes were sent to the University of Pennsylvania laboratory for calibration. Each time the makers proceeded to construct a meter to fit the laboratory calibration. About 2 months later, when the combination of meter and venturi tube was tested in the laboratory with different temperature water, the coefficient was satisfactory on the flat part of the curve and at some point lower, about 3 fps throat velocity, but in between the coefficients varied as much as 0.5 per cent. This, of course, was all very confusing, as it was contradictory to any plot against Reynolds' number.

In 1931, the Simplex Valve and Meter Company presented the laboratory with an 8 × 3$^3/_8$-in. bronze venturi meter. The results of tests on this meter for 46, 63, 69, and 74 F water are shown in Figs. 1 and 2. These are quite similar to other curves on venturi meters. It will be noticed that on the flat part of the curve the coefficient is 0.99 and at 3 fps throat velocity the coefficient is 0.969. Curves such as that for 46 F became known

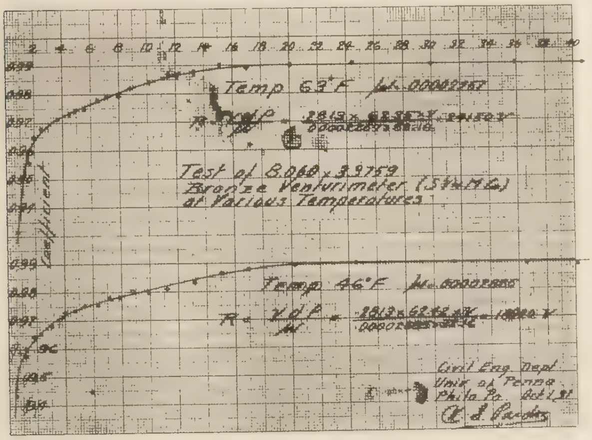

Fig. 1

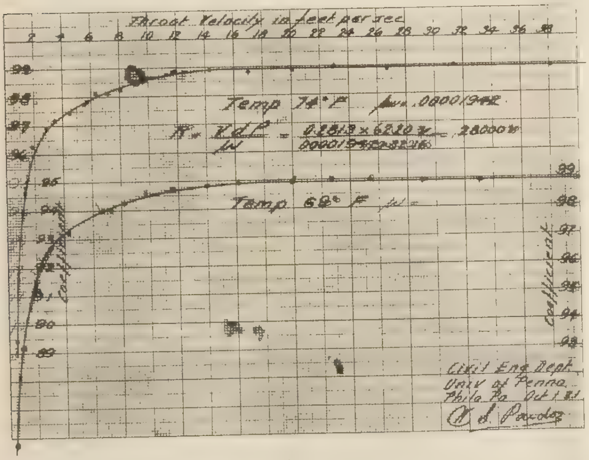

Fig. 2

Figs. 1 and 2 Test of 8.06-In. by 3.3759-In. Bronze Venturi Meter at Various Temperatures

(Courtesy of Simplex Valve and Meter Company.)

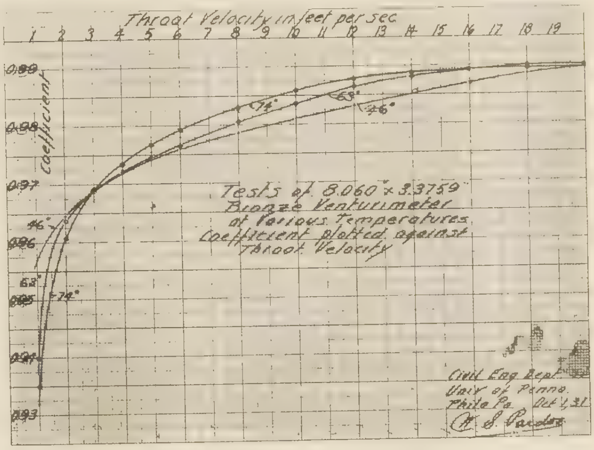

Fig. 3 Tests of Venturi Meter at Various Temperatures Showing Coefficient Plotted Against Throat Velocity

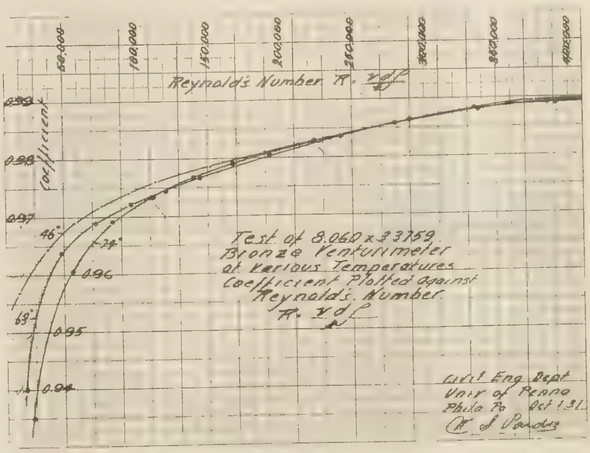

Fig. 4 Tests of Venturi Meter at Various Temperatures; Coefficient Plotted Against Reynolds' Number

[1] Professor, Department of Civil Engineering, University of Pennsylvania.

Contributed by Special Research Committee on Fluid Meters and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

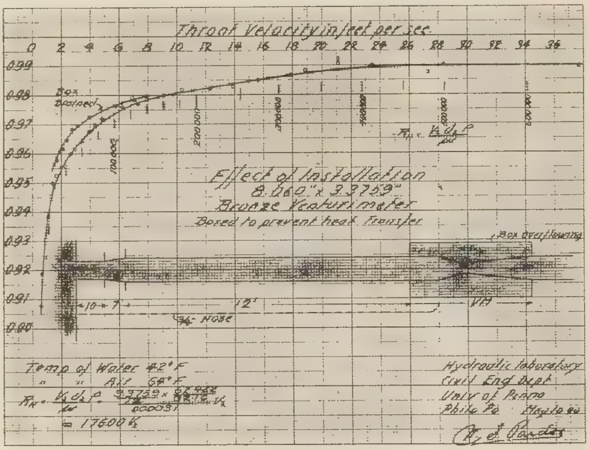

Fig. 5 Effect of Installation of Venturi Meter Boxed to Prevent Heat Transfer; Water Temperature 42 F

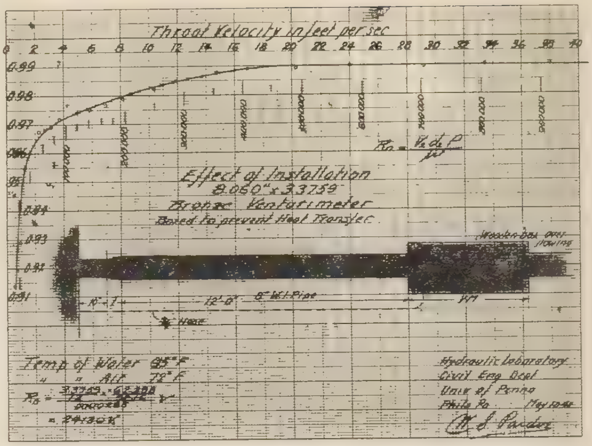

Fig. 8 Effect of Installation of Venturi Meter Boxed to Prevent Heat Transfer; Water Temperature 61.5 F

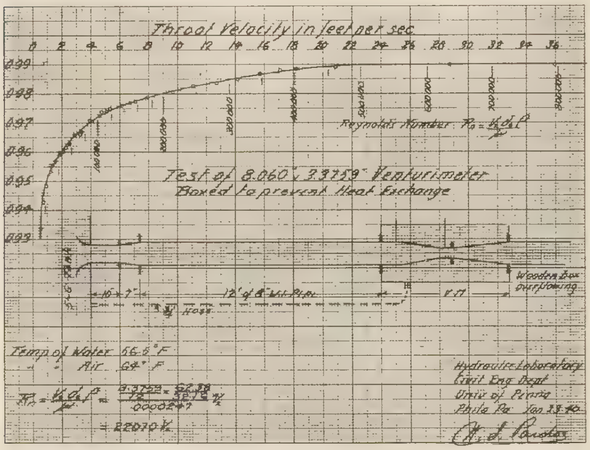

Fig. 6 Test of Venturi Meter, Boxed to Prevent Heat Transfer; Water Temperature 53 F

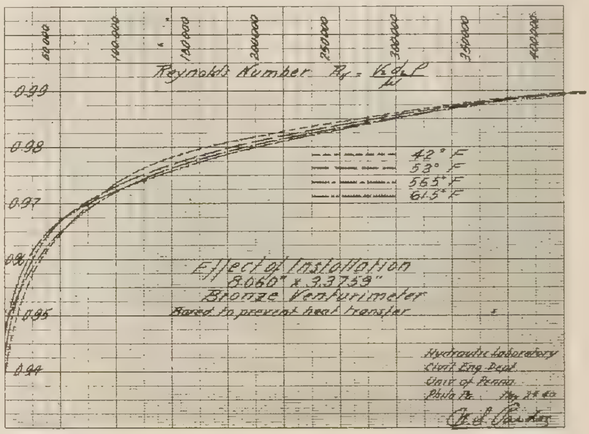

Fig. 9 Tests of Figs. 5 to 8, Inclusive, Plotted Against Reynolds' Number

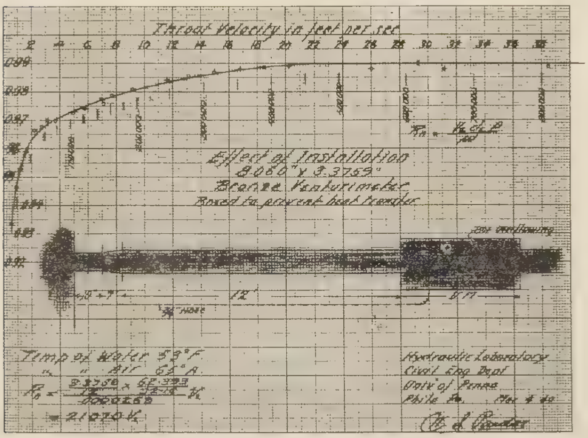

Fig. 7 Effect of Installation of Venturi Meter Boxed to Prevent Heat Transfer; Water Temperature 56.5 F

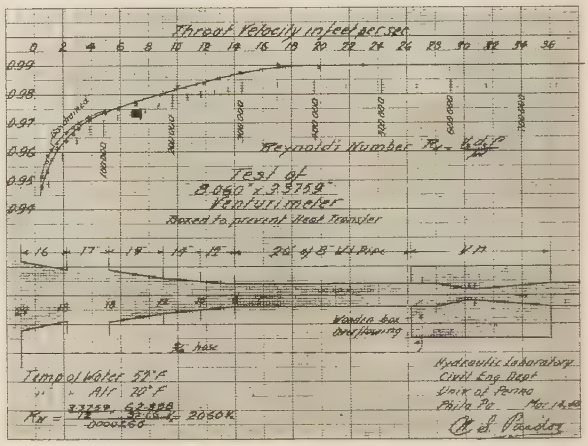

Fig. 10 Tests Made to Check the Effect of Installation on Venturi Meter

as cold-water curves and those for 74 F as hot-water curves.

In Fig. 3, are plotted the curves for 46, 63, and 74 F against throat velocity with the scales exaggerated. It will be noted that all these curves pass through 0.969 at 3 fps. These curves plotted against Reynolds' number are shown in Fig. 4, and do not make a very good showing.

On a number of occasions the author has shown these curves to the Committee and to hydraulic groups of the A.S.C.E., A.S.M.E., and S.P.E.E. The only response ever received has been to question the experimental work rather than to explain the facts. To be in error in the experimental work would mean that the author could not measure 0.15 per ft accurately, which he certainly can.

On a number of occasions, these curves have been placed in the hands of visitors to our laboratory and others who are interested in such matters. Finally a set was given to Dean Goff of this institution, who, after some time, suggested the possibility of heat transfer entering the problem, that is, with cold water being measured and the room temperature at 70 F or thereabouts. The boundary layer at the throat was raised slightly in temperature, thus decreasing the value of μ, decreasing the boundary shear τ_0, and increasing the discharge for low temperatures.

In order to check this, a wooden box was built around the venturi meter as shown in Fig. 5 and runs were made with a water temperature of 42 F, giving the results shown. The box was drained and the curve immediately rose 0.7 per cent at 3 fps, as indicated. Tests were also conducted with the same arrangement at 53, 56.5, and 61.5 F, as shown in Figs. 6, 7, and 8, respectively.

In Fig. 9, the foregoing tests are plotted against Reynolds' number making quite a creditable showing. The variation of 0.2 per cent must be blamed on the experimenter. To check the effect of installation, tests were made as indicated in Fig. 10. The variations are not extreme.

From the foregoing it would appear that there is a distinct effect of ambient conditions on the coefficient of venturi meters. Unless they are thoroughly lagged during the test and in the permanent installation, they cannot give accurate results under all circumstances.

Discussion

R. W. Angus.[2] Although this paper is brief, it is a distinct contribution to the subject and gives an explanation of what appeared to be an inconsistency in the meter experiments. Evidently the rate of heat transfer through the walls of the meter tube is sufficient to produce erratic coefficients at low throat velocities, while it does not affect those at high throat velocities, presumably because of the time element.

One of the practical applications of the results is that, in the author's meter, the coefficients for different temperatures and for a throat velocity of 10 fps differ by 0.5 per cent. There is about the same difference at 2 fps, in both cases for the meter tested unjacketed; the differences are less than this at intermediate velocities. Inasmuch as it is impossible, in most cases, to jacket meters in service, these errors would seem to be inherent in the use of this instrument but they are likely to be smaller in meters which are larger than the one covered by the author's work. As an effort would be made, wherever possible, to select a meter working with throat velocities corresponding to the flat part of the curve, the error may not be serious in practice, but yet it is there, giving a sense of uncertainty as to the meter indication. Meters are in common use on waterworks plants where the highest degree of accuracy is usually only necessary when they are being used to measure pump efficiency, in which case the rated pump discharge generally corresponds with the higher throat velocities, the coefficient being constant.

The writer has compared the coefficients in this paper with those given in the Fluid Meters Report.[3] The latter shows[4] that, for the size meter mentioned in the paper, and with water at 68 F, the coefficient has a constant value at over 7 fps throat velocity, while Fig. 2 of the paper shows that, for the meter used with water at 69 F, the coefficient reaches its constant value at about 20 fps. Both of these figures and curves apply to Simplex meters, although one is for an iron tube with bronze throat while the other is for a bronze tube.

The writer has great confidence in the accuracy of the author's work but believes that the high degree of precision he has achieved may cause his results to be misinterpreted. The report[3] mentioned bears every evidence of great care in its preparation and must be looked upon as an excellent piece of work. In that report two sets of curves[5] are given which show many sizes of venturi tubes having a diameter ratio of 0.5. From these curves the coefficients may easily be read to 0.1 per cent. Equation [261][6] enables the coefficient to be calculated for any other diameter ratio with the same degree of accuracy; in fact, in the examples given, the coefficients are worked out to 0.01 per cent. The data for these are for the most part based on the experiments of the author. Their accuracy is unquestioned.

The experimental work extended only to tubes for pipes not over 12 in. diam and for a more or less ideal setup of the tubes. However, but slight guidance is given as to what is meant by the proper setup and how it may conveniently be determined whether or not the coefficients will apply to a tube being used in a practical case.

During the summer of 1939, the writer was retained by the City of Toronto, Canada, to conduct tests on eleven new pumping units installed in one of the city pumping stations, and presented the results of these tests in a recent paper.[7] The guarantees for efficiency were unusually high and the penalties for nonfulfillment of the contract were extremely stringent, particularly on some of the units, so that the writer was obliged to use every possible precaution to secure accuracy.

Discharges were measured with venturi tubes, and the pipes leading the water thereto were as well laid out as possible, the bends being of long radius; in most cases there was a long straight pipe upstream from the tube. In the meter used on one of the pumps, the discharges, computed from the Fluid Meters Report coefficient of 0.9876, were known to be too high. A careful volumetric calibration of the tube was subsequently made, which gave an actual coefficient of 0.9545. There were slight bends upstream from the tube but, ordinarily, these could not have accounted for this great difference. Since the tube had been examined and calipered before erection and again after it had been in use for a time, there appeared to be no other cause for this difference in coefficients than a probable error in the one given by the Fluid Meters Committee, or else a disturbance from the pump.

After discovering this discrepancy, the writer discussed the matter with a number of engineers and found that there were

[2] Professor of Mechanical Engineering, University of Toronto, Toronto, Canada. Honorary Member A.S.M.E.

[3] "Theory and Application of Fluid Meters," A.S.M.E. Report of Fluid Meters Committee, Fourth edition, 1937.

[4] Ibid., Fig. 64, p. 101.

[5] Ibid., chapter "Discharge Coefficients for Venturi Tubes," Figs. 63 and 64, pp. 100 and 101, respectively.

[6] Ibid., p. 99.

[7] "An Improved Technique for Centrifugal-Pump-Efficiency Measurements," by R. W. Angus, Trans. A.S.M.E., vol. 63, January, 1941, pp. 13–19.

many cases in which the coefficients differed from those recommended by the committee. The writer, therefore, expresses the hope that the Fluid Meters Committee will give this matter further serious consideration, and present to the Society all available evidence in support of its findings, particularly in the case of large meters. Its report should be supplemented by instructions which will assist those using the meters in determining whether the coefficients are likely to apply. While such a device as a straightening vane is usually helpful there is no desire to undergo the expense of installing the device unless it is necessary. Further, the examination of conditions above the meter tube is frequently extremely difficult and will not be undertaken unless there is some definite reason to do so.

In the present state of the art no one knows whether a given setup corresponds to the Fluid Meters Committee's coefficients or not and, further, the report seems to be of theoretical rather than practical value because it does not suggest that actual coefficients are known to deviate from those given, nor does it attempt to quote these coefficients. The tolerance of ± 0.75 per cent[8] in the coefficient for cast-iron tubes, as given in the report, seems scarcely consistent with the accuracy with which the coefficient is worked out in the examples appearing elsewhere[9] in the report.

M. M. Borden.[10] Before such information can be used for the closer control of venturi coefficients and to limit their tolerances, the matter will require further investigation involving venturi sizes, ratios, effect of roughness, and through a greater range of temperature difference.

The effect of enlarging the area of the throat, because of higher temperatures of a fluid in it, should cause the coefficient to increase. Hence, the full effect of a difference of temperature within and without the tested tubes may be slightly different than indicated.

A plot of the upper and lower coefficients to throat-velocity values for the venturi boxed and unboxed shows a generally narrower zone for the boxed unit. The area, included between such upper and lower limits, is about 10 per cent less for the boxed venturi for regions of 1 to 2 fps and 25 fps throat velocity.

The author's findings suggest the use of completely insulated cold-water venturis by enclosing them in suitable coverings, as is partially done with the covered venturis measuring hot fluids.

E. S. Smith.[11] As one who is now a disinterested observer but formerly an active member of the Fluid Meters Committee, the writer can join a discussion of fluid metering only infrequently and has included in this discussion material which might otherwise be added to the discussion of a current paper[12] on the subject of nozzles. The present discussion gives something of the background of the paper and some material which is only indirectly related to the former bone of contention, i.e., the acceptance of the method of similarity in fluid metering.

Over an extended period, the subcommittee on "similarity," consisted of Messrs. Pardoe, Spitzglass, and the writer, with Messrs. Pigott and Buckingham (deceased) lending moral support on occasion. Messrs. Spitzglass and Pardoe long and spiritedly opposed the acceptance of the Reynolds number as a basis for the correlation of fluid-meter coefficients on the grounds of unstated limitations of the method of similarity and dimensional homogeneity. However, before his death, Mr. Spitzglass succeeded in making it clear that his acceptance of such correlation required that account be taken of the roughness of the line and/or meter, relative to its diameter, a phenomenon which is sometimes known as the "scale effect." In his present paper the author has likewise succeeded in making it clear that his acceptance awaited only an accounting for an additional factor, i.e., the effect of a different temperature of wall and fluid for water, under the conditions of his tests with water.

Since the acceptance of a useful engineering method is too often delayed by blind advocacy equally with opposition, there is a natural question as to the writer's position. He admittedly made several elementary and simplified statements in an effort to picture clearly the concepts of similarity involved. Some of these statements taken alone might give the impression that the writer was blind to any points such as were stressed by the opponents of the acceptance of this method.

Consequently, it is necessary to point out that a paper,[13] by the writer in 1923, covered the scale effect[14] in Fig. 2 (which is based on some of the author's values) and elsewhere in the text,[15] while the closure[16] noted that errors result from a difference of the temperature of the fluid from that of the walls of the pipe or meter.

In spite of this broad hint, the author's delay in finding the cause of the errors which his tests included is understandable, since the writer for one did not suspect that this was the cause of inconsistencies in tests which were of the order of errors common in hydraulic testing, possibly including the tests which are noted in the second paragraph of the paper. At one time, the writer checked and found correct everything but the atmosphere in the author's laboratory. He is to be congratulated both for his persistence and the excellence of his tests and to be forgiven for his delay. If he had worked much with fluids other than water, he would certainly have earlier embraced the use of the Reynolds number as the only rational basis for their metering in practice and might have missed finding the cause of the slight error which he now stresses.

His location of this error is a credit to the accuracy of his tests rather than an indication that such error amounts to much. Taking a velocity of 2 fps and the water and air temperatures, respectively, at 50 and 70 F, this error could be caused by a difference of level of only 0.6 ft of water at the stated temperatures in the vertical connecting pipes. Such differences often occur upon a change of rate with usual venturi recording and integrating instruments, with their large wells or bells, which cause a movement of liquid in the pressure pipes which is much larger than the metering head. In controlling flow, transient effects are important and such errors may be of consequence. This less than 1 per cent error would amount to less than 0.002 in. on a chart or integrator in which 0.1 in. corresponds with 1 fps throat velocity, a fact which incidentally shows the importance of precise setting at the lowest operating rate.

If the "box-drained" values of Figs. 5 and 10 of the paper be multiplied by the ratios of water viscosities at the water and air temperatures, the corrected values fall close to the "box-overflowing" values and constitute a rough check of the Goff theory. Of course at the higher rates, the effect of heat transfer disappears. Possibly a current paper[17] on heat transfer may be useful if the

[8] Reference (3), p. 128.

[9] Ibid., p. 104.

[10] Chief Engineer, Simplex Valve & Meter Company, Philadelphia, Pa. Mem. A.S.M.E.

[11] Hydraulic Engineer, C. J. Tagliabue Manufacturing Company, Brooklyn, N. Y. Mem. A.S.M.E.

[12] "Discharge Coefficients of Long-Radius Flow Nozzles When Used With Pipe Wall Pressure Taps," by H. S. Bean, S. R. Beitler, and R. E. Sprinkle, Trans. A.S.M.E., vol. 63, 1941, pp. 439–445.

[13] "The Oil Venturi Meter," by E. S. Smith, Jr., Trans. A.S.M.E., vol. 45, 1923, pp. 67–75.

[14] Ibid., Fig. 2, p. 71.

[15] Ibid., par. 17, p. 71.

[16] Ibid., first par. of closure, p. 75.

[17] "Remarks on the Analogy Between Heat Transfer and Momentum Transfer," by L. M. K. Boelter, R. C. Martinelli, and Finn Jonassen, Trans. A.S.M.E., vol. 63, 1941, pp.447–455.

generally noneddying character of the flow in the throat be considered. In other words, the stated comparisons in Figs. 5 and 10 indicate that the temperatures of both the water and the wall of the "box-drained" tube are that of the ambient air—an indication that is contrary to the accepted heat-transfer teaching that the wall temperature of a conduit containing running water will approximate that of the water instead of that of the air. An explanation is accordingly requested.

Due to the fact that the heat-transfer coefficients for turbulently flowing water and still air are of different orders, changes of wall temperature from that of the stream are ordinarily insignificant except at low velocities. It should not be difficult for the author to obtain a few temperature and pitot traverses and settle this matter in his closure, instead of relying on speculation which is supported by over-all calibrations only, of which there are many.

Errors are to be expected with uninsulated hot-water meters, supercooled liquid-ammonia meters, and gas meters which are heated to prevent the formation of deposits at the throat. A rise of gas temperature should lead to a negative instead of a positive error, since the viscosity of a gas increases with temperature while that of water then falls.

It cannot be assumed that orifice meters would be free from the Goff error, although such error should be in the opposite direction from that of the venturi. Nozzles contained in the pipe and entirely surrounded by fluid should be practically immune to this error except for large diameter ratios which, like orifices, are sensitive to the upstream velocity distribution. For highly accurate venturi measurements, it is possible to use the heated type of gas venturi with the circulation in the heating chamber provided by the differences of pressure existing in the downstream cone.

No effect is to be expected on the flat portion of the coefficient curve for any differential producer. This general principle was brought out by Swift in his papers,[18] which deal with other factors affecting the correlation of flowmeter coefficients with the Reynolds number.

Since a high value of venturi coefficient exists on the flat portion of the curve and starts to fall off at a relatively high value of the throat velocity at usual temperatures and, hence, of the Reynolds number, such a venturi is relatively sensitive to this error. By roughening the approach curve, as was originally suggested to the writer by Herbert N. Eaton, chief of the National Hydraulic Laboratory, and shortening its radius, the flat portion of the curve can be extended to much lower values of Reynolds' number. Hodgson[19] early reduced the length of the throat cylinder and placed the throat taps at the end of the curved portion of the approach curve to lengthen the flat portion of the coefficient curve. The shortening of the radius of the approach curve was used, e.g., in the German standard nozzles to lengthen the flat part of this curve, although such design also includes a lengthening of the nozzle throat to cause filling of the throat by the re-expansion of the contraction which follows the shorter approach radius.

The writer has tried a number of different combinations of approach-curve radius, length of throat, and roughness, and has obtained a worth-while extension of the flat part of the curve for nozzles. Some of these nozzles were tested by the author and accepted with a 0.25 per cent accuracy guarantee, including their indicating, recording, and integrating instruments on a portion of the flat part of the coefficient curve. This allowed 0.1 per cent full scale for the author's calibration and installation effects and 0.15 per cent for the instrument from maximum to less than one half of the maximum (which corresponds to within 0.07 per cent of full scale). This accuracy was attained with stock instruments which were special mainly as to the spacing of the dial and chart graduations, working clearances and tolerances, and the setting at the lowest operating rate. The performance of such instruments is fairly comparable with the 0.1 per cent of full scale reading reported with a recently developed, highly special instrument used in steam-turbine traverses. These instruments[20] also involved a cam and roller and an integrator wheel-on-a-disk.

Since each diameter-ratio of nozzle has a different "shape" and it seems necessary for the best shapes to be available to all, it is desirable for the A.S.M.E. Fluid Meters Committee to sponsor the necessary tests rather than that the best design become the property of any manufacturer. For nozzles, the design would be essentially the same as for the venturi although the approach may possibly be cut off short of the throat cylinder, as suggested by Witte, so that a slightly conical throat may replace the cylindrical throat of the venturi.

The writer referred his recommendations to the committee, several years before resigning from active work thereon, as a starting point toward such a design and then urged such action. The author's paper revives the importance of a design which is free from even small errors due to the Goff effect. However, a design which is best for cold water under positive pressure may have a much shorter life than the long-radius venturi or nozzle where cavitation exists.

Author's Closure

The writer agrees with Professor Angus that the effect of ambient temperature will vary with the size of the venturi meter. It will also vary with the design, paint, roughness, and insulation, if any.

The 8 × $3^3/_8$-in. bronze venturi meter (coefficient 0.99), used in these experiments, although it is of the same proportions, is much smoother than the fifty-seven cast-iron venturi meters used as a basis for the A.S.M.E. curves, from which a value of 0.983 would be obtained. Coefficients of rough venturi meters become flat at much lower throat velocities. Coefficients of venturi meters should not contain more than three significant figures, although Prof. Angus uses four.

Some idea of the "proper setup" may be obtained by reference to a previous paper[21] by the author.

The venturi meter at Toronto, which gave a coefficient of 0.9545 instead of 0.9876 or 3.3 per cent low, was one of eleven meters. All the others gave satisfactory results, using the A.S.M.E. coefficient values (in which the author is not disinterested). Prof. Angus examined and calipered the meter and found the pressure taps in good condition. Under these circumstances, the only thing which could lower the coefficient 3.3 per cent would be a vortex started in the centrifugal pump. That this could produce the result is evident from Fig. 44[22] of the author's previous paper. The author has found it advantageous in all cases where a vortex may form to use cross straightening vanes 4 diam in length ahead of the venturi meter. Until this is done or an exploration made ahead of the Toronto meter, this matter cannot be considered settled. The author cannot believe a 42

[18] "Orifice Flow as Affected by Viscosity and Capillary," by H. W. Swift, *Philosophical Magazine*, series 7, vol. 2, 1926, pp. 852–875; "Operational Factors in Orifice Flow," vol. 5, 1928, pp. 1–17; "The Calibration of an Orifice," vol. 8, 1929, pp. 409–435.

[19] "The Measurement of the Flow of Gases and Liquids by Means of Orifices, Nozzles, and Venturi Tubes," by J. L. Hodgson, World Engineering Congress, vol. 4, part 2, Tokio, 1929, p. 113, Fig. 10.

[20] "Automatic Integrating Pressure-Traverse Recorder for Study of Flow Phenomena in Steam-Turbine Nozzles and Buckets," by H. Kraft and T. M. Berry, Trans. A.S.M.E., vol. 62, Aug., 1940, pp. 479–488.

[21] "The Effect of Installation on the Coefficients of Venturi Meters," by W. S. Pardoe, Trans. A.S.M.E., vol. 58, 1936, pp. 677–684.

[22] Reference (21), p. 683.

× 27-in. venturi meter, correctly constructed and without vortex flow, could possibly give such a low coefficient.

Mr. Borden suggests a great deal of additional experimental work. The author does not propose to do any beyond establishing the fact of an effect of ambient on the coefficients of flow meters generally.

Mr. Smith has quite misinterpreted the paper and has forgotten our "bone of contention," i.e., that Reynolds' number is

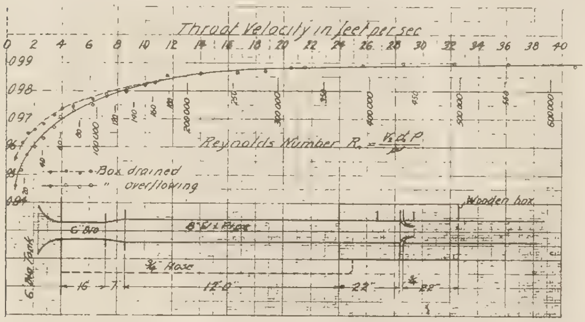

Fig. 11 Effect of Ambient Temperature on Coefficients for Flow Nozzle

(A.S.M.E. flow nozzle 8:300:1; 8.071 × 3.0012 in.; temperature of water 41 F; temperature of air 67.5 F.)

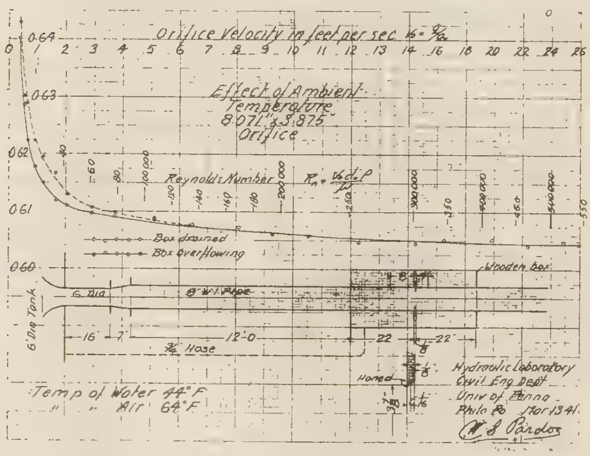

Fig. 12 Effect of Ambient Temperature on Coefficients for Pipe Orifice

(Size 8.071 × 3.875 in.; temperature of water 44 F; temperature of air 64 F.)

not a satisfactory criterion for plotting the coefficients of venturi meters as usually constructed and installed, it cannot be used on the curved part of the coefficient curve, and has no meaning on the flat part. The errors inside the metering range of manufacturers' meters have already been mentioned and are considerable. The author now includes flow nozzles and pipe orifices as being similarly affected, as indicated by the curves in Figs. 11 and 12 of this closure. Mr. Smith will have to revise his ideas with respect to the orifice and also his very "rough" check of the Goff theory. He suggests that "it should not be difficult to get a few temperature and pitot-tube traverses." The author must disagree and believes this almost impossible as we are dealing not with the laminar boundary layer but with the sublaminar boundary layer, at very low velocities. If Mr. Smith has in mind a pitot tube to measure the wall velocity (if any), there are many individuals who will be very much interested in learning of it.

For a number of years, the author kept up a correspondence with Dr. Edgar Buckingham, sending him the original curves as obtained. He expressed the belief that "we were leaving out some factor," but at no time suggested heat transfer, nor did any other member of the Committee, although all of them were kept continually aware of the unsatisfactory results obtained with water. The effect is too complicated to evaluate from any reasonable number of experiments depending upon, as it does, the specific heats of the fluids, the design, the roughness, the materials of construction, the paint, the insulation, and probably other things. In other words, life is too short for this busy "hexagenarian" even to start.

During the discussion of the paper, Messrs. Sprenkle, Beitler, and Bean suggested that the flow nozzle was not affected by ambient temperature. Herewith is shown test results of an A.S.M.E. flow nozzle 8:300:1, Fig. 11, indicating that they are not immune. Fig. 12, for an 8.071 × 3.875-in. orifice also shows that it is affected in quite a similar manner, and not the reverse as suggested.

The coefficient of a flowmeter may be expressed

$$C = \sqrt{\frac{1 - \beta^4}{\alpha_2 - \alpha_1\beta^4 + k}}$$

in which

$\beta = d_2/d_1$

k = coefficient of loss in $hf = k\,\dfrac{\overline{V}_2^2}{2g}$

α = ratio between actual kinetic energy per pound and kinetic energy per pound or $\overline{V}^2/2g$

$\alpha_2 = 1.00$

If the main velocity traverse given by the seventh-root law for smooth pipe

$$V = V_{\max}\left(\frac{y}{r_0}\right)^{1/7}$$

Pipe factor = 0.817

$$\alpha_1 = 1.056$$

$$\therefore\ k = \frac{1 - \beta^4}{C^2} - (1 - 1.056\,\beta^4)$$

Take an 8.071 × 3.0012-in. flow nozzle at 2 fps throat velocity

C_d = 0.9662 box drained
C_0 = 0.9606 box overflowing
β = 0.37184 β^4 = 0.019118

$$k_d = \frac{1 - 0.019118}{0.9662^2} - (1 - 1.056 \times 0.019118) = 0.0715$$

$$k_0 = \frac{1 - 0.019118}{0.9606^2} - (1 - 1.056 \times 0.019118) = 0.0840$$

As

$$\tau = \mu\frac{dv}{dy} = f\frac{\rho\overline{V}^2}{8} = w\,\frac{f}{4}\,\frac{\overline{V}^2}{2g} = wk\,\frac{\overline{V}^2}{2g}$$

therefore

$$\frac{k_d}{k_0} = \frac{\tau_d}{\tau_0} = \frac{\mu_d}{\mu_0} \qquad \left.\frac{dv}{dy}\right\}\ \text{is assumed constant as velocity is constant, and pipe factor = 1 approximately}$$

and

$$\mu_d = \mu_0 \frac{k_d}{k_0} = 0.0000316 \frac{0.0715}{0.0840} = 0.0000269$$

The corresponding temperature is 51 F, i.e., the inside wall of the flow nozzle would have to be 10 F above the water temperature. As the velocity at the wall is zero, this must be the water temperature at the wall. It does not seem that this is impossible with a temperature gradient of 67.5 — 41 or 26.5 F.

Similar computations for other velocities are given in Table 1.

That is, the difference in temperature between the water and the inside wall is inversely as the velocity, which seems quite reasonable.

The author desires to thank those who participated in either the written or oral discussion. Many valuable suggestions were made as to how to proceed, and a tremendous amount of additional experimental work was outlined for the author, which he regrets he will be unable to do. He is not particularly concerned with the physicist's explanation of the fact, but is tremendously concerned with the fact and how to get around it.

TABLE 1

Throat velocity, fps	Wall temperature, F	Wall temperature —41 F
1	52	11
2	51	10
3	50	9
4	49.1	8.1
5	48.1	7.1
6	47	6
8	44.8	3.8
10	42.7	1.7

Significance of Coal-Ash Fusing Temperature in the Light of Recent Furnace Studies

By E. G. Bailey,[1] New York, N. Y., and F. G. Ely,[2] New York, N. Y.

This paper reviews recent data relating to the fusing temperature of coal ash as well as of ash and slag from pulverized-coal-fired furnaces of both slag-tap and dry-ash-removal types. The highly oxidizing conditions in parts of both types of furnaces and the effect upon the slag-and-ash formation cause questions to be raised regarding the greatest usefulness of the present A.S.T.M. standard fusing-temperature method on a reducing atmosphere alone. Previous work of the Bureau of Mines is confirmed in showing that the form of the iron, whether ferrous or ferric, is of increasing importance with increased iron content. The paper also gives data showing the appreciably higher percentage of iron in the furnace ash and slag over the iron content in the coal ash burned. This increase applies to both dry-ash and slag-tap types of furnaces. More research is needed in both plant and laboratory.

The extended use of coal-ash fusing temperature in connection with the classification, evaluation, and purchase of coal, since Dr. Fieldner and his associates (1)[3] published the complete results of their extensive studies on the subject, speaks well for the thoroughness with which that work was done. From time to time, a few questions of doubt have been raised regarding the accuracy with which it represents the relative performance of different coals in different furnaces and stokers, but it has withstood the test of 22 years of most extended and of increasingly trustworthy use.

The data presented in this paper do not detract in the slightest degree from the work done by Dr. Fieldner, but rather emphasize the increasing usefulness of some parts of his work which have been overlooked, or have lain dormant until changed methods of burning coal have brought about a realization of the limitations of the abbreviated data which are so often the only determinations now made and available on a given coal. Many analyses give only the softening temperature. For stoker and fuel-bed combustion, there is general agreement that the softening temperature in a reducing atmosphere, as prescribed by the A.S.T.M. method, is most useful and the initial deformation and fluid point are of little importance in so far as the clinker formation is concerned.

With the advent of pulverized-coal-fired furnaces of the slag-tap type, there was renewed interest in the fluid temperature of the ash and much work has been done by Sherman, Nicholls, Taylor, and Reid under the direction of the Bureau of Mines (2). A portion of it was carried out in association with an A.S.M.E. Research Committee on the removal of ash as molten slag. Much data were collected from furnaces, primarily from slag as tapped and with some from fly ash. A thorough study was also made of fluxes; the latest work being concentrated on the viscosity of the molten ash from several different coals.

Both Fieldner and Nicholls recognized the importance of the form in which iron existed in ash, as affecting its fusing temperature, and the influence of the atmosphere in the laboratory furnace upon the results. The A.S.T.M. method is based upon a reducing atmosphere, hence, resulting in the iron being largely in the ferrous (FeO) form in the cone as tested. Nicholls studied slag samples and ran fusing-temperature determinations in a neutral atmosphere of nitrogen in an effort to learn what was the fusing-temperature range of the slag as it existed in the furnace. Much of these previous data will be reviewed in further detail by plotting them for comparison with later work given in this paper.

The great importance of iron in the ash and slag has become more fully realized as a result of continued field and laboratory work relating to the rate of heat absorption in furnaces and boilers, as shown in two papers by one of the authors (3). It is believed that a brief survey of this work and a comparison of it with the previous knowledge will be helpful in planning further work by others and coordinating it all through the proper agencies toward a reconsideration of the methods of fusing-temperature determination and their use in comparing coal of different kinds and the results from its combustion.

Slag Characteristics

Data relating to the characteristics of slag from different parts of a two-stage slag-tap furnace are shown in Fig. 1. The coal fired has 13.6 per cent iron (as Fe) in the ash. The ash as usually analyzed would show this as 19.4 per cent Fe_2O_3 but, since it may exist in slag in the forms of FeO and Fe_2O_3, it has been considered to be less confusing to give all percentages as Fe, designating how much is in the different oxide forms, or more simply to show the percentage of Fe which is oxidized to Fe_2O_3, the remainder being FeO, with possibly a very small amount of metallic iron, Fe.

The slag collected in the furnace at *A*, *B*, and *C*, is materially higher in iron content than is the original coal ash. This segregation is largely due to the pyrites in the coal being coarser and heavier, thereby falling into the slag bed. Also the ash particles higher in iron content are likely to be more sticky and, therefore, adhere more readily to any surface contacted. The oxidation of the iron is least in the slag tapped, that is, only 19 per cent of its iron is in the form of Fe_2O_3, and 81 per cent in the form of FeO.

It should be remembered that the iron oxides are not separate and free, but combined with the other ingredients as complicated silicates. Their properties and fluxing action, however, are affected by the form of the iron oxides entering into these silicate compounds.

The ash collected in the boiler-tube bank has about the same iron content as the ash in the original coal, and the farther on it proceeds in the gas stream the higher is the percentage of oxidation. The total travel from the burners to the tube bank is about 60 ft. The fusing-temperature ranges of the slag and ash samples, collected from different parts of the furnace, have been determined in the laboratory, both in a reducing atmosphere by the A.S.T.M.

[1] Vice-President, The Babcock & Wilcox Company; President, Bailey Meter Company. Mem. A.S.M.E.

[2] Analytical Engineer, The Babcock & Wilcox Company.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

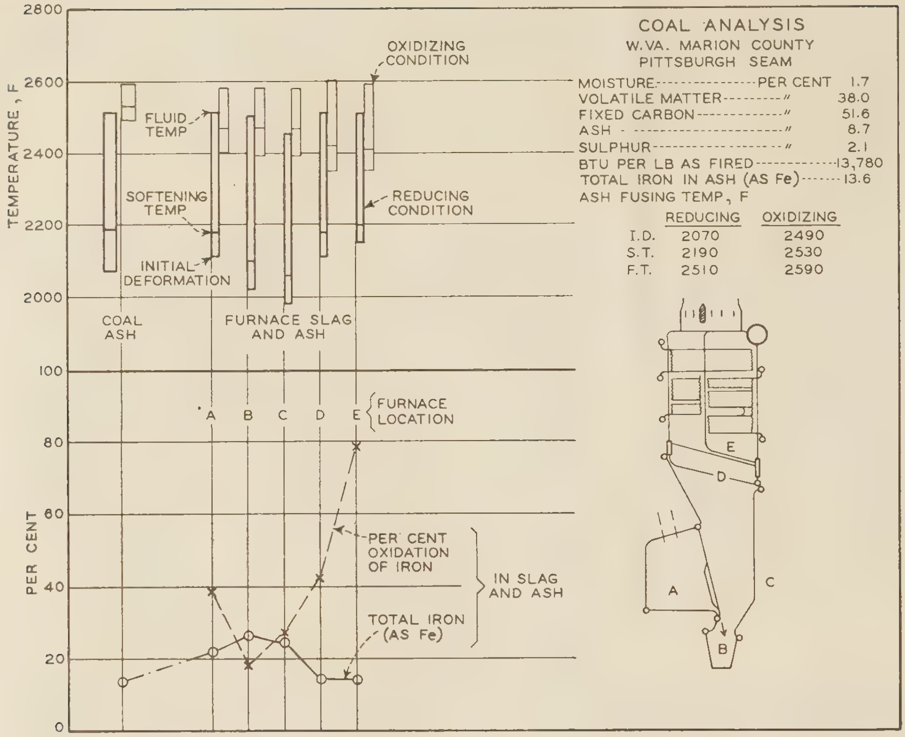

FIG. 1 CHART OF SLAG CHARACTERISTICS IN A TWO-STAGE SLAG-TAP FURNACE

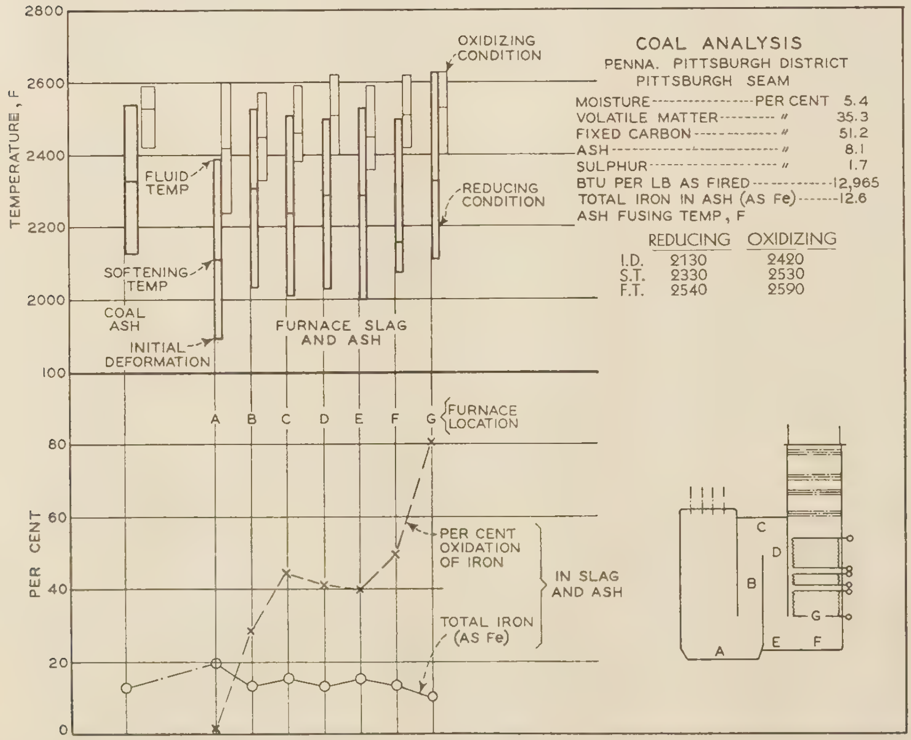

FIG. 2 CHART OF SLAG CHARACTERISTICS IN A SMALL OPEN PASS BOILER

method, and also in an oxidizing atmosphere. The temperatures on the oxidizing basis are found to be considerably higher.

Tests made in a small open-pass boiler, shown in Fig. 2, bring out similar results. This boiler has a furnace 7 ft high from the burner to the floor, and a total gas travel of only 21 ft from the burners to the superheater bank.

The slag collected in the bottom of this primary furnace shows practically all of the iron in the slag (which is 55 per cent more than the iron in the original coal ash) to be in the ferrous state.

The ash collected in the furnace at locations C, D, E, and F is only slightly higher in iron than in the coal ash, and this is 40 to 50 per cent oxidized. Both of these conditions are un-

doubtedly brought about because a considerable percentage of the ash deposited in these locations arrives in the form of unburned coke, which burns or smolders slowly. Incidentally, the rate of combustion in this furnace varied by regular cycles from 12 to 100 per cent of full rating, the latter being at the rate of 385,000 Btu per cu ft per hr for the primary furnace.

It is noted that the fluid temperature of the slag in the primary furnace is 210 F lower than the fluid temperature of the same slag if the iron were fully oxidized. This results in increasing the rate of heat absorption in the primary furnace, as its surface

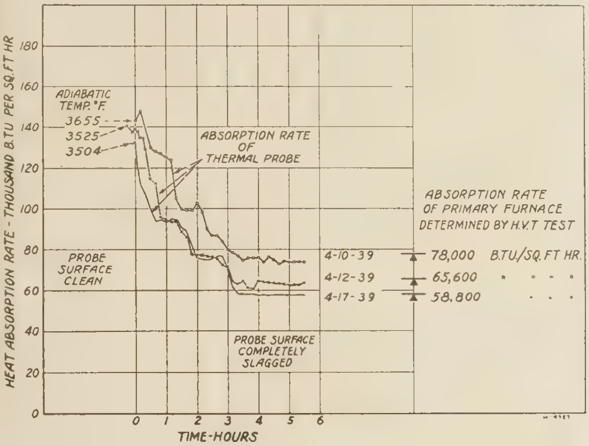

Fig. 3 Effect of Slag on Heat Absorption Rate of Thermal Probe in Primary Furnace

receiving temperature is governed by the fluid temperature of the coating of slag which is always present.

Effect of Slag on Rate of Heat Absorption

In order to learn more about the effect of slag on the rate of heat absorption in furnaces and to collect samples of slag free from contamination by any coal except that being burned when the furnace heat-absorption tests are being made, a water-cooled probe was developed. This probe is $1^1/_4$ in. diam and 5 ft long. It can be inserted into any part of any furnace.

The data plotted in Fig. 3 were obtained by means of a probe in a primary furnace, similar to that shown in Fig. 1, burning Illinois coal. They show that the probe, when first installed and kept clean by continuous scraping for a few minutes, absorbed heat at the rate of about 140,000 Btu per sq ft per hr. Within about 3 hr, the probe became completely covered with slag to a state of equilibrium, that is, it acquired a film of about $^1/_8$ inch thickness, with molten slag dripping off as fast as additional ash collected. Under these conditions the rate of heat absorption decreased to about 50 per cent of the original.

The relatively great segregation of iron in the slag collected in the furnaces previously referred to raised the question as to whether this was general in all pulverized-coal fired furnaces. Data from 15 coals mostly in different furnaces are given in Fig. 4. These are arranged to show the percentage of iron in the coal ash, and in the slag or dry ash collected from the furnaces. Also, the fusing temperature of coal ash and of slag are shown diagrammatically, and are arranged in order of decreasing softening temperature of the coal ash, as determined by the A.S.T.M. method.

Seven of the furnaces of Fig. 4 are designed for dry-ash removal, hence, they have no molten slag and little or no vitreous

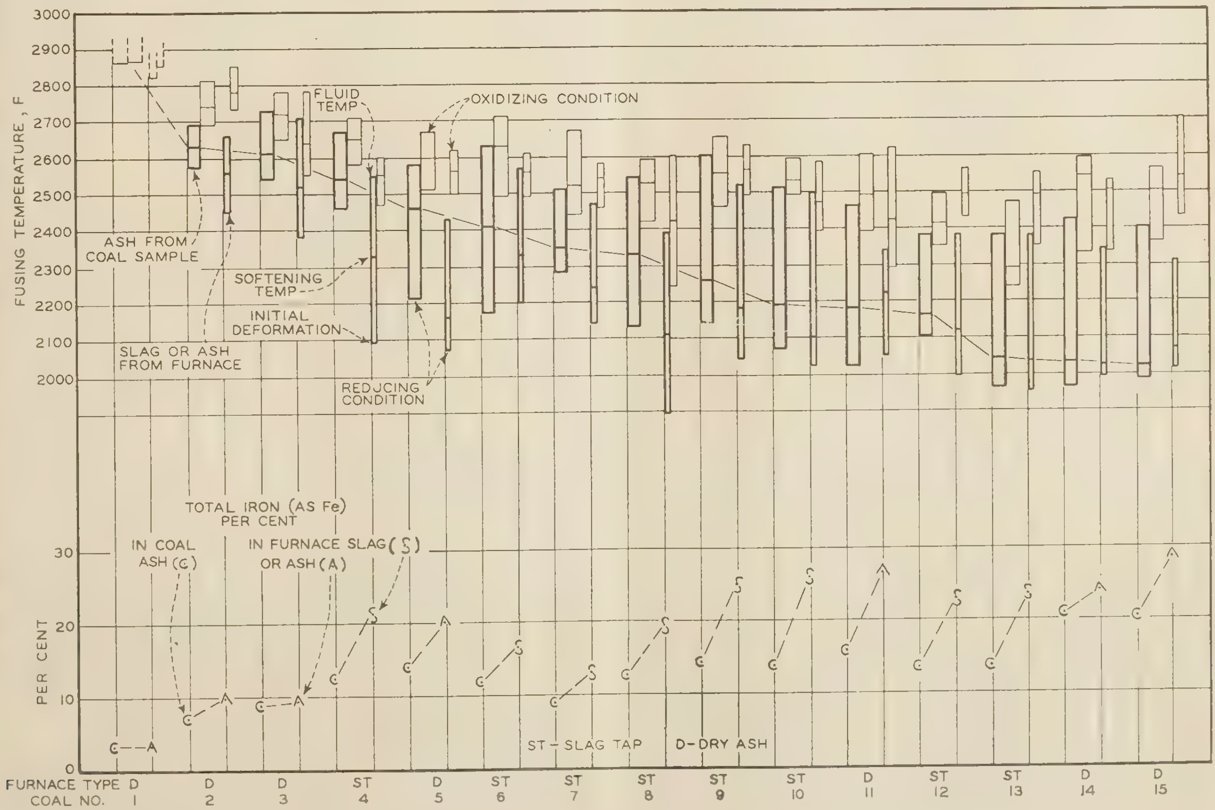

Fig. 4 Comparison of Coal Ash and Slag, Showing Fusing Temperature and Iron Content

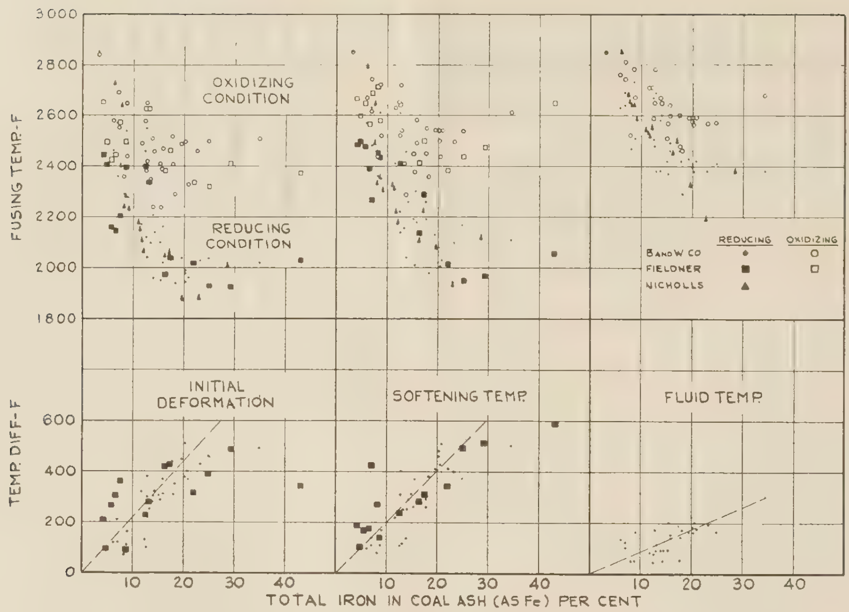

Fig. 5 Influence of Iron on Fusing Temperature of Coal Ash, as Plotted From Work by Fieldner (1), Nicholls (2), and the Authors

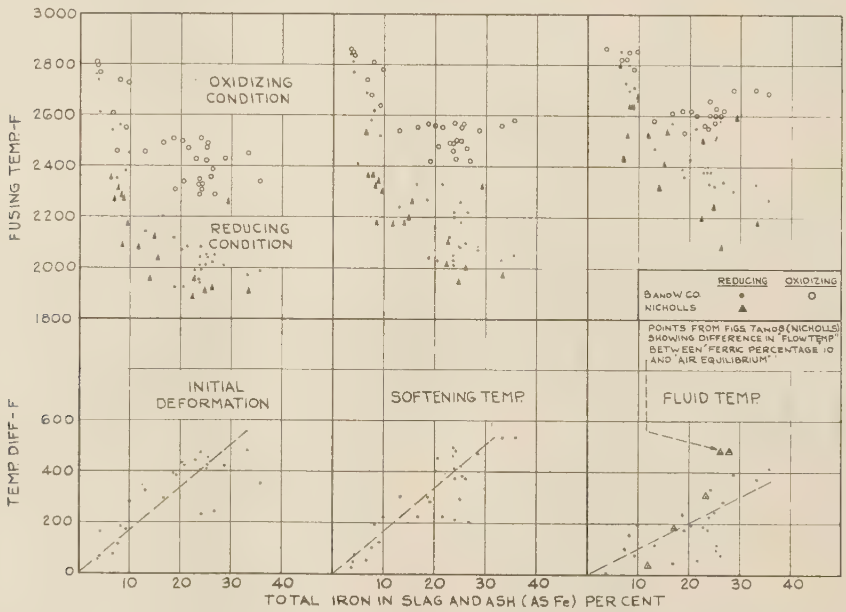

Fig. 6 Influence of Iron on Fusing Temperature of Furnace Ash and Slag as Plotted From Work by Nicholls (2), and the Authors

slag. Furnaces 1 to 3 were burning high fusing-ash coal with low percentages of iron and showed a relatively small increase in iron in the furnace ash over the coal ash. Furnaces 5, 11, 14, and 15 were burning coal having ash of lower fusing temperature and higher iron content and showed an increase in iron segregation in the furnace ash over the coal ash of the same order as that which occurs in slag-tap furnaces burning similar coal.

In passing, it should be stated that low fusing-ash coal can be burned satisfactorily in dry-ash furnaces if enough water-cooled surface is provided to keep the ash below its troublesome temperature. When the iron in the ash is largely oxidized to Fe_2O_3, the initial deformation (perhaps closely related to its initial sticky temperature) is 250 to 400 deg higher than on the A.S.T.M. or reducing-atmosphere basis. It is noted that in most cases the fusing-temperature range of the furnace ash is lower than the coal ash, when comparison is made on the reducing basis. On the oxidizing basis, there are several cases where the furnace ash has a higher fusing-temperature range than has the coal ash.

Importance of Iron in Coal Ash on Fusing Temperature

While it is fully realized that other ingredients of coal ash affect its fusing temperature, and especially the relationship be-

tween silica, alumina, lime, magnesia, and the alkalies, yet it seems that they hold a second place in importance to iron. Dr. Fieldner's early work (1) gives analyses of coal from many parts of the world. It is noted that those in which we are most interested, namely, the Appalachian and Mid-Continent fields, are of a greater similarity of ash composition than are lignite and some other coals. As the coals in which we have been working come largely from the Appalachian and Mid-Continental fields, we have restricted the plotting of points in Figs. 5 and 6 to these areas.

The data plotted in Fig. 5 include representative data from Dr. Fieldner's early work (1), coal samples covered in Nicholls' references (2), as well as a large number of tests made in connection with the authors' work. In order to simplify the plotting of data on this basis of comparison, it was thought best to mark the points by the three sources of information rather than to differentiate between the individual coals. Anyone who is interested further can take the data from Dr. Fieldner's and Mr. Nicholls' work and plot the individual samples, where complete analyses of ash and slag are available, and complete data from the authors' work will be supplied to those who are specifically interested.

It is clearly brought out that the fusing-temperature range, that is, initial deformation, softening, and fluid points, by the A.S.T.M. method in a reducing atmosphere, all vary decidedly with the percentage of iron, variations from the average, probably being due to ratios of silica, alumina, lime, alkalies, etc. It is also noted that the oxidizing basis brings all data more nearly to a uniform range, that is, there is not so much spread between the initial deformation and the fluid temperature. However, in each group, the difference between the reducing and oxidizing atmosphere appears to be a distinct function of the iron content, and approaches 500 F for about 25 per cent iron, while on the fluid basis the difference is about one half.

Similar data from the furnace ash and slag from the same coals of Fig. 5 are shown in Fig. 6, in so far as they are available. Dr. Fieldner did not have data on fluid temperature of coal ash nor any data on slag corresponding to the coal ash. Nicholls gave data on coal ash and slag from the same coal, but only on a reducing basis.

The percentage of iron in the furnace ash and slag is higher than is the iron in the laboratory-prepared ash from the same coal, as illustrated in Fig. 4, and as noted from the data given in Nicholls' work. However, Fig. 6 may show more points in a given group of iron content than does Fig. 5, because more tests have been made on the furnace ash and slag than on the coal.

Most of the ash or slag with the lowest iron content comes from southern West Virginia, where the basic ash is largely fire clay, or in approximately fire-clay proportions of silica and alumina, with a small amount of iron and lime. If the iron were entirely eliminated their fusing temperature would exceed 3000 F. The data given are limited by the furnace used to 2850 F, so that actual fluid points are undoubtedly higher than those shown.

Mr. Nicholls has done some thorough work relating to the variation in flow temperature of slag with different ferric percentages, as shown in Fig. 7. This is reproduced from Fig. 1 of his paper (4). The State Line slag has a silica-alumina ratio of 2.51, 37.5 equivalent Fe_2O_3 or 26.3 Fe, and 8.4 CaO + MgO. The "ferric percentage" equals

$$\frac{Fe_2O_3}{Fe_2O_3 + 1.11\ FeO + 1.43\ Fe} \times 100$$

In Figs. 1 and 2 this same value is called "per cent oxidation of iron." It is obvious that the same percentage will be obtained if the Fe in Fe_2O_3 is divided by the total iron in the ash.

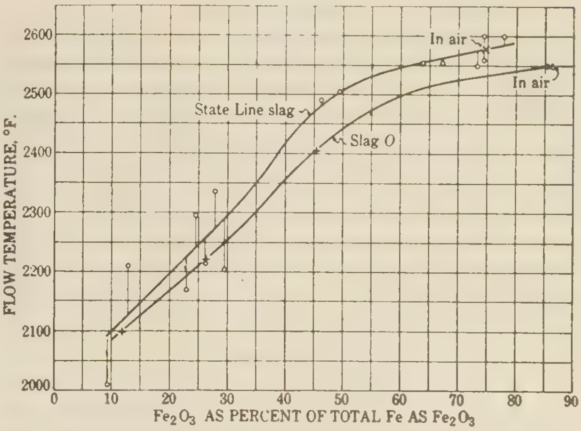

Fig. 7 Effect of Change in Forms of Iron on the Flow Temperatures of Two Slags

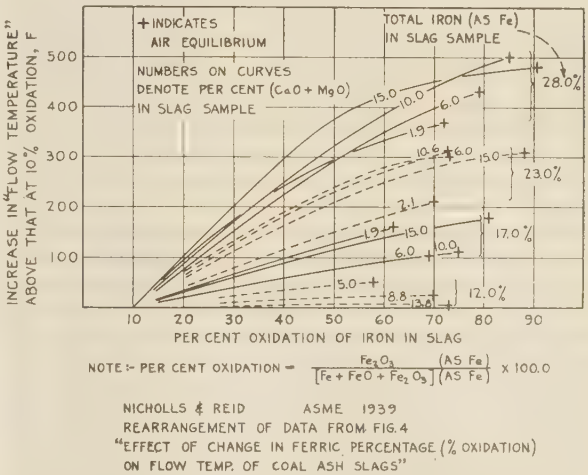

Fig. 8 Effect of Change in Ferric Percentage on Flow Temperature of Coal-Ash Slags Replotted From Data by Nicholls (5)

Mr. Nicholls does not state whether he was able to recover from all slag samples all of the iron in its existing form as either Fe, FeO, or Fe_2O_3. We have sometimes found it difficult to get all of the ferrous and ferric iron into solution without causing oxidation as in the standard nitric-acid method. Therefore, we have taken the "per cent oxidation" to be the Fe in Fe_2O_3 divided by the total iron actually recovered in Fe, FeO, and Fe_2O_3.

It should also be pointed out that Mr. Nicholls is using "flow temperature" which is not the same as the fluid temperature called for in the A.S.T.M. method. Mr. Nicholls also gives the highest ferric percentages to be 70 to 80 per cent in air. Perhaps the slag was so vitreous that the penetration of oxygen could not be complete. We have found ash from boiler- and superheater-tube banks more than 90 per cent oxidized.

We have made some fusing-temperature determinations on slag which had been pulverized and reformed into cones, and compared them with cones shaped directly from the vitreous slag in one piece. As a rule, the latter have a higher fusing-temperature range and less spread between the initial deformation and the fluid point than when pulverized and reformed.

Further work by Nicholls (5) gives in his Fig. 4 the effect of "ferric percentage" on flow temperature of coal ash. The authors have taken the liberty of replotting these data in Fig. 8, grouping

them according to the iron content of the slag, and labeling the different curves with the percentage of CaO + MgO. The end points, or those of maximum oxidation, are taken as differences from the 10 per cent ferric condition, and are shown in Fig. 6 for comparison with the other work. Also the test by Nicholls of State Line slag, previously referred to, is similarly shown in Fig. 6. It is noted that Nicholls' data give a steeper curve regarding the effect of percentage of iron on spread between the fluid temperature in oxidizing and reducing condition.

These differences may be due to the difference in the coal with which we were dealing; difference in the method of determining the fluid temperature versus flow temperature; or how actively reducing and how actively oxidizing the furnace atmospheres may have been in the two cases.

Observations From Furnace Operation

The effect of the different forms of iron in coal ash and slag is sufficient in explaining many phenomena which may have been attributed to other less accurately defined causes.

Flame impingement produces active cutting of refractory or extra high rates of heat absorption in water-cooled surfaces. In addition to the higher temperature due to the proximity of the turbulent burning fuel, the ferrous condition of the slag fluxes refractory more rapidly and keeps slag more fluid, thereby resulting in a thinner protective coating on water-cooled surfaces.

Slag can be tapped more easily when operating with low excess air, which not only produces a high adiabatic temperature, but causes a more fluid slag because of the ferrous condition of the iron.

Higher rates of heat absorption by waterwalls in slag-tap furnaces result from a combination of the preceding phenomena.

Pyrites in free pieces when properly pulverized and burned will result in the formation of Fe_2O_3, Fe_3O_4, or FeO, depending upon the degree of oxidizing atmosphere present. But, if the particles are too coarse for partial or complete oxidation under the burning conditions prevailing, FeS may be formed, a very troublesome compound which may collect as a liquid beneath the slag bed, often cutting through floors, tubes, etc. This also is likely to be the cause of cutting metal away from tube faces above slag level under certain conditions of active flame impingement.

Excess carbon, resulting from coarse unburned coal on slag surfaces may reduce any form of iron oxide to metallic iron. Such iron settles to the bottom of a molten slag-pool and forms salamanders which are difficult to remove either hot or cold. If sufficient metallic iron were removed from the slag by such precipitation, the fusing temperature of the remaining slag would be raised, due to the lower FeO or Fe_2O_3 content.

When pulverized coal burns completely, the residual ash from some particles may be quite different from the ash of other particles. Some have practically no iron present while others may have varying percentages of iron, up to practically 100 per cent iron resulting from the combustion of pyrites. As boiler furnaces are normally operated with some excess air, the tendency is for the iron to become fully oxidized to Fe_2O_3 as it passes on with the gases.

Some coke or incompletely burned coal is always present in fly ash. When it strikes walls or tubes some adheres and the carbon then burns quite rapidly, leaving the ash adhering to and combining with other ash or slag already there. Iron in this ash, liberated in the presence of burning carbon, is likely to be partially or largely in a ferrous (FeO) state. This bears out the general observation that coarse pulverization or stratification from poor burners results in more slag in boiler- or superheater-tube banks than is experienced when operating with fine pulverization and good burners.

Practically all of the ash carried from fuel beds by the gases of combustion is in the form of coke particles and the resultant action of burning out and depositing a troublesome slag takes place, similar to that described in the preceding paragraph.

For dry-ash furnaces the property of coal ash, which it seems best to use as a guide in furnace design and adaptation of coal, is the initial deformation on the oxidizing basis. With proper pulverization, and burners in good adjustment, the ash reaching the wall, and especially the tube bank or superheater, should be highly oxidized. As it is desired to operate such a furnace without sticky or vitreous slag, the gases should approximate temperatures at or below that corresponding to the initial deformation. It is recognized that coals having ash with high initial deformation such as 2800+, can be burned in a given furnace at higher rates of combustion and with higher gas temperatures, entering tube banks with less slag trouble than can be obtained from coals having ash with lower initial deformation temperature. The individual furnace and the arrangement of gas flow has a great deal to do with the limits which can be satisfactorily used. Where gases are flowing parallel to surfaces there is much less adherence of a given ash at a given temperature than where the direction of flow causes direct impact against the wall or against tubes. Therefore, for the present, it seems to be less important to learn just what is the sticky point than it is to obtain data on some already recognized basis, such as initial deformation, between the extreme conditions produced in an oxidizing atmosphere, as distinct from the reducing atmosphere.

For dry-ash furnaces, the softening temperature is not of particular importance, except that a knowledge of it together with the fluid temperature are worth-while guides to note where local trouble from flame impingement might cause the formation of vitreous slag, even though the furnace proper and the tube bank were free from any slagging trouble. By making a furnace large enough, or at least keeping the rate of combustion low with respect to the surface exposed, dry ash can be obtained with any coal such as No. 11 in Fig 4.

For slag-tap furnaces, the fluid temperature in a reducing atmosphere is probably the most important characteristic with relation to the tapping of slag and the range of rating over which the boiler can be operated and have slag tap satisfactorily. The next important point is behavior of ash in the tube banks where slag might adhere, and here the initial deformation on the oxidizing basis is again a significant index. The other information, i.e., softening temperature and fluid temperature, may be useful at least until performances become better standardized with the increased knowledge of ash and slag.

For fuel-bed and stoker operation, the softening temperature on the A.S.T.M. method with a reducing atmosphere is undoubtedly the best criterion. Clinker formed in the fuel bed is in a reducing atmosphere, and the entire elimination of clinker is impossible except for the very highest softening temperature ash, and it must in any event be kept cooler than the fluid point, or else a most serious form of trouble will result. As to the indications for trouble in the tube bank, that is not so easy where a large percentage of all the ash arrives at that point in the form of coke, which burns and leaves the ash largely reduced so it might seem that the initial deformation on the A.S.T.M. method would be some criterion to bird nesting or slagging in tube banks.

Summary

Fusing-temperature determinations of coal ash have served a useful purpose over a period of more than 20 years, and have brought about a reasonably accurate basis for comparisons between different coals, as to their relative clinkering properties.

The A.S.T.M. method, which determines the initial deformation, softening temperature, and fluid temperature in a reducing atmosphere is best adapted to fuel-bed and stoker conditions

where the ash forms into clinker in a reducing atmosphere. The softening temperature is the most important phase of the range.

With the extended use of pulverized coal, it is found that other properties of the ash are important in addition to those already recognized by the A.S.T.M. standard. The additional information desired is the fusing-temperature range in an oxidizing atmosphere, and also the determination of the percentage of iron in the ash.

There has been a tendency to abbreviate the A.S.T.M. standard data, and much published data relating to coal are restricted to the softening temperature alone. Some recent publications (6) have given only the initial deformation and softening temperature of the ash in the coal itself. Mr. Gould's paper (6) included a series of very valuable experiments which show the segregation of ash but limited the determination to initial deformation alone, which, owing to the special nature of this work, was justified. In discussing this paper, its authors expressed a great doubt as to the accuracy with which fluid temperature could be determined in different laboratories.

Mr. Nicholls' work has injected the term "flow temperature" and he has also investigated most thoroughly the matter of slag viscosity, from which work he has brought out most useful data.

It would seem that the time has arrived for reconsideration and further investigation, and perhaps new standards, with respect to the fusing temperature of coal ash and slag.

As to whether or not initial deformation holds a close relation to stickiness is uncertain. There is evidence from the collection of the first coating of dust on the water-cooled probe that the foundation of all slag is a dust of very highly segregated iron content.

Even though the characteristics of coal ash are determined by the A.S.T.M. method, plus the fusing-temperature range in an oxidizing atmosphere, and the percentage of iron, it is doubtful if that information will give all that is necessary to learn what one should know of the coal characteristics and the adaptation of this fuel to certain equipment. The high degree of segregation of the iron in the furnace ash and slag causes much doubt to be cast on any determination from the coal ash alone. Perhaps a segregation factor might be worked out for different types of furnaces and rates of combustion. The surest way is the old way of actually testing a given coal in the plant where it is to be burned and learn its characteristics and the results from over-all performance. To be of value regarding the cost of operation, slag removal, etc., such a test should be of at least a week's duration. When such tests are made, complete data not only of the coal ash but of the furnace ash and slag should be carefully investigated so that a basis of comparison can be formulated by coordinating the practical field tests with the laboratory tests.

Active work is still progressing along the line of furnace and boiler design wherein the variations in the characteristics of coal and its ash are of diminishing importance. The best results seem to be taking two seemingly divergent lines: (*a*) the slag-tap furnace of high duty, wherein a wide range of coal can be burned; (*b*) the dry-ash furnace which has been best adapted to coal having ash of high fusing temperature, but when built with ample water-cooled surfaces, is suitable to burn the lowest grade of coal with a minimum of slag or ash difficulties. To do this requires a more expensive unit, and one in which variations in performance between pulverized coal, oil, and gas are greatest, and sometimes undesirable.

For the present the evaluation of coal for pulverized firing on a basis of its fusing temperature range by A.S.T.M. method should be carefully checked with individual furnace type and rate of operation before being sure of its true value.

BIBLIOGRAPHY

1 "The Fusibility of Coal Ash and the Determination of the Softening Temperature," by A. C. Fieldner, A. E. Hall, and A. L. Field, U. S. Bureau of Mines, Bulletin 129, 1918, 146 pp.

2 "Study of Some Factors in Removal of Ash as Molten Slag From Powdered-Coal Furnaces," by Ralph A. Sherman, P. Nicholls, and Edmund Taylor, Trans. A.S.M.E., vol. 51, 1929, paper FSP-51-51, pp. 399–413.

"Fluxing of Ashes and Slags as Related to the Slagging Type Furnace," by P. Nicholls and W. T. Reid, Trans. A.S.M.E., vol. 54, paper RP-54-9, 1932, pp. 167–190.

"Slags From Slag-Tap Furnaces and Their Properties," by P. Nicholls and W. T. Reid, Trans. A.S.M.E., vol. 56, 1934, paper RP-56-3, pp. 447–465.

"Viscosity of Coal-Ash Slags," by P. Nicholls and W. T. Reid, Trans. A.S.M.E., February, 1940, pp. 141–153.

3 "Modern Boiler Furnaces," by E. G. Bailey, Trans. A.S.M.E., vol. 61, 1939, pp. 561–569; discussion, pp. 569–576.

"Steam-Boiler Performance and a Method of Comparison," by E. G. Bailey, Trans. A.S.M.E., July, 1940, pp. 367–378.

4 "Slags From Slag-Tap Furnaces and Their Properties," by P. Nicholls and W. T. Reid, Trans. A.S.M.E., vol. 56, 1934, paper RP-56-3, pp. 447–465.

5 "Viscosity of Coal-Ash Slags," by P. Nicholls and W. T. Reid, Trans. A.S.M.E., February, 1940, pp. 141–153.

6 "Proportions of Free Fusible Material in Coal Ash, as an Index of Clinker and Slag Formation," by G. B. Gould and H. L. Brunjes, Trans. American Institute of Mining and Metallurgical Engineers, vol. 139, Coal Division, 1940.

Discussion

T. G. Estep.[4] The authors of this paper are to be highly commended for the experimental work and the analysis made of it as set forth in this paper. It is the writer's belief that the only difficulties encountered in burning coal in pulverized form are from the ash. Any studies increasing our knowledge of the behavior of this constituent during the combustion process will make it possible to alter the design of furnaces to eliminate or reduce the troubles encountered and thus widen the range of coals which can be used satisfactorily in pulverized form.

The concentration of iron in the slag at the bottom of the furnace is no doubt due to the reasons given in the paper, that is, greater weight of the particles high in iron; but the fact that the iron in this slag is largely in the ferrous state may not be due entirely to lack of oxygen in the furnace. If the steam-generating unit has a primary and a secondary furnace, the atmosphere in the primary furnace will be deficient in oxygen but, if only one furnace is used, there will be excess oxygen present. In either case, there will be other reducing agents in the furnace atmosphere, such as carbon, hydrogen, and carbon monoxide. Since the iron in the ash originally is not in the form of oxides but combined with sulphur or silicon, it must first be separated from the elements in the natural ash and then oxidized. This oxidation must be carried out in an atmosphere which has very little oxygen and in the presence of strongly reducing agents, also in a very short time. The time element is very important because the rate of oxidation will depend upon the amount of surface exposed for absorption of oxygen as well as the temperature. It is not difficult to understand why the liquid particles of ash reach the slag bed with the iron in a metallic or ferrous state. In the slag bed itself, there is no opportunity for further oxidation because the carbon particles reaching the surface of this bed will use what oxygen is available.

In support of the writer's belief that time is an important factor in the oxidation of iron, a paper by M. A. Mayers[5] shows

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[5] "Flow Processes in Underfeed Stokers," by M. A. Mayers, Trans. A.S.M.E., vol. 63, 1941, pp. 479–489.

that, in fuel beds on underfeed stokers, there is some excess oxygen in the presence of combustible gases at high temperatures. Just how much of this free oxygen is due to a "blow through" of the air is not known but, if the combustion cannot be completed in the short time available, it is reasonable to assume that the oxidation of the iron cannot be completed in the same time.

Even if some of the iron should be oxidized to the ferric state, it is possible that dissociation may take place at the furnace temperature.

Nicholls and Reid[6] have shown that dissociation begins at about 2200 F and is very pronounced at 2400 F. A careful search of the literature does not reveal much more information on this subject of dissociation than that given in the reference,[6] particularly as to the time element and the effect of the atmosphere in which heating takes place. Dissociation should depend upon temperature only but it does require time for equilibrium to be established. It is doubtful if any pronounced dissociation of ferric oxide to ferrous oxide and free oxygen takes place in a boiler furnace.

It is the writer's understanding that the iron in the ash and slag from the different parts of the furnaces was determined from the material collected on the probe and that the probe was made from iron. Is it possible that some of the iron in the material so collected may have come from the probe? Since the iron in the ash or slag is small, a very small amount of iron from the probe would change the percentage materially. Since iron will oxidize in the presence of both oxygen and carbon dioxide, it might be well to use a nonferrous probe for collecting the samples.

In securing the information for plotting the curves of heat absorption by the water-cooled probe, was care taken to have the same water velocity through the probe for all points? While it is realized that these curves are intended to show, only in a general way, the effect of slag accumulation on the rate of heat transfer, the velocity of the water is a very important factor in this measurement.

A. C. Fieldner.[7] Mr. Bailey has a compelling way of focusing attention on the need of new and better methods for evaluating certain physical and chemical properties of fuels which have become important because of new types of equipment or new developments in utilization. He does this by formulating a scientific analysis of the fundamental principles involved, as in his pioneering work on the sampling of coal, or in adapting a simple test method and showing its relation to the performance of the fuel, as in his cone-fusion test and its relation to clinker formation. In the paper under discussion,[8] the authors present important evidence indicating the need for an amplification of the present standard laboratory tests for the evaluation of the fusing and clinkering properties of coal ash.

The softening temperature as determined by the present standard A.S.T.M. method has been generally accepted as a rough indication of the probable clinkering characteristics of coal ash in fuel beds where reducing conditions prevail. However, it has been recognized by fuel technologists that the softening temperature as indicated by the down point of an ash cone is only a single point in a range of temperatures, during which the ash or different constituents of the ash begin to soften and ultimately become a more or less fluid slag. The range of this melting process and the viscosity of the ash slag undoubtedly has an important bearing on clinker formation and on conditions produced in fuel-burning equipment. For this reason, the standard A.S.T.M. method includes also a procedure for observing and recording the temperatures of initial deformation of the ash cone and the temperature at which the cone has spread out over the base in a flat layer. This point is known as the "fluid temperature." These additional critical points are shown by the authors of this paper to be significant in the study of special ash problems, such as those pertaining to slag-tap furnaces and the rate of transmission of heat through slag-coated tubes.

Nicholls and his co-workers at the Bureau of Mines also have appreciated the urgent need of more knowledge of the relation between the composition of ash and its slagging properties than is afforded by the initial deformation, softening, and fluid temperatures, as given in the standard A.S.T.M. test. The authors in the Bibliography of the paper refer (2) to their work on flow temperatures and viscosity of ash slags at various temperatures. Such data are of fundamental importance in relation to slag-tap furnaces and the adherence of layers of slag on heated surfaces. They have made considerable progress but much more remains to be done. They have developed apparatus and procedures for studying the absolute viscosity of ash slags in relation to the temperature, composition of the ash, and the oxidizing or reducing condition of the atmosphere in which the slag is being formed. However, it must be kept in mind that this is no short-term investigation. It will require experimentation over a period of several years before enough data can be accumulated on which reliable predictions can be made on the basis of simple tests which may be standardized in a manner similar to our present methods.

The authors have raised the important question of determining ash-fusion temperatures in oxidizing atmospheres in order to obtain data on the extent of slag formation on tube banks or superheaters. This problem also requires extended investigation on the correlation of such tests with performance in boiler equipment. If further studies indicate that fusion tests should also be made in oxidizing atmospheres, then a standard test can readily be developed for this purpose by Committee D-5 of the American Society for Testing Materials. The writer feels sure that this committee will be glad to cooperate in extending standard tests to furnish the increasing amount of data which is required by new developments in coal-burning equipment.

The writer is in agreement with the authors of this paper in their plea for further investigation, and perhaps new or additional standards, with respect to the fusing temperature of coal ash and slag. The Bureau of Mines has been interested in this field of study for many years and expects to continue this work to the fullest extent of its facilities. The coal-ash viscosimeter, which Nicholls and Reid have developed, has provided a new tool which permits the accurate measurement of the flowing properties of ash slags under various conditions of heating. Such measurements go to the heart of the problem. The data which they are obtaining will afford fundamental information on the mechanism of the clinker- and slag-forming process in fuel beds, in slag-tap furnaces, and on the surfaces of the furnace and boiler equipment. It is hoped that they will be able to continue this work over a period of years and that practical operating engineers will assist in applying and correlating their work with practical operation in boiler furnaces.

R. S. Julsrud.[9] It has been recognized for some time that the A.S.T.M. method of determining ash-fusion temperature has not proved a wholly reliable guide in predicting the performance of coals under actual firing conditions. The reasons shown for this discrepancy have been (*a*) that ash in an oxidizing atmosphere fuses at a higher temperature than in a reducing atmosphere, such as employed in the A.S.T.M. method, since some

[6] "Viscosity of Coal-Ash Slags," by P. Nicholls and W. T. Reid, Trans. A.S.M.E., vol. 63, Feb., 1940, pp. 141–153.

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[8] Discussion of this paper is published by permission of the Director, Bureau of Mines, United States Department of the Interior.

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constituents of the ash, such as oxide of iron, are reduced to oxides with lower melting points, and (*b*) the A.S.T.M. method of sampling and analysis represents the temperature at which a thorough mixture of the whole ash of the coal fuses, the mixing having taken place before the analysis is made. In actual practice, neither the entire furnace is in a reducing atmosphere nor is the entire fuel supply thoroughly mixed upon entering the furnace.

The investigation of Gould and Brunjes[10] had for its purpose the determination of softening temperatures of the ash in physically separable portions of coal and the determination of the relative quantities of ash contributed by each portion. To this end samples of five different coals were pulverized to pass a 200-mesh screen and then separated into gravity fractions from 1.3 to 1.9. It is known that reducing coal frees the impurities, this being a method employed in coal preparation. These separations indicated that, in the main, the ash in these coals is composed of highly refractory material (2800 F) and easily fusible material (under 2300 F). It further showed that these fractions are not only physically separable, but are separated in the reduction process. The proportions of easily fusible material varied from 12.6 to 31.4 per cent of the total ash in the coal.

Applying the work of Gould and Brunjes[10] to the authors' studies of slag characteristics, as given in Figs. 1 and 2 of the paper, one can visualize that the easily fusible material quickly melts upon entering the furnace, and upon striking a tube or the furnace floor, in a sticky condition, adheres to it. The refractory ash, in union with burning-coal particles or alone, passes through the furnace to deposit itself upon generating or superheater tubes or other heating surfaces in varying forms of sticky, spongy, or granular ash, depending upon whether or not it combines with ash of different fusing temperature. The greater concentration of iron in the furnaces of Figs. 1 and 2 of the paper may in part be accounted for by the presence of a reducing atmosphere therein and also[11] to the greater density of the low-fusion-ash particles, causing them, in accordance with Stokes' law, to be more quickly precipitated to the floor. The original form in which the iron in the ash existed such as FeS_2, Fe_2SO_4 is likely reduced to Fe by combustion of the S_2 and in its passage through the furnace picks up O_2 to form FeO, Fe_2O_3, and Fe_3O_4.

The work of Gould and Brunjes,[10] therefore, complements the authors' studies and helps to explain the varying forms and constituents of ash, clinker, and slag found in pulverized-coal-fired furnaces. In this and previous similar investigations, the authors have given a clear picture of the results of coal ash in the furnaces studied, whereas, Messrs. Gould and Brunjes devoted their studies to causes creating these results. They merit the thanks of those interested in coal utilization and steam generation.

P. Nicholls[12] and W. T. Reid.[13] This paper[14] is timely and helpful because it again draws attention to our neglect of opportunities in that we have made little attempt to use the initial and fluid cone-fusion temperatures, but have optimistically hoped that all ash behaviors and all troubles could be explained as being proportional to the softening temperature. That such optimism was not justified has been proved by many failures of coordination between the softening temperatures and relative clinkering or slagging troubles.

In our attempts at coordination there has been equal fault in failing to analyze the dependence of ash troubles on the conditions to which that ash has been subjected, or, as we have expressed it, a consideration of the life history of the ash in terms of path of travel, temperatures, and time. For a number of years our reports have urged that correct interpretation of clinkering and slagging requires much more analysis of conditions than has been customary.

The latter part of the paper suggests uses that could be made of the three cone-fusion temperatures in connection with different types of furnaces or methods of burning; such extended use of the cone fusions would undoubtedly be an advance, and engineers should make a practice of asking for the three fusion temperatures when cone tests are made.

However, even with the values for the three cone-fusion temperatures, there will always be a limitation to the interpretation and analysis of observations of clinkering and slagging, and also of the ability to predict the suitability of a coal, unless and until we have data on the viscosity of slags. The ease with which a molten slag will run or the rate at which it will flow are evidently very important in the slagging of tubes or walls in all types of furnaces, in fuel beds as affecting the density of the clinkers and the clogging of grate bars, and in slag-tap furnaces as affecting the ability to tap and the time required.

The authors bring out a number of interesting points that invite discussion. First it is suggested that the paper does not place enough importance on the lime content of ashes and slags; its effects in many respects equal those of iron oxides. Lime becomes of more importance as the iron content is low, or as the ferric percentage is high, i.e., under oxidizing conditions. Plots, such as those of Figs. 5 and 6 of the paper, will give scattering of the points if the lime contents vary.

The segregation of iron in the slag bed or primary furnace, shown in Fig. 4, averages higher than in the eighteen furnaces we reported on in 1934 (Table 2, Bibliography (4) of the authors). For the slag-tap furnaces of Fig. 4, the increase in iron, expressed as percentage of iron in the coal ash, averages 70, with a maximum of 90, whereas, ours averaged 11 per cent with a maximum of 32. Presumably this is due to higher rates of burning in the primary furnace in later designs.

Tables 1 and 2 of the same paper (4) show that there was segregation not only of the iron but also of the lime; in addition, there was increase of the silica-alumina ratio. The lime was increased in twelve and the silica-alumina ratio in sixteen out of the eighteen furnaces.

The authors show that the ferric percentage of the slags and ashes increased along the path of travel of the gases and imply that what occurs is that the iron in the ash particles is highly reduced in the primary furnace and is then reoxidized along the path of travel. We did not examine wall or tube deposits from the eighteen boilers, but eleven stations furnished samples of fly ash, which were so taken as to match the slag samples; the average ferric percentage of ten samples was 77, and the maximum 84. Our deduction was that the iron in most particles does not have time or opportunity to be reduced in their passage through the furnace. However, particles deposited on a wall will have ample time to reach equilibrium, and there will be more reduction to FeO as the particle is maintained at a higher temperature, or is in contact with carbon.

Whether reduction and then reoxidation occur could be proved by collecting fly ash in water-cooled samplers at different positions along the path of travel. However, studies we have made

[10] "Proportions of Free Fusible Material in Coal Ash as an Index to Clinker and Slag Formation," by G. B. Gould and H. L. Brunjes, American Institute of Mining and Metallurgical Engineers, Tech. Pub. No. 1175, 1940.

[11] "Fusion Characteristics of Fractionated Coal Ashes," by A. H. Moody and D. D. Langan, Jr., *Combustion*, vol. 5, Oct., 1933, pp. 13–17.

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[14] Discussion of this paper is published by permission of the Director, Bureau of Mines, United States Department of the Interior.

on oxidation and reduction of slag particles under controlled conditions support our viewpoint of 1934. Reduction of Fe_2O_3 to FeO can occur relatively rapidly, as far as the change is due to increase of temperature only, but less rapidly by the action of a reducing gas or carbon. Reoxidation is relatively slow; the absorption of oxygen must start at the surface of a particle of slag and work inward; thus, the time required to oxidize will increase rapidly with the size of the particles. We do not consider it likely that the actions of reduction and reoxidation can occur in the few seconds required for the passage of the ash through the furnace.

The foregoing and Fig. 12, of reference (5) of this paper, explain why the authors found ash having 90 ferric percentage in the rear of the furnace, whereas, in our tests the ferric percentages were lower for slags that reached equilibrium in air at higher temperatures.

The paper raises a question on the dependability of the chemical analyses for total iron and iron forms in slags. The question was referred to W. A. Selvig, under whose direction our many analyses were made; his report follows:

"No method for the determination of iron forms in materials such as slags is entirely satisfactory. All methods are empirical and subject to small errors, the extent of which it is difficult to determine exactly. The method used by the Bureau is that of the Bureau of Mines Technical Paper 8, Methods of Analyzing Coal and Coke.

"By this method total iron can be determined accurately and is calculated to Fe_2O_3.

"Metallic iron is determined by digesting the slag with mercuric-chloride solution. Ferrous iron plus metallic iron are determined by digesting a portion of slag with dilute sulphuric acid under specified conditions, whereby these two forms are obtained in solution, and this soluble iron determined. This iron calculated to FeO minus the FeO equivalent of the metallic iron represents the ferrous oxide in the slag.

"The ferric oxide in the slag is obtained by subtracting the Fe_2O_3 equivalent of the ferrous oxide and the metallic iron from the total equivalent Fe_2O_3 previously found."

The authors' definition of their term "per cent oxidation" is the same as ours for "ferric percentage," which we define as the "ratio of ferric iron to total iron, expressed as a percentage." We question whether the term used by the authors has as definite a meaning because both the ferrous and ferric states represent per cent oxidation of the iron.

The authors refer to "stickiness" several times. This is an interesting and also—we have found—a complex property. It would be a blessing if slags would not stick to boiler tubes, but independent of whether or not data will be of immediate use we should have some understanding of factors involved in sticking.

A definition is required. Stickiness is more than "wetting;" a rod dipped in water will be wetted but one would not call water sticky. A rod pressed into tar, heated so that it is just soft, will stick, but heat the tar enough, and it will be so liquid that it would no longer be sticky. Thus, stickiness involves some measure of "force to separate" and, in general, as a slag is heated there will be a range of temperature over which it could be called sticky. However, it is possible that measures of ranges of wetting may also be required.

Two surfaces are always involved in sticking; so far our studies have been limited to the sticking of slag to slag, both surfaces being at the same temperature. We have records of the initial temperature of sticking of over 400 slags, included in the report of reference (4) of the paper. Most slags had one sticky range; others would have two, that is, the stickiness disappeared as the temperature was increased and then reappeared. Others seemed to have no sticky temperature.

A more intensive study of a few slags and glasses showed that the stickiness depended upon the liquid phase present in the slag, its quantity, and its viscosity; also, the appearance of stickiness was related to the rate of heating and cooling. The initial sticky temperature of slag to slag tended to equal or be less than the cone initial temperature of the premelted ash, but there was no definite relationship to chemical composition.

Studies of the stickiness of slags to other material are included in our plans for the future. We have recognized that deposits on metal tubes may materially affect the sticking of slags. Condensation of alkalies from their vapors can be one form of deposit and, under special fuel-bed conditions, silica may be deposited as the result of oxidation of silicon in the gaseous phase.

We have studied high-iron black deposits, such as the author found on the probe. One sample had 53 per cent equivalent Fe_2O_3, twice the CaO of the coal ash, and a somewhat higher silica-alumina ratio; the ferric percentage was 78. A microscopic examination showed that it was composed of particles of fly ash, lightly fused together. The conclusion was that those particles having high iron and lime stuck to the tube more readily because they were stickier than the refractory particles.

The intensive studies on pulverized-coal furnaces which Mr. Bailey has organized should be extended to other types of furnaces, to producers, and to kilns, in the attempt to correlate the life history and forms of the ash more definitely with its composition and properties. A few complete studies should give patterns for reference in each class of burning. The probe and other innovations in methods of tests devised by the authors will be valuable tools in such investigations.

However, before such studies are undertaken there must be more complete knowledge of the absolute properties of the ash than is given by even the three cone-fusion temperatures. The cone values are at best related to arbitrarily fixed conditions of test, and vary with changes in the conditions. Premelting the ash and making up a cone can change the initial deformation temperature as much as 300 F from that of the original ash. Which is the correct value to use? Ashes have definite physical properties, such as viscosity and surface tension, which are exactly, or very closely, defined by the chemical composition. To have definite meaning, the cone temperatures must be comparative measures of the physical properties; therefore, it is necessary to determine whether such relations exist and can be used, or whether the values for the primary physical properties should supplement the more easily obtained cone-fusion values.

E. B. Powell.[15] The authors devote the greater part of their comment to the influence of atmosphere on the properties of coal ash as deposited in different parts of the furnace and on heat-absorbing surfaces. In this they point out the effect of the combustion stage of the individual particle in determining the surrounding atmosphere and the importance of fineness of pulverization as a factor in determining the combustion stage of the particle in any part of the furnace or path of the combustion products. These observations are of great value. Apparently, however, the primary purpose of the paper is to urge further study of the ash-fusion determination and the inclusion in the determination of the effect of an oxidizing atmosphere. The writer is in hearty agreement with the authors in this plea. He would add, however, that in the further study special attention should be given to definition of the atmosphere. The authors themselves suggest as a possible explanation of differences obtained in determinations made in their own and the Bureau of Mines laboratories, differences in degree of reducing and oxidizing properties of the respective furnace atmospheres. A further

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illustration of the relative indefiniteness of the expressions "reducing" and "oxidizing," as applying to laboratory-furnace atmospheres, is given in Table 1 of this discussion, covering data secured, for purposes of comparison, a few months ago on ash from one of the Pennsylvania bituminous coals.

TABLE 1 FUSION TEMPERATURES OF COAL ASH AS AFFECTED BY ATMOSPHERE, IN DEG F

	A	B	C	D
Initial deformation	1925	1990	2245	1980
Softening	2640	2790	2375	2320
Fluid	3030	2905	2540	2490

NOTE: The furnace atmospheres are designated as follows: *A*, carbon monoxide; *B*, carbon dioxide, largely decomposed to CO and O_2; *C*, products of coke combustion in air, slightly reducing; *D*, air.

Atmosphere *C*, while not in exact accord with the A.S.T.M. specification, was probably in rather close accord with that usually obtained under the specification. It will be noted, however, that, except in the temperature for initial deformation, the values secured in the two atmospheres *C* and *D*, are not radically different, the lower values, to the extent of the difference which occurred, being reported for the atmosphere of air.

RALPH A. SHERMAN.[16] The authors have very properly stressed the increase in the iron content of the furnace ash and slag over that of the original coal ash and the importance of the degree of oxidation of the iron in the ash or slag in the determination of the fusion characteristics of the material. They have related the degree of oxidation of the iron to the composition of the atmosphere and suggest that for pulverized-coal furnaces, particularly, the fusion temperatures should be determined in an oxidizing as well as in the standard A.S.T.M. reducing atmosphere.

The writer agrees that the composition of the atmosphere is important both in the boiler furnace and in the ash-fusion furnace but another important factor in the determination of the degree of oxidation of iron, that the authors have not mentioned, is the temperature to which the slag is subjected either in the boiler furnace or in the ash-fusion furnace. As expressed by the writer in a Bureau of Mines bulletin:[17] "Ferric oxide is not only reduced by CO, H_2, and CH_4, which are present in boiler furnaces, but also, even in an oxidizing atmosphere, it begins to dissociate under atmospheric pressure at a temperature of about 2500 F into oxygen and a solid solution which is probably Fe_3O_4 in Fe_2O_3."

In an earlier publication[18] on the investigation of boiler-furnace refractories and later in bulletin[17] the writer called attention to the segregation of iron in the slag deposits on boiler tubes and presented data on the ratios of FeO to Fe_2O_3 in slags. For example, Table 2 of this discussion shows the content of the Fe_2O_3 in the ash of a coal from the Illinois No. 6 seam and the Fe_2O_3 and FeO contents of slags from a furnace fired with a traveling-grate stoker.

TABLE 2 ASH AND SLAGS FROM ILLINOIS NO. 6 COAL ON A TRAVELING-GRATE STOKER

	Composition, per cent Fe_2O_3	Composition, per cent FeO	Softening temp, F
Coal ash	19.8	..	2010
Slag from sampling tube in furnace	14.7	16.9	...
Slag from boiler tubes, next to tube	23.7	1.7	1960
Slag from boiler tubes, outside of pieces	22.7	7.3	2060

The slag taken from a water-cooled sampling tube in the furnace not far above the fuel bed not only contained a higher percentage of total iron than did the coal ash but the percentage of FeO was greater than that of the Fe_2O_3. The slag on the boiler tubes was divided for analysis into the less strongly fused material next to the tubes and the strongly fused material on the outside of the pieces. That next to the tube contained only 1.7 per cent FeO, whereas, that on the outside of the pieces contained 7.3 per cent. The atmosphere around the water-cooled tube may have been reducing at least part of the time, whereas, the conditions at the boiler tubes were undoubtedly oxidizing at all times. It is certain that the temperatures around the sampling tube were much higher than at the boiler tubes. We cannot be certain, therefore, as to whether the atmosphere or the temperature had the greater influence on the degree of oxidation of the iron in comparison of the material at the two positions.

Considering the material from the boiler tubes, however, it is reasonably certain that the atmosphere around the tubes was similar at all times but the temperature to which the material on the outside and inside was subjected was considerably different. The material, which collected adjacent to cool tubes, contained much less FeO than did that on the outside which had been heated to a higher temperature and this difference must have been due to the temperature alone.

The softening temperatures were unfortunately determined only under the standard reducing conditions. They were all so low that the effect of the differences in the degree of oxidation of the iron did not appear.

Another example of the difference in degree of oxidation of the iron was given in the bulletin.[17] In a furnace fired with Pittsburgh coal on a traveling-grate stoker, the percentages of Fe_2O_3 and FeO in a sample of slag collected on a water-cooled tube in the furnace were 7.2 and 39.2, whereas, the slag sample from the boiler tubes contained 46.2 and 2.9 per cent, respectively. Other examples of the degree of oxidation of slags are given in the bulletin, but none shows definitely whether the difference was due to the atmosphere or to the temperature.

In the data given in the authors' Figs. 5 and 6, the effect of temperature on the degree of oxidation may be suspected from the decrease in the differential between the fusion temperatures in oxidizing and reducing atmospheres as the temperature increases. That is, it is clearly shown that the difference in the results for the two atmospheres is less for the fluid temperatures than for the softening or initial-deformation temperatures. This undoubtedly results from the fact that the fluid temperatures were in the range where reduction of Fe_2O_3 occurred independently of the atmosphere surrounding the cone.

Likewise, the smaller differential for the two atmospheres with the lower iron content of the ash or slag may have been partly due to the effect of temperature, as these materials with lower iron content had to be raised to a higher temperature in the determination than did those with higher iron content.

Of the two factors affecting the degree of oxidation of the iron in ashes and slags, the composition of the atmosphere is probably more important than the temperature but the latter cannot be neglected. Their effects cannot be definitely determined by the determination of the fusion temperatures as, independent of the former history of the material or of the atmosphere in the fusion furnace, the cone may have to be heated high enough so that the temperature reduces the iron.

As the authors have concluded, further research is needed. To determine definitely the effect of atmosphere and temperature, the ferric and ferrous iron should be determined in samples of slags which had been heated to various temperatures in various atmospheres long enough to attain equilibrium and then quenched to maintain the iron in the state to which they had been brought.

[16] Supervisor, Fuels Division, Battelle Memorial Institute, Columbus, Ohio. Mem. A.S.M.E.

[17] "A Study of Refractories Service Conditions in Boiler Furnaces," by Ralph A. Sherman, U. S. Bureau of Mines, Bulletin 334, 1931, p. 101.

[18] "Refractories," by Ralph A. Sherman, W. E. Rice, and L. B. Berger, *Mechanical Engineering*, vol. 48, 1926, pp. 1389–1396.

J. E. Tobey.[19] Ash has always been the chief mischief-maker in the burning of coal on both stokers and pulverizers. Troubles from the other properties of coal have gradually been solved. While splendid progress has been made with ash, this has been partially offset by new critical conditions arising from the increase in boiler pressures and temperatures. The authors have forged another great link in the chain of priceless contributions Mr. Bailey has made in the solution of combustion problems. As indicated in the paper, the major importance of iron in ash and slag must be recognized and provisions made to do more exhaustive research on the problem in both plant and laboratory. Recent work by Bailey and Nicholls on ash properties furnish an excellent springboard from which to launch a comprehensive research program. It is believed that the interested technical groups should immediately set in motion the necessary machinery to carry through such a program.

Authors' Closure

The authors wish to express their thanks for the several discussions offered on this paper and for the active interest shown in further investigation of the behavior and influence of ash and slag. They acknowledge the many limitations in the data presented on a subject of such complexity and agree with Dr. Fieldner that, for ultimate solution, the problem must be resolved into its fundamentals. This will require the efforts of many investigators along different but coordinated lines.

The work of the Bureau of Mines on slag viscosities with relation to the phase of tapping, and of Gould, Brunjes, and others, as cited[10] by Mr. Julsrud, in regard to the effects of separable constituents of the ash at its source, are examples of great importance. In the meantime, much practical value may be gained from a more complete recognition of such over-all gross characteristics as can be determined from simple established tests, and from a study of their relation to conditions existing in the operating boiler furnace.

Prof. Estep has brought out one point which we wish to clarify at once by stating that all of the authors' slag samples, referred to in the paper, were taken from furnace or boiler heating surfaces and not from the thermal probe. A great many probe samples have of course been obtained, and have yielded data of rather special interest but, in the present discussion, the data are restricted to furnace or tube-bank samples. In procuring the samples, care was taken to prevent contamination, by using stainless-steel tools that were cooled intermittently by dipping them in water. The condition of these tools over extended periods of use indicates a negligible deterioration.

Laboratory fusion tests were made with standard-cone specimens heated in a gas-fired muffle furnace. For the "reducing condition," A.S.T.M. specifications were followed, by which a reducing atmosphere was maintained through the use of an excess of fuel gas, as evidenced by yellowish flame extending from the opening in the cover of the furnace for a distance of 6 to 7 in. The "oxidizing condition" was produced in the same furnace by using an excess of combustion air, controlled by means of orifices installed in the gas and air lines supplying the burner. Orsat analyses of gases within the cone chamber have shown the following composition:

	CO_2, per cent	O_2, per cent	CO, per cent
Reducing condition.......	8.2 to 10	0	2 to 4.8
Oxidizing condition.......	10.0 to 11	1.6 to 2.8	0

The final state of oxidation of the fused-cone sample, as checked in one case, after being specially mounted on a platinum plaque, was found to be:

	Ferric percentage	Total iron (as Fe), per cent	Fusing temperatures Reducing atmosphere, deg F	Oxidizing atmosphere, deg F
Original slag sample...	60.4	25.3	2000 2110 2360	2380 2490 2610
Fused in reducing atmosphere at 2360 F	4.6	..	..	..
Fused in oxidizing atmosphere at 2610 F	73.0	..	..	..

It is realized that some variation in test-furnace control, and its influence upon the final state of the sample, may add materially to the scattering of points in Figs. 5 and 6 of the paper, in addition to the effects of undetermined fluxes such as lime and magnesia, as mentioned by Mr. Nicholls. Dr. Fieldner's suggestion of establishing an A.S.T.M. standardized procedure for tests in the oxidizing condition is appropriate and is greatly needed.

The rather startling comparisons cited by Mr. Powell, in Table 2 of his discussion, may find a partial explanation in the early investigations of Dr. Fieldner, reference (1), where atmospheres of CO and CO_2 (and H_2 and H_2O) were used in various proportions. Higher softening temperatures were observed for the 100 per cent CO or the 100 per cent CO_2 atmospheres than for intermediate proportions of mixture, and this was attributed, respectively, to a predominance of metallic iron, ferric iron, and ferrous iron, as shown by final analysis of the samples.

The comparison of Mr. Powell's samples *C* and *D*, and of these with samples *A* and *B* is not at all clear, and it would be interesting to know the further composition of the ash, and whether these results are confirmed by duplicate test. It has been pointed out by Reid[20] that extremely low concentrations of oxygen, in an otherwise neutral atmosphere of unpurified nitrogen are nearly as effective as air in oxidizing molten slag. It may be possible that sample *C* would be found as high in ferric percentage as sample *D*, but there is still no apparent explanation for their relation to samples *A* and *B*.

By a somewhat opposite process, the softening temperatures of different ash samples, mentioned by Mr. Sherman, would tend to be similar to one another in spite of initially different ferric percentages, because of the reducing action of the test furnace when run according to A.S.T.M. specifications.

We are in complete agreement with Mr. Nicholls' recommendation of using the term "ferric percentage" instead of the authors' "per cent oxidation," for reasons which he mentions. This term becomes practical as an inverse index of the fluxing power, on the consideration that the content of metallic iron is usually negligible. We believe, however, that the expression for total iron content, and percentages of its oxide forms, is more readily grasped in terms of elemental Fe.

In the determination of iron forms, it is important that analysis methods be developed for identifying all forms individually. Digesting in HCl, or in H_2SO_4 and HF solutions has in some cases failed to dissolve more than 30 per cent of the total iron in the slag and, if the state of the undissolved portion is accepted, by difference, as being Fe_2O_3 the resulting value of ferric percentage may be considerably misleading.

In the paper, the presentation of data from laboratory work stressed the contrast between oxidizing and reducing atmospheres because that was the only controllable change in test procedure. It was not intended to imply that the atmosphere was the only

[19] Vice-President in Charge of Engineering, Appalachian Coals, Inc., Cincinnati, Ohio. Mem. A.S.M.E.

[20] "Control of Forms of Iron in the Determination of Fusion Temperatures of Coal Ash," by W. T. Reid, *Industrial and Engineering Chemistry*, Analytical edition, vol. 7, 1935, pp. 335–338.

factor influencing the ferric percentages as found in the ash and slag accumulating in different parts of the boiler furnace. Temperature undoubtedly has an important bearing, as mentioned by Messrs. Sherman, Nicholls, and Estep, and both effects may be modified by the time interval in which they are active.

Considering, however, that the principal source of iron from the coal ash is in the form of pyrites (FeS_2), it is apparent that the ferrous, or lower state of oxidation, may first be produced in the presence of heat and limited reaction with oxygen.

$$FeS_2 + \text{heat} + O_2 = FeS + SO_2$$

$$2\,FeS + 3\,O_2 = 2\,FEO + 2\,SO_2$$

This being the more stable form, at the higher temperatures of combustion, it is extremely probable that ferric iron can occur only after further oxidation takes place at a later stage of the process, where lower temperatures are conducive to its stability.

In other words, the higher furnace temperatures prevail where there is also a prevalence of free carbon and other reducing agents (such as hydrocarbons, CO, and H_2). These are intimately associated with the ash as it is released from the fuel particle. Although ample free oxygen is present in the primary burning zone, its reaction with the ash or slag particle is retarded by the shielding or the preferential oxidation of the combustible fractions. As the burning fuel passes on through the furnace, radiating heat to the furnace walls, the temperature decreases simultaneously with the diminishing or disappearance of the reducing agents, and a very small concentration of unused oxygen will then serve to convert the ferrous oxide to ferric oxide, as by the reaction:

$$4\,FeO + O_2 = 2\,Fe_2O_3$$

For the present it is conceded that both temperature and atmosphere (or carbon) are factors affecting the ferric percentage, but that they tend to vary simultaneously, and in a mutually assisting direction in a water-cooled boiler furnace. The higher temperatures in the initial stage may be considered as inhibiting complete oxidation of the iron, with progressive steps taking place from FeS_2 to FeO to Fe_2O_3 as the reducing action and temperature are decreased.

The relative importance of temperature and kind of atmosphere or presence of carbon remain to be established from further experiment, but it is doubtful if, in an operating furnace, any appreciable reduction takes place in a direction from the ferric to the ferrous condition.

The curves from Mr. Nicholls' Fig. 12 of reference (5), as cited by Prof. Estep, may be interpreted from a point of view that is reversed from the procedure of his tests, namely, that a slag of low initial ferric percentage, which is formed under conditions of high temperature, may remain low in ferric percentage until its temperature is sufficiently lowered to permit further oxidation. This is consistent with the findings of Mr. Sherman in his analyses of slag from the sampling tube and from the inner and outer portions of the boiler-tube samples, and is particularly significant in regard to the tap slag of Figs. 1 and 2 of the paper, where flame direction causes a direct delivery of ash and slag particles to the floor, with the presence of some carbon, and a maintenance of high slag temperatures.

In conclusion, the authors again wish to urge a greater appreciation of these important and potentially controllable properties of ash and slag in the interpretation of furnace design, operation, and coal selection, and to urge renewed activity in the study and coordination of research and practical experience.

Flow Processes in Underfeed Stokers

By MARTIN A. MAYERS,[1] PITTSBURGH, PA.

This paper is devoted to a consideration of the types of flow of materials in underfeed multiple-retort stokers. Such flow studies include solids such as the fuel and ashy refuse; gases, such as air and the products of combustion; and heat, especially that portion which recirculates within the bed and leads to preparation and ignition of the fuel. The types of flow have a definite bearing on the maintenance of the fixed pattern in the fuel bed which has been observed at each tuyère stack and the adjacent retort. From the data compiled, the author develops the features which would be possessed by a stoker designed for much higher duty than any now in existence, and analyzes the possible results to be attained.

COMBUSTION in fuel beds, though it has been practiced as an art for many years is, even today, scarcely understood; only recently has it become the subject of experimental investigation. The work of Kreisinger (1)[2] and his associates, reported in 1916, was perhaps the first attempt at a scientific approach to the elucidation of the mechanism. The results showed that, in ignited fuel, the air passing up through the bed first attacked the fuel by an over-all reaction resulting in the complete combustion of carbon to CO_2, which subsequently attacked additional carbon in higher levels of the bed, producing carbon monoxide with the absorption of a portion of the heat first liberated. It should be noted at this point that these over-all processes need not be determined by the so-called primary reactions of oxidation (2). Even though a considerable portion of the oxygen reacting initially with pure carbon goes to the monoxide rather than to the dioxide, the monoxide would be burned to the completely oxidized form before it had diffused out into the gas stream far enough to be caught by the sampling tubes in such large-scale experiments. The order in which these reactions occur in ignited beds, as shown by Kreisinger and associates, has been amply confirmed by repetitions of their work with many different fuels both here (3) and abroad (4, 5, 6, 7).

This work also showed that, in fuel beds which are more than a few inches deep, the rate of burning is proportional to the rate of air flow, a fact which is constantly used by engineers in the control of power boilers. That this must be so follows from the relative speeds of the combustion reactions with oxygen and carbon dioxide; in fact, the ratio between the weight of fuel burned by a stream of air passing through it and the stream velocity is a function of the fuel-bed depth given by an expression (8) approximating $(1 - e^{-x})$ for large values of x.

Fig. 1 shows that the great change in the function occurs for very small values of x, corresponding to fuel-bed depths of 1 to 3 in.; for larger values, the ratio increases continually but very slowly.

After these classical investigations, there was little novelty in the experiments performed until 1934, when Nicholls and Eilers (9) presented their paper on ignition by underfeed action. The new data did not conflict at all with the results of Kreisinger but, in so far as they covered the same region, confirmed them. The novelty in these results lay in the light they shed on the processes of ignition of fresh fuel entering the fire. In pure underfeed burning, as set up in these experiments, a deep fuel bed through which air is forced at controlled rates is ignited at the top and allowed to burn down, which is, obviously, equivalent to forcing fuel upward into a stationary fuel bed. It was found that only under certain conditions could the plane at which ignition occurred work its way down through the bed. At low air-flow rates the rate at which the plane of ignition advanced increased very rapidly; more rapidly, indeed, than did the rate of burning which, as previously indicated, was merely proportional to the air-flow rate. However, as shown in Fig. 2, as the air-flow rate increased still further, the rate of ignition could not maintain its advantage over the rate of burning and was finally overtaken. Beyond this point, the junction of the curve and the straight line in the figure, burning could take place only as fast as the fuel was ignited, so that the burning rate fell off from the straight line representing the burning rate of ignited fuel, and eventually reached zero along the downward course of the curve representing the ignition rate.

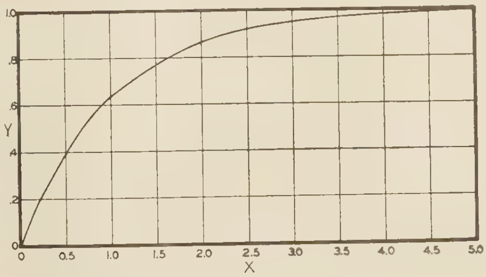


Fig. 1 The Function $y = (1 - e^{-x})$ Plotted Against x

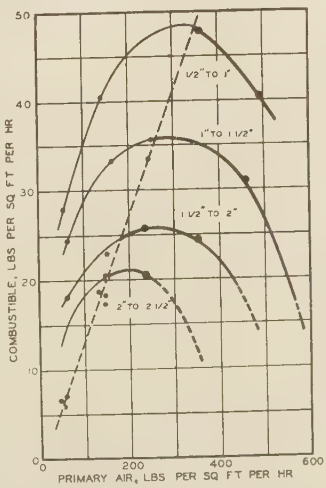


Fig. 2 Rates of Ignition (Curved Lines) and of Burning (Dashed Line) in Underfeed Burning (9)

(Figures on curves indicate coke size.)

[1] Coal Research Laboratory, Carnegie Institute of Technology. Mem. A.S.M.E.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

This behavior is so curious that it almost explained itself. It was not long before it was shown that a mathematical analysis of heat flow in a conducting bed (8) could explain the results observed, and permitted calculating rates of ignition (10) which agreed reasonably well with those measured in Nicholls' experiments, as shown in Fig. 3. The calculation is based on the concept that the heat radiated among neighboring particles within the fuel bed obeys the laws of thermal conduction in flowing in the direction of decreasing temperature. Thus, heat will flow from the hottest part of the bed, which is only a short distance above the plane of ignition, down toward the unignited fuel. A portion of this heat is picked up from the fuel by the air for combustion flowing up through the unignited fuel. As long as the air-flow rate is small, more heat may be conducted downward than is carried back again by the air, and the unignited fuel will rise in temperature at a rate depending upon the magnitude of this excess. The data indicate that the amount of heat conducted down into the unignited fuel increases very rapidly at low air-flow rates, but then reaches a nearly constant value so that, as the flow of air increases, it returns more and more of this heat to the burning zone. Thus, the rate of ignition must fall off, just as shown in Nicholls' experiments.

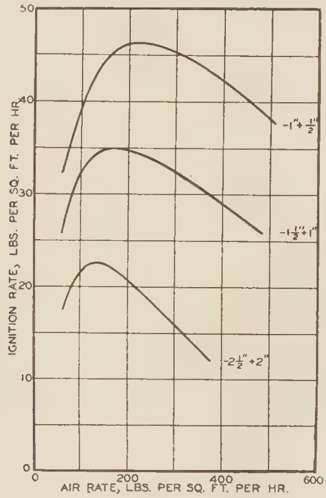


Fig. 3 Calculated Rates of Ignition in Pure Underfeed Burning
(Figures on curves indicate coke size.)

This mechanism is readily applied to the ignition process on chain-grate stokers; in fact, testing devices, similar in principle to the experimental apparatus used by Nicholls, have been developed abroad (11, 12, 13) to predict their performance with different fuels. It could not, however, account for the behavior of fuel beds in multiple-retort underfeed stokers. In the first place, higher rates of burning have been observed on such stokers than the maximum rates of ignition observed in the experimental apparatus. Since the fuel was burned, it must previously have been ignited, so it appeared that the underfeed stoker can ignite fuel faster than it could be done in pure underfeed burning. Furthermore, it is common knowledge among combustion engineers that unconsumed air could pass through the fuel beds of underfeed stokers under certain conditions of operation. If these beds, which might be 1 to 3 ft deep, had the same character as those occurring in the experimental fires, either pure underfeed, or overfeed as in Kreisinger's work, it would be utterly impossible for unconsumed oxygen to pass through them, since all the oxygen in the air disappears at levels not more than 6 in. above the plane of ignition.

More recent investigations have been undertaken to show the sources of some of these apparent discrepancies. They arise from our failure, until recently, to appreciate or understand the actual geometry of the fuel bed in underfeed stokers. A recent investigation (14) has, however, shown that the structure of the fuel bed in such stokers is quite different from the ideas current among engineers; it shows that the bed is made up of a multiplicity of similar units associated with each tuyère stack and the adjacent retort, and that conditions along each such unit do not change qualitatively from the head end of the stoker to the extension grate. The present paper is devoted to consideration of the types of flow of materials; solid, as the fuel and the ashy refuse; gaseous, as air and the products of combustion; and heat, especially that portion of it which recirculates within the bed and leads to preparation and ignition of the fuel, which takes place in such a way as to maintain the fixed pattern observed in such beds.

Structure of the Fuel Bed

A cross section through a portion of the bed of an underfeed stoker is shown in Fig. 4, while Figs. 5 and 6 show similar sections

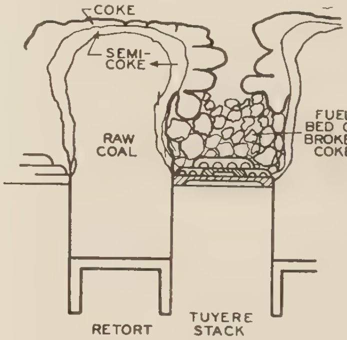


Fig. 4 Diagrammatic Cross Section of Fuel Bed of a Multiple-Retort Underfeed Stoker

on which are developed contours of equal temperature and of constant gas composition. The outstanding characteristics of the structure are as follows: The center of the retort contains green coal, extending almost the full height of the bed, with only a thin skin of coke on top of it. This coal is confined within walls of semicoke and coke, which border and confine the burning lanes directly above the tuyère stacks. The high temperatures generated in the burning lanes cause the formation of shrinkage cracks in the coke walls so that the agitation of the bed by the motion of the stoker breaks off portions of the now fully carbonized walls. These particles fall down into the burning lanes, producing a more or less well-defined fuel bed in the burning lane, the height of which is less than that of the fuel in the retort. Through this fuel bed, the main portion of the air stream flows upward. Since the fuel has already been heated to a sufficiently high temperature, the air is consumed by combustion, releasing energy.

Such cross sections as these are repeated across the stoker as many times as there are retorts; the sections at different longitudinal positions differ from that shown here, which was observed at about the middle of the stoker, only quantitatively, not in kind. Thus, at the head end of the stoker, the coke walls are thinner, and may be closer to the center line of the burning lane, producing a narrower lane; while at the tail of the stoker, the coke walls are thicker, and should, if the stoker is being correctly operated, have penetrated to the center line of the retort so that all the coal is coked.

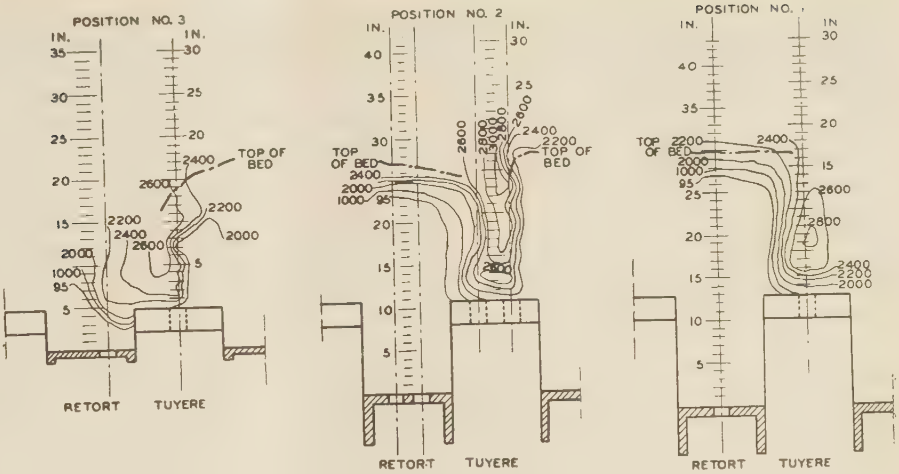


Fig. 5 Contours of Constant Temperature in Fuel Bed of a Multiple-Retort Underfeed Stoker (14)

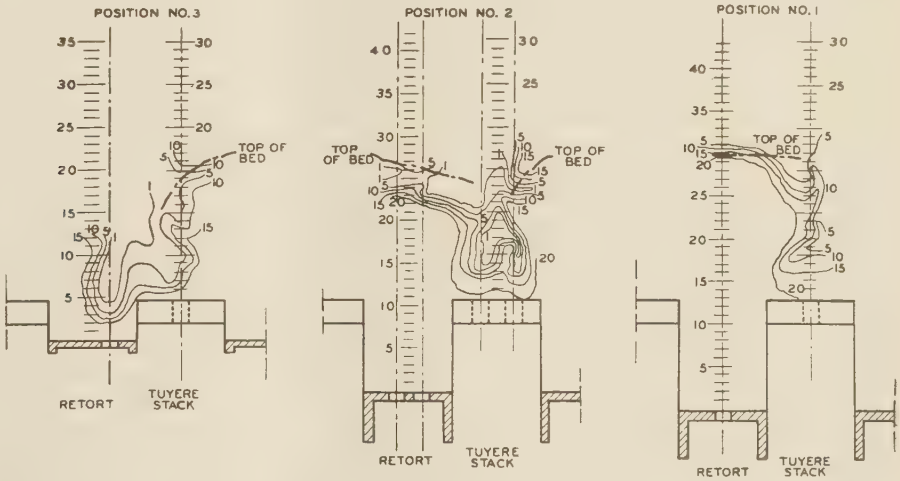


Fig. 6 Contours of Constant Oxygen Concentration in Fuel Bed of a Multiple-Retort Underfeed Stoker (14)

Gas and Air Flow

As mentioned, the main part of the primary-air stream flows from the tuyères up through the bed of broken coke at the bottom of the burning lane. During this passage it is consumed by the burning of the coke so that its oxygen content usually drops practically to zero at distances of 3 to 5 in. above the tuyères. If the bed in the lane is much deeper than that, considerable percentages of carbon monoxide may appear in the gases leaving the lane. On the other hand, there may be places within the lane where the coke bed has not been replenished at the proper rate, so that unconsumed oxygen may, at these points, pass entirely through the bed and out into the furnace. In general, the oxygen concentration at any level may vary within rather wide limits, as shown in Fig. 7, depending upon the stream velocity of the filament of flow from which the sample is taken, and the proximity of coke surfaces, but the average analysis tends to follow the pattern just described.

A relatively small portion of the primary-air stream passes under the bottom of the coke wall, which defines the burning lane, into the retort, and flows through it at low velocity because of the high resistance to flow caused by the dense packing of the small coal in this region. This air may return to the main stream passing up through the burning lane near the top, or it may pass out through the top surface of the coal in the retort into the furnace. In either case, it will carry with it the tar vapor and gases released by the coal being carbonized in the walls

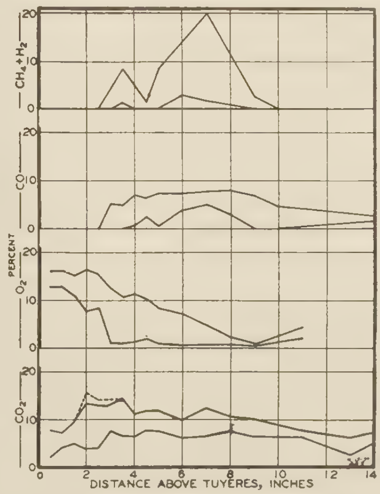


Fig. 7 Gas Composition in Fuel Bed Plotted Against Distance Above Tuyères

[The two lines show limits within which 50 per cent of the analyses lie (14).

of the lane. That this air has been within the combustion zone for only 1 in. or so is shown by the analysis of gas from the re-

torts at points inside the envelope of semicoke. It contains small concentrations of carbon dioxide, corresponding to that at levels of $^1/_2$ to 1 in. above the tuyères, and increasing concentrations of hydrogen and methane as the coking zone is approached. At yet higher levels in the retort, the carbon dioxide and monoxide also increase, while the oxygen decreases, until the skin of coke at the top of the retort is penetrated, when, in many cases the oxygen concentration again increases while the combustible constituents tend to disappear. This indicates that there may be a layer of comparatively cold stagnant air lying on top of the retorts, which is only slowly aspirated into the main gas stream rising from the burning lanes. Additional evidence of the existence of this phenomenon will be apparent later.

Some of the air, carrying products of combustion from the retort, may pass into the burning lane at levels below the top of the retort through shrinkage cracks in the coke and semicoke walls. Such shrinkage cracks are likely to be much deeper and more open in these walls than they are in coke formed in a by-product oven, because of the higher temperatures to which the walls are exposed, and the agitating action of the feeding mechanism. Thus, there may be quite free passage, at some levels, from the retorts out to the open space in the top of the burning lane. The entrance of this gas mixture into the burning lane is marked by the appearance of an increase in the concentration of the constituents hydrogen and methane in the gas, and a simultaneous increase in the concentration of oxygen. The combustible gases, both those brought by the air from the retort, and the carbon monoxide from the gasification that took place in the lower parts of the burning lane where a continuous fuel bed existed, combine rapidly with the available oxygen. The resulting gas flame, in the protection of the burning-lane walls, results in the highest temperatures observed in the bed, observations of 3000 F and even higher being not uncommon. In general, even with the air entering from the retort, there is insufficient oxygen completely to burn out the combustible gases issuing from the top of the burning lane. But, as the gases leave the lane, the oxygen concentration again increases, as shown in Fig. 7 for levels 9 to 11 in. above the tuyères. This also supports the conclusion that there is a rather stagnant layer of relatively cool unconsumed air above the retorts which is slowly aspirated into the jets rising from the burning lanes.

Visual evidence of the existence of this stagnant zone above the retorts appeared during tests of high-volatile coal in the work (14) referred to. During some of the preliminary runs with this coal, the boiler was brought off bank to a comparatively high load rather rapidly, so that about 50 per cent of the maximum rating (75 per cent of the test rating) was obtained before the furnace walls reached their normal temperature. At this time, clouds of smoke came out of the top of the retorts, but did not ignite as they did later on when the walls were hot. These clouds did not flow rapidly; they appeared to ooze out, and to roll slowly downward along the top of the retort. The ends gradually approached the stream rising from the burning lane, into which they were sucked and dispersed. Thus, the flow above the retorts must have been very slow; it had a negligible upward component, but was mainly along the top of the retort, lengthwise of the stoker. Evidently, the gas velocity distribution in the vertical direction shows marked discontinuities in the furnace just above the fuel bed; high values exist in the jets rising from the burning lanes, with relatively small upward velocities in the spaces between. A vector plot of gas velocities at the level of the top of the bed would look like a comb, with teeth pointing upward above each burning lane. The jets spread, as has been shown for other types of jet, with increasing distance from the level of the bed, and gradually entrain the gases from the space above the retorts. If cold secondary air is admitted to the furnace at low velocity, it will fall down on the retorts because of its high density and blanket them, only slowly being drawn into the main stream of combustion gases where it is needed. Thus, the need for very high pressures in the secondary-air jets is demonstrated, for only if sufficient velocity is supplied will the jets carry through the furnace and produce adequate mixing of the high-velocity streams from the burning lanes and the relatively stagnant gas above the retorts.

In its passage through the fuel bed at the bottom of the burning lane, the air and combustion gas obey laws similar to those observed for flow of cold gas through beds of broken solids (15, 16). The pressure drop of the gas is proportional to the height of the bed traversed, i.e., the pressure gradient is practically constant over a considerable portion of the height of the bed. This is the normal behavior of a stream of fluid passing through a uniformly packed passage of constant cross section. Moreover, the pressure gradient is proportional to a power of the rate of gas flow slightly greater than 2. While the exponent observed in the Hell Gate tests 2.1 is somewhat greater than has been observed in closely controlled experiments at normal temperatures, this discrepancy may be accounted for by the relatively few data from which it was determined, by probable inaccuracies in the data inseparable from plant-scale testing, by a possible temperature effect on turbulent flow through such beds not now recognized because of the lack of data for such high-temperature zones, or, by an effect due to the expansion and contraction of the gas caused by the violent changes in temperature during its passage through the burning zone.

Since the gas flow appears to obey the laws previously found by investigations on cool fluids not undergoing reaction, it is probably permissible to turn to these investigations for other information concerning flow phenomena in fuel beds. These experiments show that for small flow velocities, a bed of broken solids acts like a pipe, tube, or other resistance to flow. For very small flows, the streaming is viscous, and the resistance to flow increases as the first power of the mass velocity. Beyond a rather well-defined critical point, flow enters the turbulent regime, when the resistance increases with a power of the velocity (15) somewhat less than 2. This continues up to the point when the pressure gradient through the bed approaches in value the bulk density of the bed reduced to the same element of volume. That is, for a bed of coke, whose bulk density may be 36 lb per cu ft, the limiting pressure gradient will be $\frac{36}{1728} = 0.0209$ psi per in. of bed depth, or expressed differently $\frac{36}{62.5} \times 12 = 7$ in. of water column per ft of bed depth.

When this critical flow is reached, the bed is, on the average, supported by a force equal to its own weight, so that it becomes loose, almost fluid, and the interlocking of neighboring pieces, which gave it its character as a pure resistance, is reduced (16). If the air-flow rate is increased slightly beyond this point, the bed maintains its general form, but becomes capable of great extension in the direction of flow; any slight disturbance, however, or a slight further increase in flow is sufficient to disrupt it entirely so that the bed is completely blown away. The last phenomenon is, of course, known to every fuel engineer; it has not, however, been generally recognized that the point at which it occurs is so definitely fixed by the fundamental properties of the bed as an interlocking aggregate of particles.

Carman (15) has shown that the resistance to flow through a bed of broken solids can be estimated if the void volume and the specific surface of the bed are known. These properties can be measured (17, 18) or estimated from the size analysis (19). Fig. 8 shows the resistance to air flow in the turbulent regime for

beds with two different values of void volume and specific surface. The asterisks indicate the points at which instability would be reached. That the air rate at which this occurs is greatly affected by the character of the coke of which the bed is made is not of merely academic interest. In the Hell Gate tests, the air rate corresponding to an average burning rate of about 35 lb of coal per sq ft per hr was calculated to be about 1100 lb per sq ft per hr when referred to the air-admitting surface of the stoker. However, as shown in Fig. 4, the width of the burning lane was seldom as great as the width of the tuyères, so that the air-flow velocity in the lanes might be anywhere from $1^1/_2$ to 3 times as great as the air rate referred to the entire air-admission surface. Evidently, encroachment of the coke walls of the burning lane on the space above the tuyères is likely to lead to conditions of instability and should be prevented by provision of ample means for breaking them down.

In the neighborhood of the critical flow, when the bed is in the condition designated as "just moving," it has some remarkable properties, which are used, for example, in advanced German designs of gas producers (20). Particles of different densities are segregated in such a bed; if the particles are lighter than those of

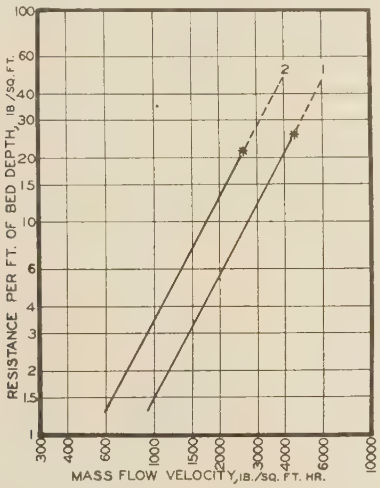


FIG. 8 DEPENDENCE OF RESISTANCE TO FLOW THROUGH FUEL BED ON AIR-FLOW RATE, REFERRED TO AIR-ADMISSION SURFACE

(Line 1 refers to a bed of coke having the size distribution described by line 2 of Fig. 10; line 2 to that described by line 3 of Fig. 10. Asterisks indicate upper limits of stability.)

the bed, they float out on top, while if they are heavier, they sink to the bottom. Thus, if a fuel bed could be operated stably at this point, it would automatically allow the ash to fall through the bed into the coolest regions where there would be little or no danger of slagging. Moreover, such a bed, which is most stable when operated with an intermittent blast of air (21), as in coal-cleaning tables, has the property of healing itself, that is, if channeling begins at any point in the bed, instead of the air flow being disturbed in such a way as to make conditions worse, the bed is so fluid that it immediately flows over the incipient channel and stabilizes the flow resistance. This, obviously, would be a desirable characteristic to obtain in a fuel bed. It seems likely that apparatus designed to employ a bed in the "just moving" condition would surmount several of the disadvantages now faced by grate firing of solid fuels.

If full use is made of air, flowing at velocities close to the critical by provision of a bed of adequate depth, the attainable burning rates would be much greater than are now possible. This is shown by the values of combustion rate equivalent to

TABLE 1 CALCULATED DEPENDENCE OF COMBUSTION RATE ON DEPTH OF FUEL BED AND AIR FLOW

	Combustion rates[a]					
	Referred to air-admission area, lb per sq ft per hr			Referred to total area,[b] lb per sq ft per hr		
CO at top of bed, per cent	0	15	30	0	15	30
Approximate fuel bed depth, in.[c]	<1	4.4	15	...	...	...
Air rates, lb per sq ft per hr						
150	13.0	18.0	23.9	6.2	8.6	11.4
1000	86.6	120	159	41.1	57.1	75.6
3000	260	360	479	124	171	224

[a] Rate of burning of combustible in coke, assumed to be carbon.
[b] Assuming ratio of tuyère stack area to total area corresponds to tuyère width of 10 in., retort width 11 in.
[c] For a typical high-temperature coke burned at an air rate of 150 lb per sq ft per hr, but not in proportion to the air rate.

various air-flow rates calculated in Table 1. Those in columns 2, 3, and 4 are the rates calculated for 0, 15, and 30 per cent carbon monoxide in the issuing gas, referred to the air-admission or burning-lane area; while those in columns 5, 6, and 7 are referred to the projected stoker area, on the assumption that the proportions of airadmission to projected area are approximately the same as in the Hell Gate stokers. It is evident that material increases in rating could be obtained if we could be assured that an unstable condition, due to encroachment of the coke walls on the burning lane, could be avoided.

FLOW OF COAL AND ASH

The flow of coal in the stoker takes place almost entirely in the retorts and in the walls of the burning lanes. There is practically no motion either lengthwise or crosswise of the stoker in the burning lanes. This was shown by the fact that porcelain probes inserted into the burning lanes could remain in position almost indefinitely without being subjected to severe transverse stresses. On the other hand, when such probes were inserted into the retorts through holes in the secondary rams, they were broken off during the return stroke of the ram at every stroke.

The retort performs a double function. In the first place, it has the obvious function of distributing coal to the entire length of the stoker. Coal which enters the retort near its top rises close to the front wall and is delivered to the burning lane at the head end of the stoker. Coal which enters at the bottom of the retort passes well down the stoker before being delivered to the burning lane and, in fact, in existing stokers may be forced straight out onto the overfeed section without ever having reached the burning lanes over the tuyère rows. The distribution is effected by the longitudinal flow of the coal which takes place at relatively high velocity, speeds of the order of 2.5 fph.

The second function of the retort is to coke the green coal and prepare it for smokeless burning above the tuyère stacks. This process takes place because the retort produces a transverse component of flow of coal which has risen above the level of the tuyères, causing it to flow toward and through the coke walls which border the burning lane. In this passage the coal is carbonized so that it is delivered to the burning lane as coke. The magnitude of this transverse component of velocity was about 0.75 fph in the stoker tested at Hell Gate, a value obtained from speeded-up motion pictures made as a part of the tests. The magnitude of this component may also be calculated from the load and the dimensions of the fuel bed, since it is obvious that coal can be burned on the tuyère stacks only as fast as it is delivered to the stacks by the retorts as a result of this transverse flow. Hence, it follows that fuel must flow across the boundary of the burning lane at the same rate as it is burned on the stack. Knowing the rate of burning from the load and the dimensions of the tuyère stacks, it is evident that the average flow across the boundary between retort and burning lane must be equal to the

rate of burning per foot of length of tuyère stack, divided by the depth of the fuel bed over the burning lane, i.e., the height of that boundary. An average value of the last-mentioned quantity representing the entire fuel bed cannot be estimated accurately, but Table 2 shows that the calculated values are comparable with those estimated from the motion pictures.

The transverse motion of the fuel from the retort toward the center line of the tuyère stack is terminated when the fuel mass breaks off from the wall of the burning lane. Such detached masses then fall vertically downward into the lane and help to form the continuous bed of discrete particles which exists at the bottom of the lane. On the burning lane itself the motion of the fuel is downward, just as in any other overfeed bed and, just as in an overfeed bed, the size of the particles decreases as they fall lower in the bed.

TABLE 2 TRANSVERSE VELOCITIES OF FLOW OF FUEL

Fire condition	Observed values, fph	Calculated, fph
Long pusher strokes at load of 125,000 lb per hr	1.85	
	0.81	
	1.33	
	-0.34	
	0.72	
	0.67	
	1.85	
	—1.18	
	1.45	...
Average	0.68	0.58
Probable error	0.22	....
Short pusher strokes at load of 119,000 lb per hr	1.06	
	-1.25	
	1.76	
	0.70	
	0.96	....
	1.83	
	0.65	
	1.26	
	0.70	
	0.51	
	0.55	....
	0.94	....
	1.38	....
	-0.42	....
	0.52	....
Average	0.75	0.65
Probable error	0.14	....

At this point a mysterious situation exists. Eventually practically all of the fuel is burned out and at the bottom of the bed ash will be collected, perhaps in a dry, powdery form if the cooling effect of the air is great enough, or perhaps as small clinkers. Somehow or other a large portion of this ash gets down to the ashpit, but so far the path by which it reaches that destination has not been discovered. During the Hell Gate tests, it was sometimes found that a layer of ash existed along the tops of the tuyères. When this occurred, opening the guide tubes preparatory to inserting a probe was followed by a shower of ash blown down through the tubes, but in an equal number of cases no ash came down and the probes read high temperatures right at the level of the tuyères. Is the agitation of the incoming air sufficient to cause the ash to flow along the slope of the tuyère stacks down to the ashpit? This seems hardly likely and yet the only other alternative, that all of the ash is blown up through the burning lane and then dropped out of the air stream when it slows down in the furnace, seems still more unlikely. Thus, there remains a hiatus in our knowledge of the flow of solid materials in the fuel bed.

The flow of the fuel during its distribution, coking, and burning is represented in Fig. 9, which shows a perspective view of the stoker and approximate lines of travel of representative portions of the fuel. This representation is, of course, not exact or complete but represents a first estimate of how coal flows into the fire in a multiple-retort underfeed stoker.

The multiple function of the retort may be responsible for some of the instability of stoker fuel beds. The rate of air flow through a bed at which instability sets in is greater the smaller the specific surface of the fuel, i.e., the more uniform in size are the particles which make up the bed, and the larger their size. Now, it has been found that, when coke is broken by impact, as in the shatter test, its size distribution may be plotted on probability coordinates, as in Fig. 10 (line 1). The slope and mean size of this distribution are characteristic of the coal and of the temperature of carbonization. This is the kind of size distribution

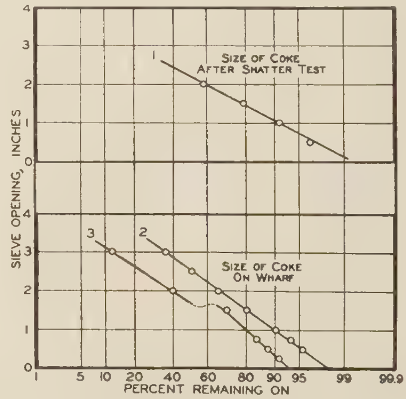
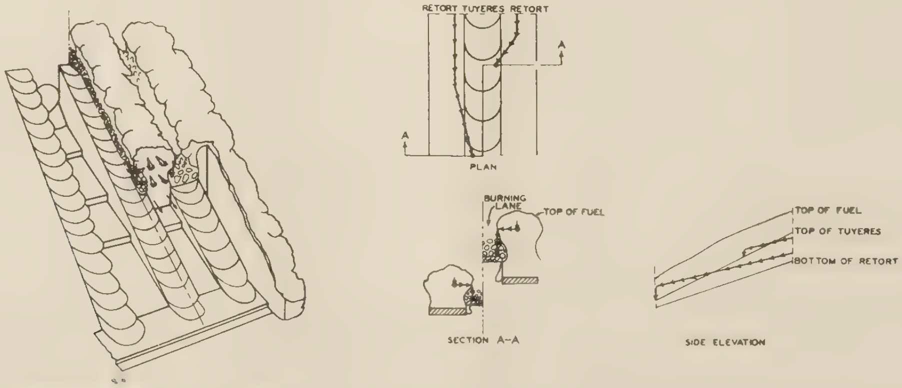


FIG. 10 COKE-SIZE DISTRIBUTIONS PLOTTED TO PROBABILITY COORDINATES

(Line 1, size of coke after shatter test. Line 2, size of coke discharged from a by-product coke oven. Line 3, size of coke made from same coal as discharged from a continuous vertical retort.)



APPROXIMATE DIRECTIONS OF FUEL FLOW LINES OF TRAVEL OF REPRESENTATIVE FUEL PARTICLES

FIG. 9 DIAGRAMMATIC REPRESENTATIONS OF RATE AND DIRECTIONS OF FUEL FLOW IN STOKER FUEL BEDS

that is also found in coke discharged from by-product coke ovens, as shown by line 2 in Fig. 10. When, however, the same coal is coked at about the same temperature in continuous vertical retorts, in which the coke is severely abraded during formation, the size distribution of the coke produced is that given as line 3. It is evident that a larger proportion of fine sizes is produced than corresponds to impact breakage. A fuel bed formed of the coke represented by line 3 has a much lower limit of instability than one formed of that represented by line 2 and would produce a much larger proportion of fly coke. This is shown in Fig. 8, where line 1 represents coke 2 and line 2 represents coke 3.

It can be readily seen that the superposition of a large longitudinal component on the flow of coal in the retort may produce abrasive action resulting in a yield of coke whose size distribution is more like that represented by line 3 than that represented by line 2, thus limiting the rate of air flow through the bed and, hence, the rating attainable. Therefore, it may be desirable to separate the function of longitudinal distribution from that of carbonization and delivery of coke to the burning line, if by so doing, it is possible to produce coke having characteristics more like that of line 2. Separation of these functions would, in addition, permit improved control of the fuel feed to different portions of the bed.

Flow of Heat

The flow of heat in a stoker is very complex and has so far defied exact analysis. We may, however, by drawing on the results of several other simpler processes, gain some insight into what must take place in the stoker. The principal portion of the heat released in the burning lanes remains in the gaseous products of the combustion and flows with them out of the burning lanes into the furnace and so to the boiler. This portion obviously will be distributed at its release from the bed in the same way as the gas flow itself is distributed, and will be transferred from the gases to cold surfaces according to well-understood laws of radiation- and convection-heat transfer. Another but very much smaller portion of the heat released will be transferred by radiation from the top of the fuel bed directly to cold surfaces. This portion is probably smaller than is usually estimated, since only a very limited fraction of the top surface of the bed is at extremely high temperatures, e.g., those portions represented by the burning lanes. The other 60 to 85 per cent of the fuel bed, i.e., the area over the retorts and walls of the burning lane, is at a very much lower temperature, probably not above 2200 to 2300 F and so radiates at a very much lower rate. It is well known that this portion of the fuel bed looks black when observed through a fire glass.

A third portion of the heat released is used to ignite the incoming fuel and thus recirculates within the fuel bed itself in the same way as the heat in preheated air recirculates in the steam-generating unit. Most of this heat is conducted through the coke and semicoke walls of the burning lane transversely into the green coal in the retorts. Thus, its direction of travel is directly opposed to the direction of flow of the fuel. In any steadily burning fire, this results in setting up a quasi-steady state, in which the temperatures at any point either do not change or fluctuate within limits about a constant mean value. Under these conditions just enough heat flows across the walls of the burning lane at any point to heat up the fuel flowing out from the retort toward the burning lane at that same point to a constant temperature.

Taking the retort as a whole, it is evident that at any distance from the front wall the conditions of heat flow are similar to those in a by-product coke oven at some stage during the process of coking the charge. At the head end of the stoker, this stage is similar to that immediately after the oven is charged. At the tail end of the stoker, the condition should be, if the stoker is being properly operated, similar to that just before the oven is pushed. That is, at the head end of the retort, almost all of the width of the retort is filled with green coal, with only a very thin skin of coke and semicoke set up at the walls of the burning lane. As we progress further from the front wall of the stoker, the coke wall increases in thickness, just as in the coke oven it does at later times during the coking period, until finally, at the tail end, the plastic layers produced in the coking process have met at the center of the retort, just as they do at the end of the coking period in a by-product coke oven.

The high rates of ignition, by comparison with those found in pure underfeed burning, observed in multiple-retort stokers can be understood in the light of this picture. In the first place, the ignition surface is not a plane parallel to the plane of the stoker, but the sum of all the nearly vertical surfaces which separate the semicoke walls of the burning lanes from green coal. Thus, the ignition surface may be greater than was previously thought. In the second place, there is no flow of air through the ignition zone, which was previously shown to return conducted heat to the high-temperature region in pure underfeed burning, so that all of the heat conducted into coke and green coal is available for coking and igniting it.

The similarity between the stoker retort and the by-product coke oven provides a means for calculating the rate of coking in a retort and so of controlling it, for a great deal of research has been done on this problem in connection with by-product coke ovens and the results of such research are immediately applicable here. It has long been known that the carbonizing time in coke ovens, other factors being held constant, varies as a power of the oven width greater than 1. Since the volume of coke produced is directly proportional to the oven width, it follows that an increase in output can be obtained by the use of narrower ovens, a fact which is made use of in modern construction. If simple heat conduction were responsible for the coking process, the coking time would vary as the square of the oven width, but the correlation by H. H. Lowry, director of the Coal Research Laboratory, of data on carbonization in experimental retorts (22), indicates that the exponent 1.6 more nearly represents the dependence of coking time on oven width. The same exponent appears to apply, as well, to many different types of commercial ovens.

Thus, we are justified in saying that the average rate of coking in an oven, hence in a retort, varies inversely as the 1.6 power of the retort width, so that the rate of coking per retort varies as the inverse 0.6 power of the width. It is evident that, in order to secure higher rates of coking with other conditions constant, it is necessary to decrease the width of the oven or retort. By the use of this principle, a formula can be derived expressing the correct proportioning of retort and tuyère widths for any desired rates of burning.

A Heavy-Duty Stoker

On the basis of the descriptions of flow processes in the stoker which have been presented, we may state the features that would be possessed by a stoker designed for much higher duty than any now in existence. In the first place, such a stoker must have narrower retorts than appear in existing stokers. This will make possible the preparation and coking of coal at a much higher rate than existing stokers can now perform. If the retort width were reduced to 25 per cent of that in conventional stokers, coal could be coked in each retort at nearly 2.3 times the rate it is now done. This would permit the maintenance of deeper fuel beds over tuyères of the same width as now are used, thus raising the ratio of active surface to the total area of the stoker and permitting the average burning rate to approach more closely the burning rate referred to the air-admission surface.

It is probably essential that the functions of distribution, coking, and delivery, now combined in the retort flow, be separated so that the coke supplied to the tuyères be as nearly uniform in size as possible. As part of this separation of function, there must be provided sufficient agitation to eliminate the possibility that coke walls may expand and constrict the burning lane.

Furthermore, such a stoker should be supplied with a pulsating air blast the maximum flow rate of which is sufficient to produce a condition close to that corresponding to instability of the bed. Under these conditions, the bed is fluid enough to flow over any incipient blowholes, and would allow ash liberated by combustion to sink through it to be discharged at the bottom of the bed in a cool zone. Control of a stoker operated on this principle would be secured by varying the portion of its cycle during which the flutter valve controlling the pulsation remained open, not on variation of the forced-draft pressure.

Finally, since it is desirable, by the use of a deep fuel bed, to require the air injected beneath the stoker to take up as large a weight of the solid fuel in the form of combustion gases as it will carry and also, since operation at air-flow rates close to full teeter will cause a portion of the dust to be carried out of the bed, it will be necessary to supply a large proportion of the air for combustion as overfire air. Under these conditions we may expect that only from 40 to 75 per cent of the air for combustion will be supplied as primary air. The remainder will be injected over the fire under sufficiently high pressure to enforce good mixing with the stream of combustion gases rising from the fuel bed. This secondary air may be preheated to as high temperature as may be desirable, thus making possible the use of this very flexible method of heat recovery in stoker-fired installations.

BIBLIOGRAPHY

1 "Combustion in the Fuel Bed of Hand-Fired Furnaces," by Henry Kreisinger, F. K. Ovitz, and C. E. Augustine, U. S. Bureau of Mines, Technical Paper No. 137, 1916.

2 "The Chemistry of Gasification With Reference to Small Producers," by D. H. Bangham and J. G. Bennett, *Fuel*, vol. 19, May, 1940, pp. 95–101.

3 "Properties of Cokes Made From Pittsburgh Coals in Various Types of Plants," by P. Nicholls, G. S. Brewer, and Edmund Taylor, Proceedings American Gas Association, 1926, pp. 1129–1143.

4 "Verbrennungsverlauf von Steinkohle an einer Wanderrostfeuerung," by R. Löwenstein, *Die Wärme*, vol. 57, 1934, pp. 97–101, 121–125.

5 "Coke for Blast Furnaces," by R. A. Mott and R. V. Wheeler, The Colliery Guardian Company, London, 1930, pp. 126–140; "The Quality of Coke," by the same authors, Chapman & Hall, Ltd., London, 1939, chap. 11, pp. 132–167.

6 "Verfahren zur Betriebsmässigen Überwachung der Reaktionsfähigkeit von Koks," by G. Speckhardt, *Glückauf*, vol. 72, 1936, pp. 225–231.

7 "Verbrennungsverlauf bei Steinkohlen mittlerer Korngrossen," by H. Werkmeister, *Archiv. für Wärmewirtschaft*, vol. 12, 1931, pp. 225–232.

8 "Temperatures and Combustion Rates in Fuel Beds," by M. A. Mayers, Trans. A.S.M.E., vol. 59, 1937, pp. 279–288; see Eq. [15].

9 "The Principles of Underfeed Combustion and the Effect of Preheated Air on Overfeed and Underfeed Fuel Beds," by P. Nicholls and M. G. Eilers, Trans. A.S.M.E., vol. 56, 1934, pp. 321–336.

10 "Ignition in Beds of Solid Fuel," by M. A. Mayers and H. G. Landau, *Industrial and Engineering Chemistry*, vol. 32, April, 1940, pp. 563–568.

11 "The Swelling of Coal During Combustion," by A. C. Dunningham and E. S. Grumell, *Fuel*, vol. 17, 1938, pp. 324–327; "Factors Affecting the Temperature of Travelling Grate Stokers," by the same authors, ibid., pp. 327–334.

12 "Pt. 1. Die Zündung fester Brennstoffe auf dem Rost. Pt. 2. Untersuchungen über das Zündverhalten," by P. Rosin, H. G. Kayser, and R. Fehling, Berichte der technisch-wirtschaftlichen Sachverständigenauschüsse des Reichskohlenrats D51, pub. by V.D.I. Verlag in Komm., Berlin, 1935.

13 "Combustion Conditions in Fuel Beds," by E. Tanner, Veröffentlichungen des Vereins zur Überwachung der Kraftwirtschaft der Ruhrzechen, no. 1, 1934-1935.

14 "The Fuel Bed Tests at Hell Gate Generating Station—1937-1938," by M. A. Mayers, W. H. Dargan, J. Gershberg, B. C. Dalway, M. J. Williams, and E. R. Kaiser, presented at 1938 Meeting of The American Society of Mechanical Engineers.

15 "Fluid Flow Through Granular Beds," by P. C. Carman, Trans. Institution of Chemical Engineers, vol. 15, 1937, pp. 150–166.

16 "The Cleaning of Coal: An Experimental Determination of the Fundamental Principles of Pneumatic and Wet Separators," by A. A. Hirst, Trans. Institution of Mining Engineers, vol. 79, 1930, pp. 463–497.

17 "Test for Cubic Foot Weight of Coke," A.S.T.M. Standard D292-29, *A.S.T.M. Standards on Coal and Coke*, American Society for Testing Materials, Philadelphia, Pa., 1938, pp. 133–134. "Test for Volume of Cell Space and Lump Coke. A. Determination of Apparent Specific Gravity," A.S.T.M. Standard D167-24, ibid., pp. 119–122.

18 "Determination of the Specific Surface of Powders—I," by P. C. Carman, *Journal of the Society of Chemical Industry*, vol. 57, 1938, pp. 225–34T. II, ibid., vol. 58, 1939, pp. 1–7T.

19 "Methods of Representing Distribution of Particle Size," by J. B. Austin, *Industrial and Engineering Chemistry*, Analytical edition, vol. 11, 1939, pp. 334–339.

20 "Gasification of Fine-Grained Solid Fuels," U. S. Pat. No. 2,111,579, issued March 22, 1938, to F. Winkler and Ed. Linckh, assignors to I. G. Farbenindustrie A-G.

21 "Co-Ordination of Theories of Gravity Separation," by A. A. Hirst, Trans. Institution of Mining Engineers, vol. 94, 1937, pp. 93–113.

22 "Gas-, Coke- and Byproduct-Making Properties of American Coals and Their Determination," by A. C. Fieldner and J. D. Davis, U. S. Bureau of Mines Monograph No. 5, 1934.

Discussion

H. F. Lawrence.[3] The writer would like to emphasize the fact, as stated in the paper, that "conditions along each such unit (retort and tuyère) do not change qualitatively from the head end of the stoker to the extension grate." Many operators do not realize that the lower ends of the retorts should contain only green coal and that the coking and burning zone should be entirely above the top boundary line of the retort. When this is understood and the strokes so adjusted as to maintain this condition, better results are obtained both in combustion efficiency and maintenance.

The structure of the fuel bed, as shown in Fig. 4 of the paper, is, to the writer's mind, greatly exaggerated. In this connection, it might be interesting to consider the inventor's conception of this structure. The patent application of Elwood E. Taylor, filed December 26, 1903, describes this structure as follows:

> "In the operation of the stoker, the fuel bodies in the several retorts constitute legs of a single fuel bed spreading over the mouths of the retorts, this fuel bed burning with the incandescent fuel on top and the coking fuel underneath and extending back into the retorts. The fuel bed receives its support from the walls of the retorts and, owing to the cohesion and arching property of the coking fuel as it swells during the coking process and is fed outwardly by the retort pushers, the fuel bed is kept substantially free from the twyer faces by arching over them. The feed of the ash is a gravity feed down the slope of the fuel bed, induced by the outward feed movement of the fuel across the plane of the retort mouths."

In order to check this, the writer's company made several three-piece tuyères which could be removed from underneath without disturbing the fuel bed above them. These tuyères were removed while the stoker was in normal operation. It was found that a perfect arch of coke existed over the tuyères which maintained itself after the tuyères were removed, very little material falling through.

With reference to the use of secondary air, all of our experience and experimental investigations indicate that the maximum quantity of secondary air should not exceed 10 per cent of

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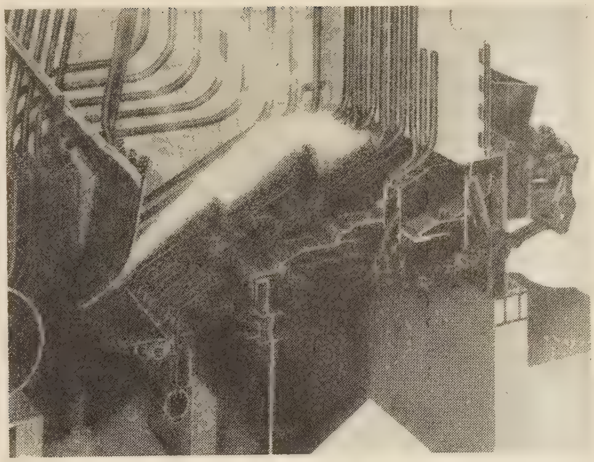

FIG. 11 ARRANGEMENT OF STOKER FOR OBSERVING VARIOUS COMBINATIONS OF COAL-DISTRIBUTING ELEMENTS

the total air and, in most instances, this limit is much lower.

We have tried a number of times to produce large quantities of combustible gases to be burned later with secondary air and have had no success. It is impossible to make a gas producer from the multiple-retort underfeed stoker.

We agree that secondary air to be most effective should be introduced through high-velocity jets. From the author's analysis, it appears that the secondary air would also be more effective if introduced at right angles to the retorts. We will have an installation operating in the near future with secondary air introduced transversely of the retorts and also using high-velocity jets.

For turbulence, we have also used the combustion gases taken from one of the rear passes of the boiler. These gases are introduced through high-velocity jets. Since the oxygen content of these gases is low, larger quantities can be used with no increase in the total weight of gas discharged.

Secondary air is used primarily to control smoke emission from the stack. In addition to the elimination of smoke, there is also more complete combustion of solid carbon in the furnace. Slagging of tube surfaces is also decreased.

We have been studying the flow of fuel through the retort. These studies have been made on actual stokers, with windows placed so that the movement of the coal in the retort could be observed. The object has been to find some method by which more coal could be fed through the retort without increasing the surface movement of the fuel; in other words, reduce the agitation of the burning fuel over the tuyères.

Various combinations of coal-distributing elements have been observed by arranging a stoker of the type illustrated in Fig. 11 of this discussion, for inspection of the moving coal. The velocity and direction of flow in the various sections were marked on the glass windows and later charted to scale for further analysis.

The construction eventually decided upon was installed in a stoker which was operating on low-volatile bituminous coal. The customer desired additional capacity from the unit and the change permitted him to operate reliably at a 27 per cent increase in output. Additional applications of the principle demonstrated have indicated improvements with many other types of coals.

D. J. MOSSHART.[4] In the main, the concepts and hypotheses constructed by the author parallel and confirm principles which for many years have been recognized and employed in the design and operation of stokers. These principles, evolved under operating conditions and based on uncounted observations made from above and below the fuel bed, have been proved in the trial, success, and adoption of designs and methods employing them.

The writer feels that the author's excellent analysis is worthy of objective study by all combustion engineers, and that it is appropriate to offer certain comment supplementary to it.

As a preliminary remark, any hypothesis or analysis of a combustion process may be more thoroughly understood (or safely rejected) if it is constructed in two ways, (a) by considering the coal as fuel and the air as supporting combustion thereof, (b) by considering the air as fuel and the coal as supporting combustion. The phenomena discussed in the early part of the paper offer an excellent illustration. Ignition will be lost if the flow of coal is too rapid through a stream of air constant in quantity.

The illustrated typical structure of the fuel bed is fundamental. It is obtained with all sorts of solid fuel; with coke breeze and with subbituminous coals, which do not coke or agglutinate. The formation of coke walls along each side of the burning lane is helpfully incidental; it provides stability and inhibits blowing away of the fuel.

It is suggested that the finding of free oxygen just above the fuel in the retort was probably due to some specific circumstance associated with the particular setup used. Ordinarily there is a short, low-velocity but hot flame, or rather assortment of flames, flowing hither and yon over the coal in the retort. This flame is aspirated into the jet over the burning lane, the slowness of its aspiration being due to the fact that it lies between two aspirators of substantially equal power and wavers between them. The hypothesis of a stagnant layer of cool, unconsumed air is questioned. How can it long exist in the presence of hot coke and combustible gases?

The practical application of the principles here discussed is quite simple. The fuel burns almost entirely in the burning lane. It burns partly as raw coal (finer particles sifting through the coke walls), partly as coke, and partly as gases and vapors, evolved in the formation of the coke walls and aspirated into the flame or jet of partly consumed air which issues from the lane. The average combustion result (CO_2 or excess air) obtained is a function of the depth of the lane and the size and disposition of the fuel particles therein. Under given conditions of fuel and load, this establishes the thickness of fuel bed required and this thickness is controlled to maintain the desired result, i.e., the feed of coal is controlled and the distribution of coal is adjusted to give a fairly uniform combustion condition over the entire stoker.

At any given spot, the quality of combustion probably varies over a wide range of excess air with such great rapidity that the 50- or 100-cc sample taken by the Orsat is a composite of several values. However, with an infinite number of spots, the integrated condition obtained some distance above the fuel bed is one of fairly uniform quality.

This burning in lanes is by no means exclusively characteristic of the underfeed section of the stoker. Link-grate stokers, with link-grate overfeed sections of length equal to that of the retorts, burn coal in precisely the same way. The link grates have two components of motion, propelling and breaking, separately regulated. The laned fuel bed received by them from the underfeed section is carried along in this characteristic form until it nears the end of the grate, the burning lanes gradually widening, and the fuel lanes finally disappearing. Quality of combustion is controlled by controlling the rapidity with which the laning is obliterated, i.e., by altering the relation between the propelling and the breaking components of motion.

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This naturally leads to consideration of the author's statement of mystification over the manner in which ash gets to the ashpit. This question has so far been answered only by observation of the progress of the ash and concurrent rationalization. It is believed that the following conclusions are not only tenable but fairly close to the complete story:

1 Since nearly all of the ash must remain as a residue in the coke which burns over the tuyères, it is exposed to temperatures generally higher than the fusing temperature of any ash. Most of it fuses and melts.

2 The fused ash is heavier than the burning coke with which it is associated, so it drips or drops down through the burning lane to be chilled to solid form and comes to rest on the tuyères and among the coolest particles of coke above them. Thus, there is deposited on the tuyères a layer of ash of a thickness which is minimum at the upper end of the stoker and maximum at the lower.

3 The stream or column of coal in the retort with its confining walls of coke moves as already described. The ash is slowly but surely dragged along by the moving column. Sometimes it agglomerates to form a clinker on the tuyères which stands still and grows until positively seized by adjacent columns and moved integrally with them. (Probably, the loose ash flowed around the author's probes and the clinkers, if any, obligingly withheld movement while the probes were in place. High temperatures right at the tuyères usually mean that a clinker has just passed by and swept the ash away.)

4 Thus, the ash is moved by the passing coal and, thus, it must finally be ejected from the underfeed section of the stoker, i.e., by ejecting enough coal from the retorts to carry the ash away. Overfeed sections are provided for the express purpose of burning the coal used for this function.

Herein is the principal reason for the application of the link-grate overfeed section. It provides for burning the coal ejected from the retorts in order to dispose of the ash. Additionally, it comprises a large portion of the total stoker area where the concentration of ash is highest and, being a positive conveyer, eliminates in this area dependence upon movement of the coal to move the ash.

In undertaking to burn any and all varieties of coal on standardized apparatus there naturally occur instances where it is impractical or impossible adequately to control combustion solely by regulating distribution of the coal. Then it becomes necessary to use secondary air.

The method of injecting the air involves more than velocity. Basically it involves the energy required to obtain penetration of the jet to the area needing the air, i.e., the area of the stoker above which the flame is dense and smoky. This energy is the product of mass and velocity and, if the mass usable is great enough, the velocity can be low.

Again, method is important. Penetration rather than flame deflection is to be sought. Turbulence will more readily be obtained with a few large jets boring into the flame than with many small but powerful jets trying to push the entire mass of flame hither or yon. For instance, we find that jets of 20 to 40 sq in. area spaced 3 to 5 ft apart and using air pressures of 8 in., and often less, give greatest penetration and produce greatest turbulence.

J. E. Tobey.[5] This paper represents the culmination of many years of special study, both theoretical and practical, of combustion on multiple-retort underfeed stokers. The writer has followed the author's work with a great deal of interest and anticipation. In fact on two occasions, he visited Hell Gate Station, while the tests were being conducted, and crawled into the doghouse underneath the stoker with the author, from which location the probes were introduced into the fuel bed. The author has been an earnest seeker after truth and has worked tirelessly to secure the data presented in this paper.

From personal observations, the writer agrees that there is need for a speed up in the coking process on underfeed stokers and that many present burning troubles would be eliminated if the fuel could pass more quickly and completely from the raw-coal state to coke.

It is believed that the author's idea of a pulsating air blast has merit from the standpoint of classifying ashy material and coke in the tuyère lane and increasing the mobility of the former. In addition, the writer has been of the opinion for a long time that a pulsating air blast might create a scavenging effect on burning coke, exposing fresh surfaces to oxygen attack, tending to remove insulating films of ash and inert gas. It is hoped that stoker engineers will collaborate with the author in developing a higher-duty stoker.

R. S. Julsrud.[6] In his description of the structure of the fuel bed, the author speaks of a "thin skin of coke" covering the top of the green coal in the retort. The writer's observation of numerous stoker fuel beds, both with furnace glasses and the naked eye, has never revealed such a "coke skin." On the contrary his observations have indicated that, as the heated coal emerges from the retort, the pieces become soft, their edges round off, and they become plastic. The degree of plasticity attained varies with the fuel, some coals becoming so plastic as to appear a "frothy" mass floating over the retort, whereas, others soften but slightly and can be distinguished throughout their evolution from green coal to coke formation. It is this plasticity, which if extreme, produces "matting" or "islands," sometimes extending into the burning lanes, which appear black when viewed with furnace glasses. Optical-pyrometer readings of these plastic masses reveal temperatures from 1600 to 2000 F, depending upon the fuel sizing, degree of plasticity of the fuel, combustion rate, and other factors.

The presence of a stagnant layer of unconsumed combustible over the retort of both single- and multiple-retort stokers has been observed by the writer. This is particularly noticeable with single-retort stokers of the "on-off" type after lengthy off periods. A steam or heat demand starting up the feed and forced-draft fan creates dense clouds of slowly rising gases over the retort which ignite as they are drawn into the high-velocity air and gas streams issuing from the coke fissures in the burning zone.

Not only is high-velocity primary air desirable for entraining and burning unconsumed combustible gases above the retort, but it also assists in the rapid formation of coke as the fuel passes from the retort to the burning lane, provided of course that such velocity does not "lift" or "blow" considerable fuel from the bed.

Calculation of the transverse velocity of fuel over the grates of single-retort stokers at various combustion rates have shown speeds of from 0.3 to 1 fph, which would appear in line with those determined by the author at the Hell Gate tests at probably higher combustion rates.

Again referring to "on-off" type single-retort stokers, the writer has observed the tendency of the fuel bed to "heal" and correct unstable fuel-bed conditions such as "blowholes," large coke masses, and "islands" of plastic fuel during off periods. This is probably due to the plastic nature of the fuel bed permitting it to flow into cavities and the additional creation of coke fissures due to contraction of the coke caused by fuel-bed temperature changes.

The author's four suggestions for the design of a heavy-duty

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[6] Address, Harmon-on-Hudson, N. Y. Assoc. Mem. A.S.M.E.

stoker have considerable merit. Retorts of narrower width should present no unusual difficulties. The delivery of uniform-size fuel to the tuyères is more difficult due to the variable size reduction occurring in the retort because of attrition, impact, and other causes. It should be best accomplished by delivery of a uniformly sized fuel to the stoker hopper. As the author states, pulsating primary-air pressure has been applied to advanced German gas producers and should be applicable to stokers. Overfire air of relatively high velocity is being at present applied to underfeed, chain-grate, and spreader-type stokers with good results.

Author's Closure

The discussers have made material additions to our knowledge of the underfeed stoker. In particular, the author was greatly interested in Mr. Lawrence's description of the fuel bed arched over the tuyères and so keyed that little or none of the material fell down when the tuyères were removed. The question arises whether this is advantageous or the reverse; such a fuel bed would appear to be so densely packed as to set up very high resistance to the primary-air stream.

Mr. Mosshart's analysis of ash flow along the tuyère stack was also very interesting. The mechanism he describes is similar to that believed to exist by the author before the tests at Hell Gate; the absence of confirmatory observations during those tests led him to question the mechanism. The ash certainly gets down to the pit; its path of flow may be that described by Mr. Mosshart; but that it is so, is not yet proved.

The author is grateful to Mr. Mosshart for his confirmation of the "lane" mechanism of burning and for its extension to fuels and stokers other than those which were observed in the Hell Gate tests. Mr. Mosshart's experience of fuels and of stokers is so wide that the generality of this conclusion can now hardly be questioned.

His explanation of the function of the link grate suggests that it is used to palliate conditions arising from the need to adjust coal flow along the retorts to two different requirements: first, so that the coal is fully coked at the end of the retort; and second, so that the longitudinal movement is large enough to discharge the ash. It is evident that these requirements are not likely to be compatible so that it might be desirable to separate the ash-discharge function from the coal-distribution function. This is done in the design of which the last section of the paper is a partial description.

It is very helpful to have Mr. Julsrud's confirmation of the observation of slowly moving clouds of combustible gases rising from the retorts and being aspirated into the stream from the burning lanes when starting up from a banked condition. In connection with Mr. Mosshart's discussion of the same subject, the author wishes to emphasize that he described a *relatively*, not an absolutely, stagnant gas layer over the retorts. The author believes his picture and that given by Mr. Mosshart are essentially the same and differ only in the description.

Finally, the author wishes to thank Mr. Tobey for his encouragement. The acceptance in principle of the suggested novelties in underfeed-stoker design by an engineer of such broad experience as Mr. Tobey's indicates that they may be profitable lines of attack to follow. It seems that the establishment of research programs, to supplement their development engineering, might be a profitable venture for stoker manufacturers.

Lubrication of General Electric Steam Turbines

By C. DANTSIZEN,[1] SCHENECTADY, N. Y.

This paper constitutes a discussion of the viscosity index, organic acids, and oxidation as related to oils used in lubricating steam turbines manufactured by the author's company. Advisedly, responsibility for the quality of a turbine oil is placed as it should be upon the supplier of the oil. Nothing in this paper should be construed as affecting that responsibility. The lubricating-oil experience of turbines operated at the Schenectady Works powerhouse is cited.

Because turbine oils, having an appreciable variety of physical characteristics, are functioning well in steam turbines, the General Electric Company does not issue narrow specifications for such oils, but does issue the relatively broad recommendations contained in Table 1.

TABLE 1 RECOMMENDED LUBRICATING-OIL SPECIFICATIONS FOR TURBOGENERATOR SETS

Properties	Land and marine direct-connected turbine-generator sets	Land gear sets and oil-ring lubrication, turbine-generator sets	Marine turbines: Auxiliary, geared generator sets	Marine turbines: Propulsion, geared sets
Saybolt viscosity, sec:				
at 100 F	140–170	280–325	280–325	....
at 120 F	....	....	....	220–260
at 210 F (approx)	43	49	49	55
Flash point, F (min)	330	350	350	360
Neutralization number	0.05 Max	0.05 Max	0.05 Max	0.05 Max
A.S.T.M. steam-emulsion number	90 Max	90 Max	90 Max	120 Max
Maximum viscosity before starting, sec	800	800	800	800
Minimum oil temperature before starting, F	50	70	70	85
Operating bearing-inlet oil temp, F	110–120	110–120	110–120	110–120
Operating bearing-outlet oil temp, F	140–160	140–160	140–160	140–160
Minimum oil-tank temp, F	130	130	130	130

Unfortunately, the recommendations in Table 1, purposely made so as to include the well-tested products of many oil companies, are so broad that they may include not only good but also poor oils. For this reason, it is suggested that the oil to be used should have a good service record in the field. It may be argued that, if this suggestion is followed too literally, it tends to discourage the development of new refining processes in the production of turbine oils. However, most oil companies have within their organizations turbines with which they can experiment on new oils before placing such oils on the market.

The author's company also states that the oil should be a petroleum derivative free from water, sediment, soap, and resins or any materials which in service will prove injurious to the oil or to the turbine with its accessory equipment.

The variation in viscosity of mineral oil with temperature can best be plotted on a special A.S.T.M.[2] chart designed so that the viscosity-temperature curve will be nearly a straight line. An example of the use of this chart is shown in Fig. 1. The two oblique lines indicate the upper and lower viscosity limits for oil recommended for direct-connected steam-turbine sets. It is assumed that, in the lower viscosity range, by approximately 43 sec Saybolt is meant a spread of ±1 deg. The slopes of such lines of different fractions of oils obtained from the same crude are approximately equal and the lines are therefore parallel, but the lines from the fractions of another crude although parallel to each other may have slopes different from those of the first-mentioned

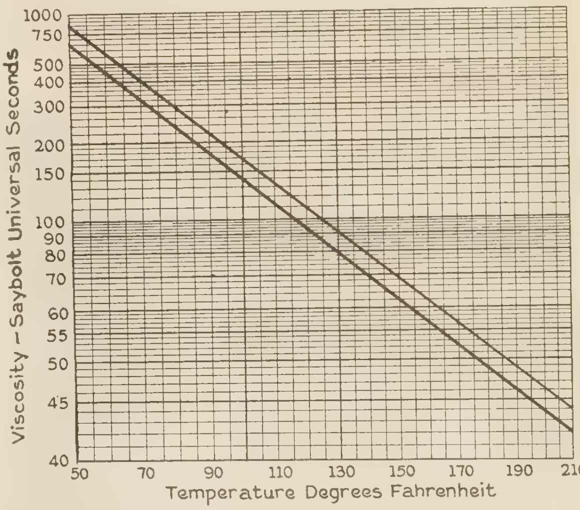


Fig. 1 Upper and Lower Viscosity Limits for Land and Marine Direct-Connected Turbine-Generator Sets

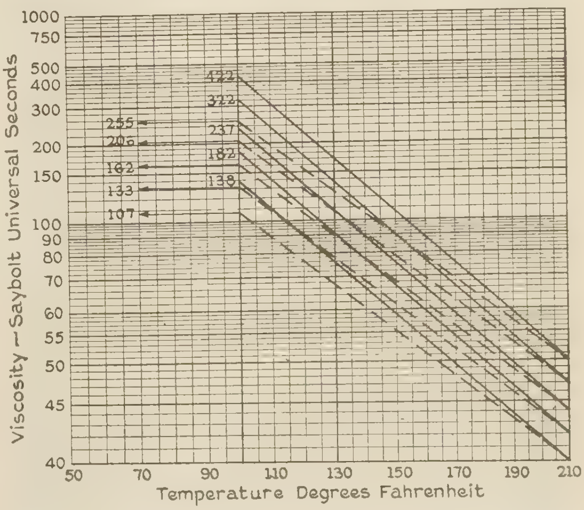


Fig. 2 Oil-Viscosity Chart

(Solid lines, oils of viscosity-index 0; dotted lines, oils of viscosity-index 100.)

[1] Works Chemist, General Electric Company.

[2] "Standard Viscosity-Temperature Charts for Liquid Petroleum Products," A.S.T.M. Standards, 1939, part 3, pp. 221–224.

Contributed by the Committee on Lubrication of the Machine Shop Practice Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

series. Pennsylvania oils have a gentler slope than do coastal oils.

In Fig. 2 is shown in solid parallel lines the viscosity-temperature characteristics of a series of oils derived from a special coastal crude and, in dotted parallel lines, similar characteristics of a series of oils derived from a special Pennsylvania crude. Dean and Davis[3] in 1929, gave to all the oils in the first-named series a viscosity-index number of zero and gave all those in the second series a viscosity-index number of 100. Then, to obtain the viscosity-index number of any other oil, they compared that oil with a zero-viscosity-index oil and a 100-viscosity-index oil in the following formula, all three oils having exactly the same viscosity at 210 F

$$\text{V.I.} = \frac{L - U}{L - H} \times 100$$

where

V.I. = Viscosity index of oil in question
U = Viscosity at 100 F of oil in question
L = Viscosity at 100 F of zero-index oil
H = Viscosity at 100 F of 100-index oil

Viscosity of Oils for Turbine Use

Since 1929, many oils, having a gentler viscosity-temperature slope than the Pennsylvania oils used in the formula, have been placed on the market. These new oils, therefore, have viscosity indexes of over 100. There are also oils on the market which have steeper viscosity-temperature lines than those of the series of coastal oils used in the formula. These oils have negative viscosity indexes.

This brings up the controversial question as to whether a high-viscosity-index oil is better than a low-viscosity-index oil in a turbine. While the tendency among steam-turbine operators is to use oils with high viscosity indexes, we know of successful operations with low-viscosity-index oil. So far as it is possible to judge from data obtained up to the present time from the field, either type of oil may be used, provided it is refined in such a way that it is satisfactory in the other respects cited in the recommendations.

The organic-acid content of an oil is expressed as the number of milligrams of caustic potash required to neutralize the acids in 1 g of oil.[4] The value obtained is called the neutralization number of the oil; the higher the neutralization number, the greater of course is the acid content of the oil.

Although the suggestion is made in the recommendations that a new oil may have a neutralization number as high as 0.05, many oils on the market have a neutralization number practically of zero. As an oil slowly oxidizes in use, organic acids are gradually formed and the neutralization number rises. Under exactly the same operating conditions, the rate at which one oil will oxidize may be quite different from the rate at which another oil will oxidize. Oils derived from some crudes seem to have in them natural oxidation retarders which slow down the rate of oxidation over what it would be if the retarders were not present. Then, too, a few oils on the market have oxidation inhibitors purposely added to them which, under selected conditions, bring down the oxidation rate to an exceedingly low value for long periods of time.

The effort displayed by many oil refiners in recent years to reduce their turbine-oil neutralization number down to zero is praiseworthy in so far as effort is concerned. However, there is evidence that many of these oils, with a neutralization number of zero (no organic acid present), in combination with water, have given rise to excessive iron corrosion within oiling systems, such corrosion products being carried along and interfering with the operation of governors and other parts depending upon close tolerances.

The corrosion product formed on iron parts in contact with a small amount of water disbursed through certain turbine oils is generally black and is magnetic. Examination of an X-ray diffraction pattern shows that ferro-ferric oxide, Fe_3O_4, is the principal oxide present in the corrosion product. Although not a true rust, it is called rust by most turbine operators.

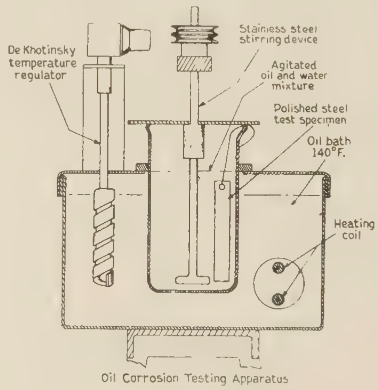


Fig. 3 · Oil-Corrosion-Testing Apparatus

Fig. 4 Modified Kuebler Apparatus for Testing Rusting Tendency of Oils

This corrosion has been so considerable in some cases that oil suppliers have been accused of furnishing oil which was prone to sludge, whereas, in such cases, it was probably too highly refined as far as retention of natural corrosion inhibitors was concerned. Of course, from an operator's point of view, a quantity of iron corrosion mixed with oil is a sludge, even though an oil man might not consider it as such. Moreover, a large amount of iron oxide in oil certainly will act as a catalyzer when the oil does start to oxidize. Then, an actual oil sludge may develop rapidly.

It has been noticed in recent years that new turbine installations with new turbine oil were the ones which sometimes corroded. Old ones with used oil never corroded. A turbine oil tends to improve in service, as far as corrosion inhibition is concerned.

Determining the Rusting Tendency of Oils

A simple laboratory apparatus has been devised by W. F. Kuebler[5] for determining the rusting tendency of oils. This ap-

[3] "Viscosity Variation of Oils With Temperature," by E. W. Dean and G. H. B. Davis, *Chemical and Metallurgical Engineering*, vol. 36, 1929, pp. 618–619.

[4] "Neutralization Number of Petroleum Products and Lubricants," A.S.T.M. Standards, 1939, part 3, pp. 617–619.

[5] Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

paratus consists of a beaker in which water may be held in suspension in the oil by agitation with a paddle. In this same beaker, polished-steel specimens are placed and the test is carried on for several days at room temperature. Many new oils on the market, when tested in this apparatus, permit water-rusting of the steel specimen, and these same oils in service have also not prevented rusting in some turbine installations. The only modification that we have made in this apparatus is indicated in Figs. 3 and 4. The change consists of surrounding the beaker with an oil bath running at 140 F, in order to simulate more nearly the temperature conditions of the oil in actual use.

With this apparatus, several remarkable facts have been developed concerning the oil which we use in the turbines in our Schenectady Works powerhouse. We found that the new unused oil will, when agitated with water, allow rust to form on iron, whereas, oil which has been in use in the same turbines for several years and which has a neutralization number of 0.7 will inhibit rusting.

We also found that this used turbine oil is a very effective rust inhibitor when it is mixed with new oil in proportions even as low as 1 per cent of used oil to 99 per cent of new oil.

With new oils which are not rust inhibited, we would suggest that, provided the oil supplier approves, a 10 per cent addition of

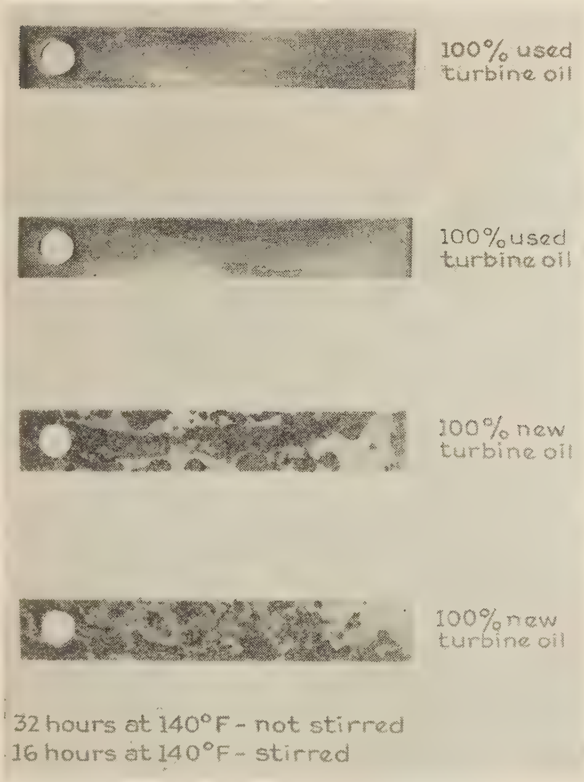


Fig. 5 Metal-Corrosion Specimens

a well-used oil of the same brand be added to the new oil when first putting a new turbine into service. Another sound piece of advice, relating to starting up a new turbine, is to limit water entrained in the oil to a minimum.

In Fig. 5 are shown two steel test pieces which failed to rust in 100 per cent used turbine oil and two test pieces which rusted severely in new oil of the same brand when tested in the modified Kuebler apparatus.

Theory of Rust-Inhibiting Quality of Oil

In our laboratory, we have found that the rust-inhibiting quality of turbine oil, developed in service, is in some way connected with the formation of hydrophilic groups by oxidation of the oil. Following Dr. Langmuir's procedure with oil films on distilled water, we found that new turbine oils which permitted iron corrosion in our laboratory tests did not spread when dropped on water, whereas, a drop of the same type of oil used for several years in our factory turbines spread over the surface of water like a flash, Fig. 6.

All the new oils on test, not giving rise to rust, spread on water

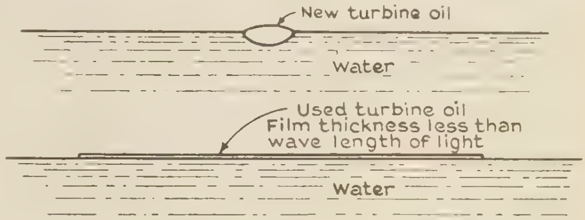


Fig. 6 Spread of New and Used Oils on Distilled Water

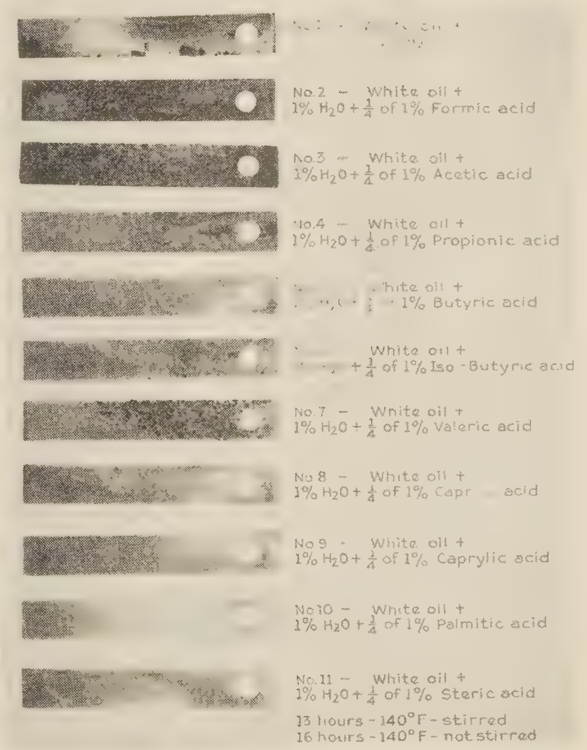


Fig. 7 Corrosion Effect of Organic Acids on Steel Specimens

to a greater or lesser degree. It requires 5 min for the drops of some of these to double in diameter, while others spread almost as rapidly as used oils.

Since, theoretically, organic acids of all manner of molecular weights are produced when mineral oils are oxidized, we carried out the experiment of adding 0.25 per cent of various aliphatic acids to a neutral white oil, not with the idea that these particular acids are formed by oxidation in the oil, but solely to provide a clew as to what molecular-weight organic acids were giving rust

inhibition, and what molecular-weight acids might give rise to rusting. It was found, as might be expected, that the acids of lower molecular weight, such as formic, acetic, and even propionic, were rust producers. In fact, more rust was produced by such acids than by a neutral untreated white oil. Ascending the scale of acids, complete protection was found through the use of 0.25 per cent caproic acid, having a molecular weight of 116. All acids of higher molecular weight also give protection. Later experiments indicated that 0.1 per cent of caproic acid was equally effective.

Fig. 7 shows a series of ten steel-corrosion specimens, indicating the relative rusting and rust-inhibiting qualities of 0.25 per cent of various organic acids in white medicinal oil. All tests were carried out at 140 F, the oils with 1 per cent water mixture having been stirred for a total of 16 hr; and left in a quiescent state for a total of 13 hr. The results are given in Table 2.

TABLE 2 RESULTS OF CORROSION TESTS ON STEEL SPECIMENS SHOWN IN FIG. 7

	Carbon atoms in acid	Molecular weight of acid	Percentage of surface rusted
No acid			5
Formic acid	1	46	100
Acetic acid	2	60	100
Propionic acid	3	74	50
Butyric acid	4	88	20
Valeric acid	5	88	2
Caproic acid	6	116	0
Caprylic acid	8	144	0
Palmitic acid	16	256	0
Stearic acid	18	284	0

Fig. 8, which gives the structure of one of the forms of caproic acid, illustrates how caproic acid or any organic acid of higher

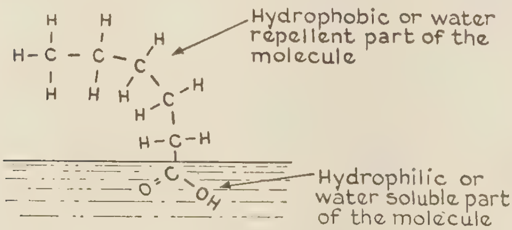


FIG. 8 ACTION OF CAPROIC ACID AS A CORROSION INHIBITOR

molecular weight tends to inhibit iron corrosion in contact with water suspended in oil.

The water-soluble or hydrophilic part of the molecule of organic acids is called the carboxyl group. This is the same group which attaches itself to iron surfaces. After a layer of such molecules has become attached to the iron, the oil-soluble or hydrophobic parts of the molecules are presented to the oil and prevent water globules, suspended in motion in that oil, from coming in contact with the iron, thereby, inhibiting rusting of the iron.

OTHER SOURCES OF RUST INHIBITION

Organic acids are not the only oxidation products of oil which inhibit rusting of turbine oils since, while the organic acids are being formed, there are probably compounds with hydroxyl groups and ketone groups which are likewise hydrophilic and capable of wetting the iron. It is therefore possible to treat a new oil with the nonacid oxidation products of mineral oil and, thereby, effect iron-corrosion inhibition without raising the neutralization number.

Because the traces of organic acids and associated oxidation products in turbine oils tend to inhibit rusting under turbine-operating conditions, it must not be assumed that a used turbine oil, for example, can be used as a rust-inhibiting coating for iron or steel under all conditions. Such an oil-treated specimen if placed out in the rain or in salt water would certainly rust.

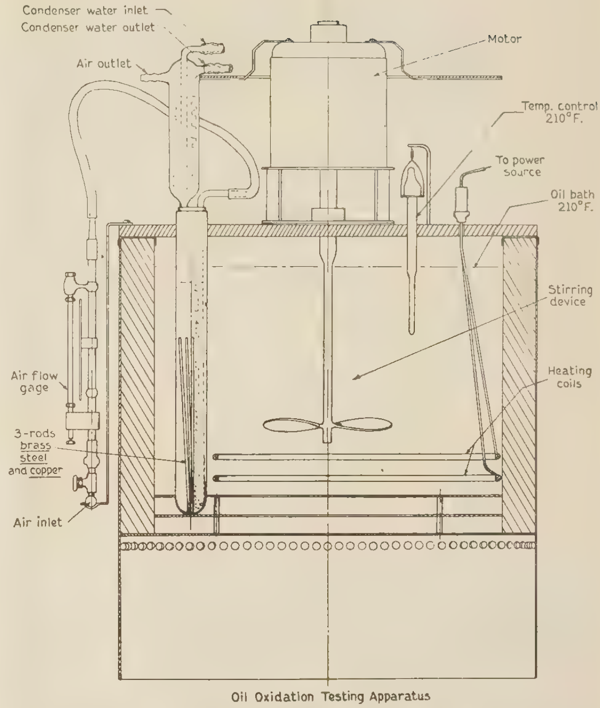


FIG. 9 OIL-OXIDATION-TESTING APPARATUS

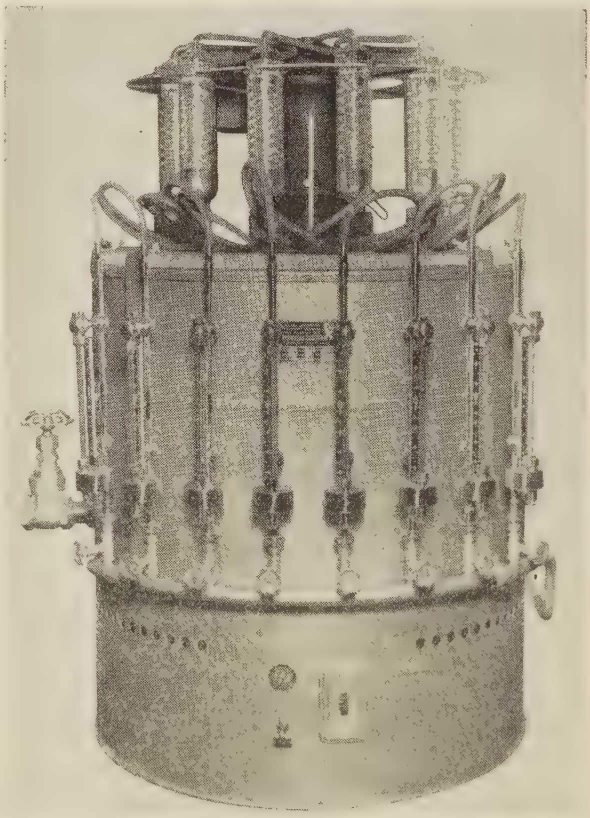


FIG. 10 APPARATUS DEVELOPED TO DETERMINE RELATIVE OXIDATION RATES OF VARIOUS OILS

A possible explanation for this may be that, although the protective molecules on the iron are packed closely enough to keep water in fine globules in suspension in oil away from the iron, large and heavy globules not in rapid motion will tend to break through the film.

Another point to be seriously considered is that too high a content of free organic acids in a turbine oil is detrimental, in that it may give rise to or be associated with:

1 The formation of metallic soaps.
2 Formation of permanent water emulsions.
3 Sludging of the oil.
4 Destruction of oxidation inhibitors.

The maximum permissible amount of free organic acid in an oil depends upon the type of oil used.

With oils having antioxidants present, the amount of acid permissible should probably not exceed that which corresponds to a neutralization number of 0.15. With no antioxidants present and depending upon the oil used, an amount equivalent to a neutralization number of 2.5 might be permitted.

The carboxyl groups which, in an organic acid are responsible for inhibition of rusting, are also responsible for attack on metals and oxides of metals forming soaps which are almost insoluble in both oil and water. Most of these metallic soaps act as active catalyzers for the further oxidation of the oil.

A high percentage of organic acids and associated oxidation products give rise to permanent emulsions of water in oil because there is a sufficient number of the hydrophilic groups present to take care of the large contact area between the water globules, which constitute the discontinuous phase of such an emulsion, and the oil which constitutes the continuous phase.

High acid values in oils are associated with sludging of such oils. Such sludges are insoluble oxidation products of the oil.

Farmer Test for Oxidation Characteristics of Oil

One of the best tests for the oxidation characteristics of an oil has been developed by Harold Farmer[6] of the Philadelphia Electric Company. He has pointed out that iron, brass, and copper, particularly the last, are active catalyzers in speeding the oxidation of turbine oil.

Because the American Society for Testing Materials has no test method for finding the relative oxidation rates of various turbine oils, in the Schenectady Works laboratory, we are using a modification of the Farmer test; 400 ml of oil are placed in a pyrex tube with three $^1/_8$-in-diam rods, 10 in. long, of three different metals, i.e., iron, brass, and copper. In the same tube at the start of the test is placed 12 cc of water. Examination of the tubes is made daily to assure that approximately that amount of water is present. The oil and water are maintained at 210 F and, through the oil and water, air is bubbled at the rate of 10 l per hr. Samples of oil are tested once a week for neutralization number, viscosity, and precipitation number. The viscosity sample is returned to the tube and oil lost in the determination of the neutralization number is replaced by the addition of new oil to that under test in the tube. The apparatus is shown in Figs. 9 and 10.

While we find that wide variations exist in the oxidation rate of different oils, at present there is no inclination to condemn some oils which do not prove to be among those giving the best results in this apparatus. The oil, for example, which we are using successfully in the Schenectady Works turbines cannot be graded as excellent from the oxidation standpoint. On the other hand, we appreciate that the oxidation-inhibited turbine oils now on the market are a valuable contribution to the turbine industry.

[6] "Copper Catalysis Accelerates Turbine Oil Oxidation," by Harold Farmer, *Electrical World*, vol. 111, May 20, 1939, pp. 1452–1453 and 1519.
T. H. Rogers and B. H. Shoemaker, *Industrial and Engineering Chemistry*, Analytical edition, 1934, vol. 6, p. 419.

Discussion

M. D. Baker.[7] The difficulty in writing specifications for the purchase of turbine oils can be fully appreciated. As the author points out, specifications of a very general nature can be made, and generally these do not give information that will predict the service life of the oil to be purchased.

During one of our investigations, twelve light turbine oils of different brands were studied in the laboratory. All twelve oils were found to be within the limits of the specifications which included tests for gravity, flash, fire, viscosity, and steam emulsion. No life tests were made. Six of these oils were picked for an accelerated life test in a small unit. These accelerated tests gave service records varying from 50 hr on the poorest oil to 1800 hr on the best oil. The life of the oil was determined by the length of time required to reach a neutral number of 0.8.

Refining methods have been changed to improve the quality of the oils delivered. In conjunction with this improvement, antioxidants have been added to the oils to retard the rate of oxidation and prolong their service life. An example of this increased life can be illustrated by the service record of one oil. Without the antioxidant after 13,000 service hr, the neutral number was 0.85. This same oil with the antioxidant added now has 37,000 service hr and a neutral number of 0.08. The addition of the antioxidant has decreased the rate of sludge precipitation in the oil coolers, so that now 15,000 hr elapse between cleanings where previously 5000 hr was considered a long period of time. This lengthening of time between cleanings has given longer periods of metal passivity in the coolers and has reduced the rate of metal poisoning.

The detrimental side to the addition of antioxidants must also be considered. The author mentions that in the older types of oils the neutral number of 2.5 could be carried with safety, but that in the newer types of oils one company has set a limit of 0.15 as the maximum neutral number that can be carried with safety. The older type of oils increased in neutral number at a more or less accelerated but definite rate with no sudden or rapid increases. Knowing the condition of the oil, its life could be predicted with some degree of accuracy when the rate of rise of the neutral number was plotted against the service hours.

The accelerated-service test on one oil containing an antioxidant gave the following results: At 1250 service hr, the neutral number was 0.08; at 1394 service hr, the neutral number was 17.4. This rise in neutral number when it occurred was very sudden and very rapid. So, unless some test other than neutral number is devised and used, the life of an oil containing an antioxidant cannot be accurately predicted.

The neutral number of 0.8 was set for the top limit of the oils at Springdale Station, as experience has shown that, when an oil reached this value and under normal operation with no water present in the oil, any water contamination would create a heavy sludge. Units having constant moisture infiltration into the oil system have been operated safely with neutral numbers as high as 4.0 but this is too high for a unit that does not have regular or frequent infiltration of water.

Corrosion in turbine systems has been experienced for years,

[7] Springdale Power Station, West Penn Power Company, Pittsburgh, Pa.

but it has only been during recent years that it has been an item of great importance. As the newer oils are more resistant to oxidation, compounds that protect the metal from corroding are slower in forming; and corrosion will occur, where previously it was not experienced. Also, the newer oils will dissolve films that have been coated upon metal surfaces and thus allow the water to contact the bare metal and cause corrosion. Previously, the corrosion products formed were principally ferric oxide (Fe_2O_3), which, when carried in the oil stream, did not cause severe abrasion. The product formed with the newer oils is magnetic iron oxide (Fe_3O_4), which when formed in the presence of oil appears to be small, dense, hard crystals, and very abrasive.

When corrosion occurred in the high-pressure unit at Springdale, it was impractical at that time to remove the unit from service for a prolonged period and means for preventing the corrosion had to be devised. Laboratory tests showed that the addition of used oil which had become partially oxidized would prevent corrosion. This remedy was applied to the oil in the unit, and corrosion was completely stopped. Before adding the used oil, moisture to the extent of 0.1 per cent would cause corrosion; after the addition of the used oil, the unit was operated for 3 weeks with 3 per cent of water in the oil, and no corrosion occurred. Since these results have been obtained at Springdale, it has become a general and successful practice to stop corrosion in turbine oil systems by the addition of used oil to the new oil.

We differ from the author in our theory as to why the addition of the old oil prevents corrosion. Laboratory experimental work and service records indicate that, when water is added to a used oil, it makes a coating of a very minute film on the metal surface. The first water added causes the formation of this film, but in so doing it also removes the film-forming properties from the oil. So until the oil is further oxidized any other additions of water can cause corrosion of new metal that was not in the system when the protective film was formed. Several experiences with corrosion on changed metal parts has lead us to believe that the inhibition was imparted to the metal and was not permanently given to the oil, by having imparted to the oil an increased wetting action.

The addition of the old oil to the new oil in a turbine system is ideal for corrosion prevention but is not to be desired when considering the service life of the new oil. The used oil apparently serves as a catalyst to start oxidation in the new oil and reduce its service life. Laboratory tests have shown that the life may be reduced to as much as 20 to 30 per cent of the life expected if the used oil had not been added.

Turbine-lubrication problems and their solutions can be arrived at only by the cooperation of the turbine designers, the oil refiners, and the users of the oil. Committees composed of these three groups have been formed, and the results of their cooperative studies will be very valuable.

D. L. Barbour.[8] Operators of geared turbines would be interested in the author's comments concerning the desirability of using for this service, as contrasted with direct-connected units, an oil having a low viscosity index, i.e., a relatively steep temperature-viscosity curve.

For a geared unit, it is general practice to employ a common oiling system to serve the bearings of the turbine, reduction gear, and generator or other driven machine and also the oil sprays for gear-tooth lubrication. The operating conditions of the bearings, particularly those of the turbine and high-speed pinion, are the same as on a direct-connected unit. The viscosity range for an oil most suitable for the bearings is as recommended

[8] Chief Turbine Engineer, Elliott Company, Jeannette, Pa. Mem. A.S.M.E.

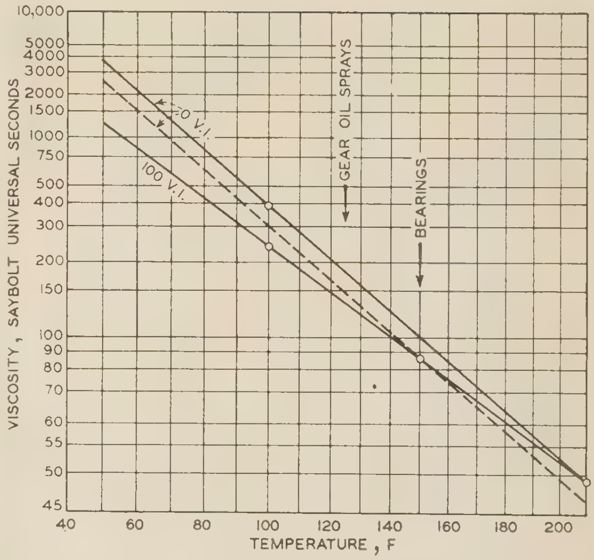


Fig. 11 Temperature-Viscosity Curves

(Oil viscosity index 100, and 49 sec Saybolt Universal Viscosity at 210 F, and oil with zero viscosity index, same Saybolt Universal Viscosity and temperature.)

in the first column in Table 1 of the paper. On the other hand, an oil of much higher viscosity is preferred for gear-tooth lubrication. This necessarily results in a compromise viscosity, intermediate between the two, as recommended in the second and third columns of Table 1.

There is also a considerable difference in oil-temperature rise in the bearings and in the gear teeth. While the bearing-outlet oil temperature is generally between 140 and 160 F with 120 F inlet, the oil leaving the gear teeth may be at only approximately 125 F.

Assuming (1) an oil with a viscosity index of 100 and 49 sec S.U.V. at 210 F and (2) an oil with zero viscosity index and the same S.U.V. at 210 F, the temperature-viscosity curves are as shown by the solid lines in Fig. 11 of this discussion. For the 100 V.I. oil, the viscosities at a bearing-outlet temperature of 150 F and the gear-tooth oil temperature of 125 F are, respectively, 86 and 132 S.U.V. The corresponding viscosities at the same operating temperatures for the zero V.I. oil are respectively 100 and 180 S.U.V. Or similarly, as shown by the broken line, a zero V.I. oil with the same viscosity as the 100 V.I. oil, at the bearing operating temperature of 150 F, would have 146 S.U.V. at the gear-tooth oil temperature of 125 F, compared with 132 S.U.V. for the 100 V.I. oil at this temperature.

By taking advantage of the steeper viscosity-temperature curve of the zero or low-index oil, a somewhat better compromise can be realized for the desired different viscosities for the bearings and gear teeth at their respective operating temperatures. It is questionable, however, whether this advantage is of sufficient importance to justify the recommendation of low-index oil for geared units.

By way of corroborative operating experience in connection with the corrosion-inhibitive value of used turbine oil, as compared with 100 per cent new oil of the same grade, the writer's company has had several cases of corrosion of steel parts during the first few weeks of operation of new units with all fresh high-grade oil. In each instance there were traces of water in suspension in the oil, probably from gland leakage, although the leakage and the clearance in the oil baffles were not excessive. It has been found in most cases that, with no change in operating

conditions, the tendency toward corrosion of steel parts exposed to the oil gradually disappears, probably as a result of the oil developing corrosion resistance from a gradual increase in organic-acid content of high molecular weight as described by the author.

Ronald Bulkley.[9] The author displays a commendable understanding of the difficulty of fashioning an oil in such a manner that it shall possess the conflicting properties of separating out quickly from water on the one hand, and of preventing corrosion by attaching itself firmly to metal, on the other. In addition to performing this chemical legerdemain, a good turbine oil is expected, as a matter of course, to be neutral as regards acidity and practically everlasting as regards oxidation stability.

The statement regarding the amount of free organic acid permissible in a turbine oil before withdrawal is open to some question. Most antioxidants are of the type which are effective only in oils which have been overrefined either by acid or by solvents. It is universally recognized that such oils are basically unstable, and the rate of oxidation is alarmingly rapid when the induction period is ended. It is logical, therefore, to require the immediate withdrawal of the oil as soon as the exhaustion of the antioxidant is made apparent by the upward movement of the acidity value. The maximum limit suggested by the author for this type of oil is 0.15, a very low figure.

Oils which contain no added antioxidant are, as a rule, much more stable products than the base oils selected for the addition of antioxidants. The rate of deterioration of such oils is slow and they may safely be allowed to develop a much higher neutralization number before withdrawal. The limit set in this paper is 2.5.

It has been found possible in some instances, however, to combine an effective inhibitor with the type of oil which possesses a high degree of stability of its own. In these cases the rate of deterioration after the inhibitor is exhausted should, theoretically, not be any higher than the rate for the base oil without the inhibitor. As a matter of fact, in the several field installations with which we have had experience the rate of oxidation, after the so-called "break," has been lower than for the base oil alone in the same turbine. This is perhaps explainable on the supposition that the first reaction product which the inhibitor forms is itself a compound capable of retarding the rate of oxidation somewhat, although not able to suppress it entirely as a true inhibitor does. However this may be, the net result is that, for such combinations of oil and antioxidant, the neutralization number may safely be allowed to rise to values at least as high as would be permitted for the base oil without the antioxidant.

B. F. Hunter.[10] The results given in this paper confirm investigations of the corrosion problem in new turbine units, carried out over a period of several years in the research laboratories of the writer's company. Turbine oils, refined to minimize oxidation in service and to separate readily from entrained water when new, show little if any tendency to retard corrosion with small percentages of entrained water. In service, partially oxidized oil and perhaps formations of small percentages of metallic soaps appear to adhere to the surfaces of exposed metal, providing a thin protective film impervious to moisture. Our investigations have indicated definitely that as little as 0.02 per cent of high-molecular-weight fatty acids added to highly refined turbine oils would prevent entrained moisture from corroding metal surfaces during the initial breaking-in period of a turbine unit.

[9] Research and Development Division, Socony-Vacuum Oil Company, Inc., Paulsboro, N. J.

[10] Gulf Oil Corporation, Pittsburgh, Pa.

Actual field experience for the last several years has proved this theory to be correct. Such oils are generally available and are recommended for the initial fill for new units. Conventional or regular turbine oils are recommended as subsequent make-up oil. No corrosion difficulties have come to our attention with any turbine oil which has been in service for a period of at least 6 weeks, indicating that special inhibited or compounded turbine oils are not required for turbines other than new units during the breaking-in period for the first few weeks of service.

Our investigations have likewise indicated that from 10 to 15 per cent of old or used turbine oil added to new turbine oil is equally satisfactory as a corrosion preventive for new units.

The advice given by the author should aid in safeguarding new turbines in their early stages of service. Emphasis should be placed on the use of special compounded oils or a mixture of old and new oils for new units during the breaking-in period, since corrosion is unknown in turbine units after a few weeks or months of operation.

F. C. Linn.[11] The author has given a clear presentation of some of the problems involved in the lubrication of steam-turbine sets. There has been considerable activity during the last year by various oil companies, turbine manufacturers, and central-station chemists in the development of an apparatus whereby it will be possible to predict in a relatively short period of time whether an oil with water present will permit rusting to take place. The apparatus as described in this paper, which is a modification of a design developed by W. F. Kuebler, appears to meet with the approval of most of the investigators.

The problem of rusting of metallic parts in the oiling system is a serious one since it is found that, in many units, rusting takes place during the first few days or months of operation and in some cases after years of operation. For proper lubrication, few oil companies are in a position to supply an inhibited oil which will pass the oil-corrosion test as outlined by the author.

This problem is also prevalent in apparatus other than steam turbines. Rusting occurs in the case of journals, oil rings, etc., in motor bearings in localities where, because of climatic conditions, a certain amount of condensation takes place in the bearing housings.

It is true that rusting can take place only when moisture is present in the oil. It is also true today that in most installations there is less water finding its way into the oiling systems than was the case in former years. Yet the rusting problem is more serious today than it was in the past.

The author suggests used oil mixed with new oil as a lubricant for a new installation in order to inhibit rust. However, this is a practice which will decrease the life of the oil. Since there are oils on the market which have excellent oxidation, as well as rust-inhibiting characteristics, it would be advisable to purchase such oils for use in new turbine sets.

It is realized that proper maintenance of the oiling system is essential to extend the life of the oil. Every precaution should be taken to keep the oiling system free from dirt, water, and all foreign matter, even though oils are used which contain antioxidants and rust inhibitors. This will insure continuous and satisfactory service from the oil. Operators should follow instructions in the care of the oil provided by the oil company's representative. If this is done, there is reasonable certainty that troubles due to rusting, sludging, etc., which now cause considerable inconvenience and service cost, will be reduced to a minimum.

[11] Turbine Engineering Department, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

Author's Closure

The author has never been fortunate enough to observe the phenomenon which Mr. Baker mentions, namely, that when water is added to a used oil, it makes a coating of a very minute film on the metal surface. He has, however, watched carefully the behavior toward steel test samples of globules of water in suspension in two types of oils: (*a*) those containing no rust inhibitors, and (*b*) those containing rust inhibitors.

In oils having no rust inhibitors present, a suspended water globule coming in contact with a vertical steel surface sticks immediately to that surface. Agitation of the oil or the steel specimen will not dislodge the adhering drop of water. This drop generally does not spread much more than its own diameter over the surface of the steel.

In oils having rust inhibitors present, particularly rust inhibitors containing hydrophilic groups, globules of water in suspension coming near the vertical surfaces of steel test specimens seem to bounce away from such specimens. It seems very probable that the hydrophilic polar groups not only form a protective film on the surface of the steel specimens, but also put a "plating" of oil over the entire surface of each globule of water, making each globule substantially a globule of oil as far as its effect on the steel specimen is concerned.

With regard to Mr. Barbour's request for comments concerning the desirability of using oils having a low viscosity index for geared turbine, the author would rather not give an opinion.

The combination of an effective oxidation inhibitor with a type of oil which has a high stability of its own, should find favor with many turbine operators. Such a combination would not only require less accurate control through determinations of neutralization numbers, but also might take care of cases where the oxidation inhibitor might be accidentally destroyed because of factors other than oxidation, such as, for example, an acid-containing atmosphere in a powerhouse.

Stress and Deflection in Reciprocating Parts of Internal-Combustion Engines

By R. L. Boyer[1] and T. O. Kuivinen,[2] Mount Vernon, Ohio

A consideration of stresses and deflections in reciprocating parts of internal-combustion engines is presented in this paper, from the viewpoint of the heavy-duty-engine builder. The authors give suggested rules for design of pistons, piston pins, connecting rods and piston rods, bolts and connecting-rod caps. All recommendations given are based on what has proved to be successful practice in this classification of machinery.

The following discussion has to do with methods for attacking problems of stresses and deflections in reciprocating parts of internal-combustion engines. The viewpoint is largely that of the builder of heavy-duty machinery designed for continuous operation. It should be pointed out that the figures given do not necessarily apply to high-speed automotive-type engines. It will be noted that in many cases practical rather than theoretical considerations determine the design of a part. Whether this is true or whether the part is such that it can be logically and mathematically analyzed, successful and unsuccessful experience determine the factors to be employed. This paper deals not only with stress but also with deflection.

However, it will be noted that in all cases values are given for stress and not for deflection, because it is impractical to give deflection limits in many cases. Furthermore, the proper stress allowance is made responsible for keeping the deflection within allowable values. The theoretical desired deflection in most cases would be zero, which is, of course, impossible and, therefore, finite amounts of deflection must always be permitted.

With the increase of speeds, which is always a consideration of the engine builder, the weight of reciprocating parts becomes of greater relative importance. In order to secure the minimum weight and yet retain the maximum rigidity and allowable stress, careful design is necessary and most rule-of-thumb methods have had to be discarded.

Reciprocating engines fall generally into classes, i.e., trunk-piston and crosshead types. Greater emphasis is placed on the trunk-piston engine in this paper, although crosshead types are also considered.

Pistons

Cast iron has been and still is, of course, the predominating material for pistons of internal-combustion engines. However, aluminum alloys have been used to a considerable extent and, particularly, as speeds are increased, they are finding wider application. Weight saving is the most important factor in determining whether lightweight alloys need be used, although heat conductivity must also sometimes be taken into account. The determination of the point in speed at which lightweight alloys should be used is governed mostly by stress in the connecting-rod bolt and by over-all balance. In the case of most multicylinder engines, the over-all inertia forces and couples are balanced out to zero but, in the heavy-engine industry, there are many forms of machines built which are not in themselves inherently balanced, and lightweight alloys are of importance in reducing these forces considerably.

It is not the purpose of this paper to enter into details of piston design, as that in itself could quite readily become the subject of a separate discussion. However, since the piston is an important element in the reciprocating masses, the general design will be touched upon. Various combustion systems and general engine designs call for various shapes in the piston head and these shapes, of course, vitally affect the stress and deflection in the top of the piston. No attempt is here being made to take into account the variations in stress and deflection due to these shapes. The head of a piston should logically be considered as a uniformly loaded plate with fixed edges. If it were freely supported, the maximum stress would always be at the center; but considering the plate fixed at the edges, most authorities agree that the maximum stress occurs at the edges. Taking all of the theories into consideration, it has been generally accepted in recent years that the ideal design for the head of a piston is that of uniform thickness. For uniform thickness and a uniform load with fixed edges the maximum stress at the edge becomes

$$S = \frac{3wr^2}{4t^2} \qquad [1]^3$$

and the stress at the center becomes

$$S = \frac{3wr^2}{8t^2}(1 + m) \qquad [2]$$

where S = tension stress, psi; w = distributed load, psi; r = radius of plate to point of fixation, in.; t = thickness of plate, in.; and m = Poisson's ratio, approximately 0.3.

Aside from mechanical-stress considerations, however, the heat-flow factor is probably of greater importance. The shape of the piston head and its point of connection into the side walls must be such as to permit heat flow readily without creating concentrated stresses. It is difficult to state a definite rule for the thickness of piston heads from the standpoint of heat flow because each design is a distinct problem in itself, being dependent upon such factors as compression pressure, cycle, operating speed, type of service, horsepower, and ring setup. In aluminum alloys, it is also necessary to consider whether the piston is a forging, a sand casting, a semipermanent mold casting, or a full permanent mold casting. Taking this into consideration, in addition to mechanical stresses, the thickness of the piston head has finally become a rather empirical value secured through experience.

The following values of t_1, shown in Fig. 1, have been found quite successful:

[1] Chief Engineer, The Cooper-Bessemer Corporation.

[2] Assistant to Chief Engineer, The Cooper-Bessemer Corporation.

Presented at the 1940 Conference of the Oil and Gas Power Division, Asbury Park, N. J., June 19–22, of The American Society of Mechanical Engineers.

[3] Grashof's formulas as given in "Handbook of Engineering Fundamentals," by O. W. Eshbach, John Wiley & Sons, Inc., New York, N. Y., 1936. Also "Applied Elasticity," by S. Timoshenko and J. M. Lessells, Westinghouse Technical Night School Press, East Pittsburgh, Pa., 1925.

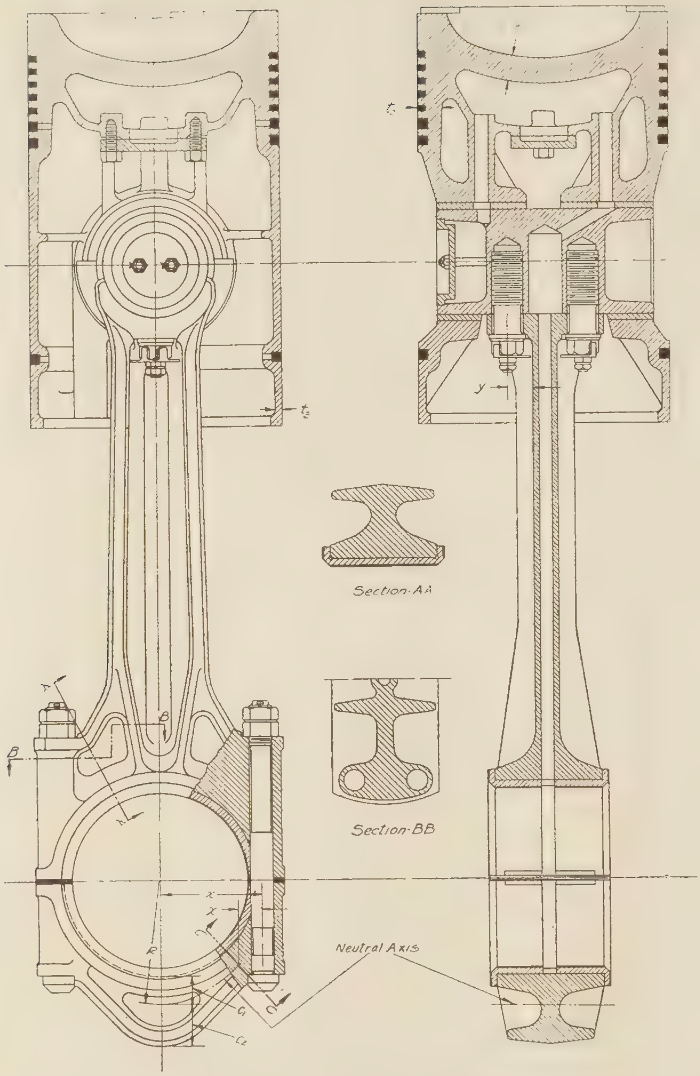


Fig. 1 Piston Connecting-Rod Assembly

Four-cycle Diesel, cast iron..................	t_1 = 0.11D to 0.13D
Four-cycle gas, cast iron.....................	t_1 = 0.12D to 0.14D
Four-cycle Diesel, aluminum..................	t_1 = 0.13D to 0.16D
Two-cycle Diesel, cast iron..................	t_1 = 0.16D to 0.18D
Two-cycle gas, cast iron.....................	t_1 = 0.20D to 0.22D

where D = piston diameter, in. These values apply only to non-cooled pistons. The thickness of cooled piston heads may and should be materially less than the foregoing.

It is necessary to retain a considerable thickness of metal where the piston head joins the ring belt, but this may then, of course, taper considerably toward the lower rings. The total thickness of metal at the lower compression ring should be approximately 0.10D for cast iron, while a factor of 0.13 is more nearly correct for aluminum. In the latter case, the increased thickness is for the purpose of relieving stresses rather than providing heat flow (see t_2, Fig. 1).

The skirt thickness, provided it is well ribbed against distortion, is mostly dictated by foundry considerations. The common factor to be used for the thickness of the lower end of the piston skirt, t_3 Fig. 1, exclusive of any ribbing, is 0.025D to 0.035D. It need not be mentioned that sudden changes of section in the stressed portion of pistons should not be tolerated and that proper support of the load on the piston-pin bosses is of the utmost importance. It will be noted that, in practically all modern cast-iron pistons, some form of baffle plate is iserted beneath the piston head. This has been found to be practically a necessity in cast-iron pistons, because the piston head frequently runs at such a temperature as occasionally to cause crankcase explosion. This baffle completely seals the crankcase from the hot head. Of course, with aluminum pistons such a baffle is unnecessary, because of increased heat-flow characteristics and resulting low temperatures.

Wristpins

The term wristpin does not necessarily refer to piston pins for trunk pistons only, but also includes crosshead pins. There are three general types of wristpins used, namely, tight in the piston, full floating, and tight in the connecting rod. Minimum deflection is desired to maintain uniform distribution of load on the wristpin bearing. Some types of wristpin constructions result in quite high bearing pressures, and this uniform distribution of load is necessary in order to avoid areas of highly concentrated bearing pressures and subsequent damage to the wristpin bearings.

However, from a stress standpoint, it is desirable to determine the bending stresses by the most severe consideration of loading. This would be to consider the entire peak cylinder-pressure load as concentrated at the center of the wristpin, with the wristpin considered as a simple supported beam, and the reactions approximately at the longitudinal center of the pin extension into the piston bosses. Bending stresses and shear stresses from standard beam formulas can be determined and combined to give the maximum stress. It is suggested that the allowable stress be based upon a factor of safety of approximately 10

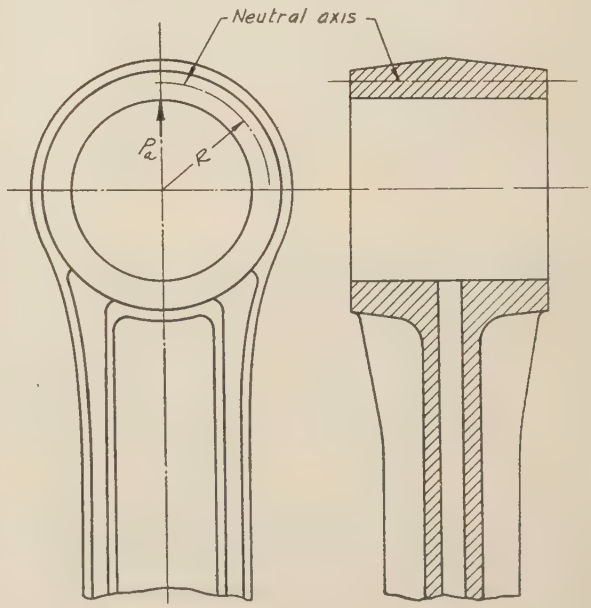


Fig. 2 Eye-Type Rod End

on the ultimate strength of the material used. Pins so proportioned will usually be found to have sufficient stiffness.

When the wristpin is tight in the connecting rod, the stress and deflection are somewhat less than the results determined by the foregoing method of calculation, due to the supporting effect of the connecting rod. It is advisable, however, to calculate all types of construction by the severest consideration of loading.

A successful method of obtaining rigidity in wristpins, and lightweight reciprocating parts, so as to obtain uniform loading on the wristpin bearing is illustrated in Fig. 1. This type of construction is best applied to single-acting engines or compressors and not to double-acting machines, for in the latter it becomes difficult to obtain sufficient bearing area for the return stroke. It is also particularly applicable to two-cycle engines and to four-cycle machines where the gas-pressure load is always higher than the inertia load. The piston or crosshead end of the connecting rod is saddled and bolted to the wristpin. The thrust sides of the bosses in the piston (or crosshead) are extended almost to join at the center. A much greater bearing area is obtained on the thrust side. The bushing is cut out on the lower side to provide space for the saddle part of the connecting rod. The wristpin "beam length" is considerably shortened, thus reducing bending stress and deflection.

Connecting Rods

The wristpin eye of the solid-eye type of connecting rods, wherein the eye is round (not wedge-adjusted wristpin bearings) may be designed as follows for stresses and deflections.[4] This method gives lightweight construction without excessive bending stresses. The method to be outlined has been actually checked experimentally by the authors on model connecting-rod eyes and found to be quite satisfactory. It is realized that the method of calculation of connecting-rod eyes has always been a controversial subject. It is hoped that other viewpoints will be presented in the discussions.

Knowing a definite wristpin diameter, bearing-performance experience will enable the designer to choose a definite clearance between wristpin eye and the pin. The pull on the ring necessary to take up this clearance is

$$P = \frac{\delta EI}{0.137R^3} \quad \text{[3]}$$

where P = pull on the eye, lb; δ = reduction of eye diameter in a plane 90 deg to the load P, in.; I = cross-section moment of inertia; R = radius to neutral axis of ring cross section, in., as shown in Fig. 2. Then the bending moment at the point of load application is

$$M = 0.318PR \quad \text{[4]}$$

From the foregoing moment M, the bending stress can be determined by the formulas for curved beams described later in detail under "Connecting-Rod Bearing Caps."

[4] "Strength of Materials," by S. Timoshenko, D. Van Nostrand Company, New York, N. Y., 1930 edition, pp. 436–440.

The eye cross section should be small enough so that P, as determined by Equation [3], comes out considerably less than the actual load to be applied on the connecting-rod eye. Calling the actual load P_a, the direct tensile stress on the cross section of the top of the eye is

$$S_t = P_a/2A \quad \text{[5]}$$

where A = cross-sectional area of eye. The maximum tensile stress in the eye is the sum of the stress from the bending moment M, Equation [4], and the direct tensile stress obtained from Equation [5].

Large sweeping radii should be used to join the wristpin eye to the column of the connecting rod.

For the saddle type of wristpin design, the bolts on single-acting four-cycle engines must withstand the inertia force of the piston and wristpin (plus piston rod and crosshead on crosshead-type engines). On other engine types, the bolts must withstand the maximum load determined by analysis of the loading conditions. These loads are resisted by tensile stresses in the bolts and failure would cause considerable damage to the engine. Therefore

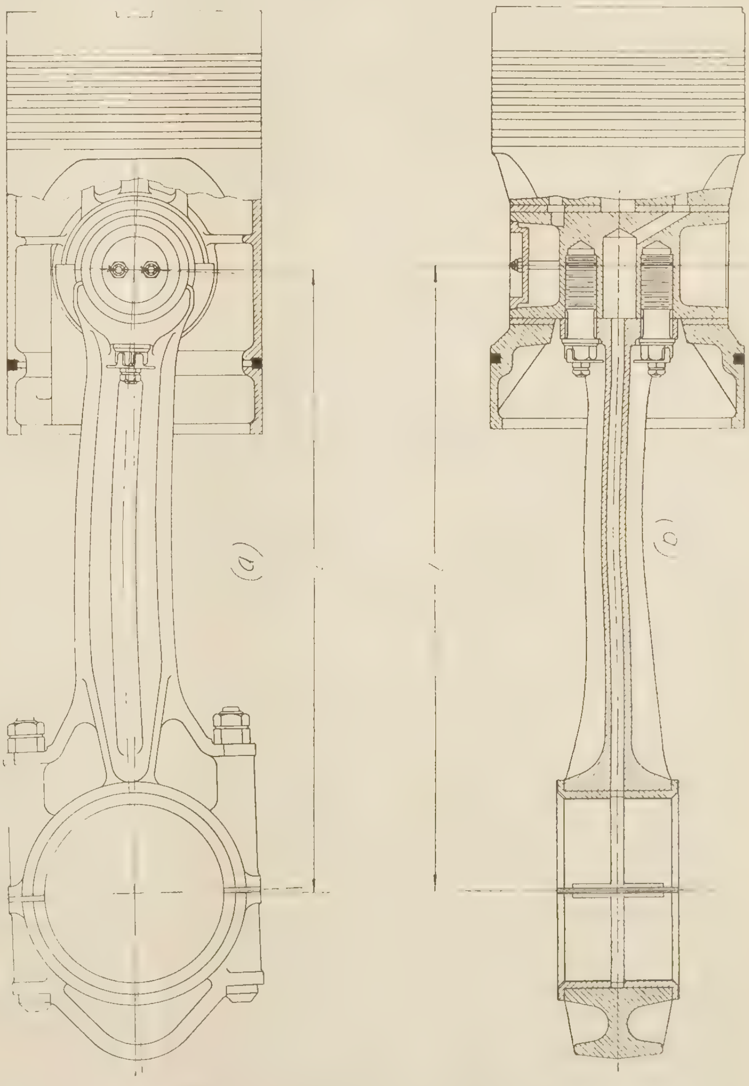


Fig. 3 Connecting-Rod Column Action

stresses must be conservative and material must be high-strength steel. A factor of safety of approximately 12 to 13, based on the ultimate strength of the steel, is recommended.

The wings of the saddle type of connecting rod, through which the wristpin bolts pass, are partly held at their sides. However, to simplify the method of calculating the depth required, it is convenient to consider them as simple cantilever beams with the bolt load as the applied load. The length of the cantilever beam is the distance y shown in Fig. 1. Bending stress at the junction of the cantilever and the main shank of the connecting rod should not be more than the ultimate strength divided by about 10.

For the main-column section of the connecting rod, three cross-sectional types are used: I-beam, round, and rectangular.

The minimum area of any cross section should be such that the compression stress per square inch of area from peak cylinder load is not more than the ultimate strength of the steel divided by approximately 7.

In addition to direct stress on the minimum cross-sectional area, the connecting rod is a column and should be designed for safe stress as a column. When bending in the plane of the crankshaft center line, both ends of the column are considered fixed, with the column length as the distance from the crankpin center line to the wristpin center line, Fig. 3(b). Rankine's formula for short columns has been found to be the most satisfactory, as very rarely is the l/r ratio of connecting rods greater than 120. When bending in a plane 90 deg to the crankshaft center line, the ends of the rod are considered free, Fig. 3(a). Thus, in the latter direction, the cross section of the rod must be designed with a greater radius of gyration r than in the other direction. Rankine's formula for short columns is

$$S_c = (P/A)[1 + k(l/r)^2] \quad \text{[6]}$$

where P = total thrust load on rod, lb; A = cross-sectional area at mid-point of rod, sq in.; k = 0.00004 for columns of fixed ends; k = 0.00016 for columns with free ends; l = length of rod from center line of crankpin to center line of wristpin, in.; r = radius of gyration of mid-point cross section in the direction that bending is being considered, in.; and S_c = column stress, psi. The latter value should not exceed the ultimate tensile strength of the material divided by a factor of safety 6.5.

Using this method of design, it is quite obvious that the maximum stiffness and strength, with the minimum weight of material, is obtained by the use of I-beam cross-sectional shapes with the long axis of the I-beam in the plane 90 deg to the crankshaft center line. On some types and sizes of engines, where a low rate of production does not warrant the expense of dies for forging the I-beam section, it is more economical to use either the round or rectangular shape.

The crankpin end of the connecting rod must be made stiff to prevent bending of the bolts holding the cap, and to prevent deflection of the crankpin-bearing shell. Most failures of crankpin-bearing bolts and some failures of crankpin bearings may be attributed to the lack of stiffness in this end of the connecting rod. It is impractical to state the maximum allowable deflection in this end of the rod as this must be judged mostly by experience obtained from previous designs. The ideal deflection would, of course, be zero which is obviously impossible.

The flanges of the I-beam of the connecting-rod shank should be extended right to the bearing-shell backing to help stiffen the support for the bearing, as shown in Fig. 1. The angular cross section on approximately a radius from the center line of a crankpin bearing to the curve where the rod flares out to form seats for the bolts should also be of I-beam cross section for greatest rigidity with minimum weight. The bolts should be fitted at the joint between the rod foot and the connecting-rod-bearing cap, or heavy closely fitted dowels should be used.

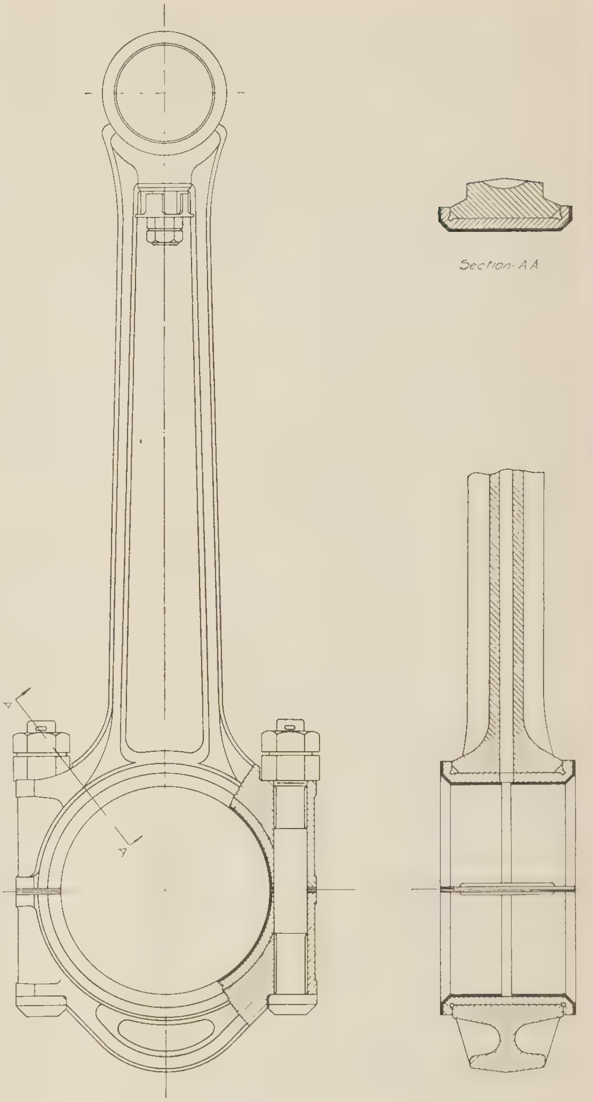


Fig. 4 A Weak Connecting-Rod Foot

A lightweight design which caused considerable difficulty in broken bolts, due to deflection of the rod foot, is shown in Fig. 4. A comparison of Fig. 4 with Fig. 1 will show the great gain in stiffness in the latter design with only a slight increase in weight, the increase in stiffness being in the ratio of approximately 4.5 to 1. The type of design shown in Fig. 1, which is now more universally used, has eliminated crank-bolt breakage previously caused by bending, due to deflection of the crank end of the connecting rod. It can also be shown mathematically that the centrifugal force of the rotating end of the connecting rod applied outward, when crank and connecting-rod center lines are approximately 90 deg with each other, that the deflection of the design, shown in Fig. 4, will cause a bending stress in the bolts of approximately 34,000 psi. Adding to this the direct tensile stresses on the bolts, we find that the actual working stress is practically at the endurance limit of the material. This accounts for the failures.

Crank-Bearing Bolts

In four-cycle engines, the crank-bearing bolts must withstand

in tension the force due to reciprocating inertia plus the centrifugal force of the rotating end of the connecting rod (less that caused by the connecting-rod cap). In other types of engines, particularly double-acting engines, the actual maximum load must be determined by a complete analysis of loads for the entire cycle. The proportion of rotating and reciprocating portions of the rod can be determined by calculating its center of gravity or by weighing simultaneously the assembly on two scales.

Crank-bearing bolts are very vital parts of an engine and failure may cause considerable damage to the balance of the engine. For this reason, stresses must be conservative and the material should be the best steel that is obtainable. A factor of safety of approximately 12 to 13, based on the ultimate strength of the steel, is recommended in selecting the bolt size for a known working load. Heat-treated chrome-molybdenum steels of excellent physical properties, both in ultimate tensile strength and impact resistance, can be obtained. On assembly, these bolts should be tightened to give an initial stress of approximately twice the actual working stress.

It has been suggested that an important part of an engine, such as a crank-bearing bolt, should be replaced with a new one after a certain period of service because the bolt "fatigues" and must be replaced or it will ultimately fail. This should not be necessary if the engine is so designed that these bolts are not stressed to the endurance limit of the steel being used. Physical research on steels has proved that, when repeated stresses, below the static ultimate strength of the steel, are applied, failure occurs after a certain number of cycles. However, as various stresses are applied, there is a stress value below which the repeated loads can be applied with an indefinite number of cycles without failure. Research laboratories have accepted 10,000,000 cycles as the point of measurement of the endurance limit. Therefore, if the stress is at such a value that the specimen will absorb 10,000,000 or more cycles without failure, it will go on indefinitely without failure at this stress. As long as the actual stresses used on the bolt are safely below this value, known as the "endurance limit," these stresses therefore could be repeated indefinitely without failure. The fallacy of the practice of changing bolts every few years is apparent when considering that it would be necessary to change them perhaps each week or each month, depending upon engine speed, in order not to exceed 10,000,000 cycles.

Connecting-Rod-Bearing Caps

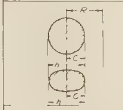
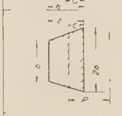
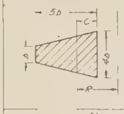
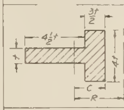
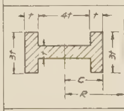
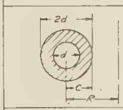
Regardless of the shape proposed for the cross section of the connecting-rod-bearing cap, the cap stress should be determined by the most severe consideration in order to gain rigidity. Again, it is not practical to state what definite deflections should be permitted, because the ideal would be zero and the limiting value is determined by good performance experience. The simplest method of gaining rigidity is to consider the cap as a freely supported curved beam. The supports are at the crank-bearing-bolt center lines and the maximum load applied to the center of the cap as a concentrated load. The actual total load in all crank-bearing bolts, obtained from a study of the forces (inertia forces and others), can be this concentrated load. Again, I-beam cross sections give the most strength and rigidity with the least weight.

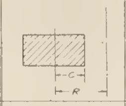
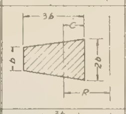
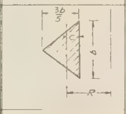
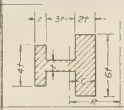
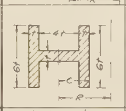
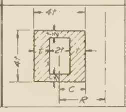
The bending moment on the beam is the simple beam formula

$$M = PX/2 \qquad [7]$$

where M = moment, in-lb; P = load on cap, lb; and X = distance from nearest bolt to cross section under consideration, in.

The stress caused by moment M in Equation [7] is determined by using the straight beam-stress formulas and applying a correction factor K. These K values are given in Fig. 5 for various

Section	$\frac{R}{C}$	Values of K: Inside Fiber	Values of K: Outside Fiber	$\frac{Y_0}{R}$*
	1.2	3.41	0.54	0.224
	1.4	2.40	0.60	0.151
	1.6	1.96	0.65	0.108
	1.8	1.75	0.68	0.084
	2.0	1.62	0.71	0.069
	3.0	1.33	0.79	0.030
	4.0	1.23	0.84	0.016
	6.0	1.14	0.89	0.0070
	8.0	1.10	0.91	0.0039
	10.0	1.08	0.93	0.0025
	1.2	3.01	0.54	0.336
	1.4	2.18	0.60	0.229
	1.6	1.87	0.65	0.168
	1.8	1.69	0.68	0.128
	2.0	1.58	0.71	0.102
	3.0	1.33	0.80	0.046
	4.0	1.23	0.84	0.024
	6.0	1.13	0.88	0.011
	8.0	1.10	0.91	0.0060
	10.0	1.08	0.93	0.0039
	1.2	3.14	0.52	0.352
	1.4	2.29	0.54	0.243
	1.6	1.93	0.62	0.179
	1.8	1.74	0.65	0.138
	2.0	1.61	0.68	0.110
	3.0	1.34	0.76	0.050
	4.0	1.24	0.82	0.028
	6.0	1.15	0.87	0.012
	8.0	1.12	0.91	0.0060
	10.0	1.12	0.93	0.0039
	1.2	3.63	0.58	0.418
	1.4	2.54	0.63	0.299
	1.6	2.14	0.67	0.229
	1.8	1.89	0.70	0.183
	2.0	1.73	0.72	0.149
	3.0	1.41	0.79	0.069
	4.0	1.29	0.83	0.040
	6.0	1.18	0.88	0.018
	8.0	1.13	0.91	0.010
	10.0	1.10	0.92	0.0065
	1.2	2.52	0.67	0.408
	1.4	1.90	0.71	0.285
	1.6	1.63	0.75	0.208
	1.8	1.50	0.77	0.160
	2.0	1.41	0.79	0.127
	3.0	1.23	0.86	0.058
	4.0	1.16	0.89	0.030
	6.0	1.10	0.92	0.013
	8.0	1.07	0.94	0.0076
	10.0	1.05	0.95	0.0048
	1.2	3.28	0.58	0.269
	1.4	2.31	0.64	0.182
	1.6	1.89	0.68	0.134
	1.8	1.70	0.71	0.104
	2.0	1.57	0.73	0.083
	3.0	1.31	0.81	0.038
	4.0	1.21	0.85	0.020
	6.0	1.13	0.90	0.0087
	8.0	1.10	0.92	0.0049
	10.0	1.07	0.93	0.0031

Section	$\frac{R}{C}$	Values of K: Inside Fiber	Values of K: Outside Fiber	$\frac{Y_0}{R}$*
	1.2	2.89	0.57	0.305
	1.4	2.13	0.63	0.204
	1.6	1.79	0.67	0.149
	1.8	1.63	0.70	0.112
	2.0	1.52	0.73	0.090
	3.0	1.30	0.81	0.041
	4.0	1.20	0.85	0.021
	6.0	1.12	0.90	0.0093
	8.0	1.09	0.92	0.0052
	10.0	1.07	0.94	0.0033
	1.2	3.09	0.56	0.336
	1.4	2.25	0.62	0.229
	1.6	1.91	0.66	0.168
	1.8	1.73	0.70	0.128
	2.0	1.61	0.73	0.102
	3.0	1.37	0.81	0.046
	4.0	1.26	0.86	0.024
	6.0	1.17	0.91	0.011
	8.0	1.13	0.94	0.0060
	10.0	1.11	0.95	0.0039
	1.2	3.26	0.44	0.361
	1.4	2.39	0.50	0.251
	1.6	1.99	0.54	0.186
	1.8	1.78	0.57	0.144
	2.0	1.66	0.60	0.116
	3.0	1.37	0.70	0.052
	4.0	1.27	0.75	0.029
	6.0	1.16	0.82	0.013
	8.0	1.12	0.86	0.0060
	10.0	1.09	0.88	0.0039
	1.2	3.55	0.67	0.409
	1.4	2.48	0.72	0.292
	1.6	2.07	0.76	0.224
	1.8	1.83	0.78	0.178
	2.0	1.69	0.80	0.144
	3.0	1.38	0.86	0.067
	4.0	1.26	0.89	0.038
	6.0	1.15	0.92	0.018
	8.0	1.10	0.94	0.010
	10.0	1.08	0.95	0.0065
	1.2	2.37	0.73	0.453
	1.4	1.79	0.77	0.319
	1.6	1.56	0.79	0.236
	1.8	1.44	0.81	0.183
	2.0	1.36	0.83	0.147
	3.0	1.19	0.88	0.067
	4.0	1.13	0.91	0.036
	6.0	1.08	0.94	0.016
	8.0	1.06	0.95	0.0089
	10.0	1.05	0.96	0.0057
	1.2	2.63	0.68	0.399
	1.4	1.97	0.73	0.280
	1.6	1.66	0.76	0.205
	1.8	1.51	0.78	0.159
	2.0	1.43	0.80	0.127
	3.0	1.23	0.86	0.058
	4.0	1.15	0.89	0.031
	6.0	1.09	0.92	0.014
	8.0	1.07	0.94	0.0076
	10.0	1.06	0.95	0.0048

* Yo is distance from centroidal axis to neutral axis, where beam is subjected to pure bending

Fig. 5 Correction Factors for Curved Beams

types of cross sections.[5] Thus, the stress at the section being considered is

$$S_b = KMc/I \qquad [8]$$

where S_b = bending stress, psi; M = bending moment from Equation [7]; c = distance from neutral axis to extreme fiber on side (tension or compression) being considered, in.; I = cross-sectional moment of inertia, in.^{4}; K = correction factor for curvature of beam and is variable depending upon shape of cross section and also on R/c_1 where R = radius of curvature of path of neutral axis at point where the cross section is being studied and c_1 = distance from neutral axis to extreme fiber on concave side of curved beam.

Thus, for a connecting-rod cap with a load P on the concave side we have

$$S_b = K_1Mc_1/I \qquad [8a]$$

compression on the inner fiber in psi, and

$$S_b = K_2Mc_2/I \qquad [8b]$$

tension on the outer fiber, psi. In Equations [8], K_1 = correction factor for inner fiber; K_2 = correction factor for outer fiber; c_1 = distance to inner fiber, in.; and c_2 = distance to outer fiber, in. Both K_1 and K_2 are given in Fig. 5 for the determined value of R/c_1 for a definite section shape.

By referring to Fig. 1, a better tie-in of the foregoing method to solving an actual problem is apparent. Various cross sections of the cap should be analyzed for stress in the same manner. At section CC in Fig. 1, note that there is a small radius where a flat is milled to prevent the head of the crank-bearing bolt from turning. Even if this radius is made as generous as possible, the tension stress at this point, as determined previously, should be

[5] Fig. 5 is reproduced from Seely's results in "Handbook of Engineering Fundamentals," reference (3), pp. 5–34.

increased by 50 per cent to allow for stress concentration due to this fillet.

It is suggested that the allowable bending stress on the cap, when determined as described previously, should not exceed approximately 10 per cent of the ultimate strength of the material used.

Piston Rods for Crosshead-Type Engines

The actual load on the piston rod, both in compression and tension, should be determined by a careful analysis of all the forces occurring during the engine cycle.

Piston-rod stresses in compression should also be determined by using Rankine's formula for short columns. The column should be considered fixed at the crosshead end and rounded and free at the piston end. This consideration gives the following formula

$$S_c = (P/A)[1 + 0.000078(l/r)^2] \qquad [9]$$

where S_c = column stress, psi; P = total thrust load on rod, lb; A = cross-sectional area of rod, sq in.; l = length of piston rod from center line of crosshead to center line of piston boss, in.; r = radius of gyration of cross section of rod, in.; for solid circular rods $r = d/4$, while for hollow circular rods

$$r = \sqrt{(d^2 - d_1^2)}/4$$

where d = outside diameter, in., and d_1 = inside diameter, in.

A factor of safety of approximately 5.5 based on the ultimate strength of the material is suggested.

For tension loads on the piston rod, the direct tensile stress on the least cross-sectional area (which may or may not be less than A given in Equation [9]) should not exceed about 10 per cent of the ultimate strength of the material.

Some Particulars of Design and Operation of Twin-Furnace Boilers

By JOHN BLIZARD,[1] NEW YORK, N. Y., and A. C. FOSTER,[2] NEW YORK, N. Y.

This paper outlines the advantages to be gained by making a boiler with two furnaces instead of one. These advantages comprise a lower furnace exit-gas temperature in a boiler provided with two furnaces than in a boiler with a single furnace, for the same rate of heat liberation per unit of furnace volume. Further, it is pointed out that, when a radiant superheater is installed in one of the furnaces, the final temperature of the steam may be controlled over a wide range of steaming by suitable adjustment of the rate of firing in both furnaces. A particular boiler generating 750,000 lb of steam per hr with two furnaces, fitted with both radiant and convection superheaters, is described and some particulars of its operation are given. Other boilers with two furnaces and provided with radiant superheaters are also illustrated and described.

It is well known that, as the size of a furnace increases, the ratio of the area of the walls of the furnace to the volume of the furnace decreases. In fact for precisely geometrically similar furnaces, cooled by steam or waterwalls, doubling the linear dimensions of the furnace halves the ratio of the cooling surface to the volume of the furnace. Furnaces, however, are designed to give approximately the same rate of liberation of heat per unit volume regardless of the size of the furnace and, consequently, the gases will leave the large furnace at a higher temperature than they will leave the small furnace under these conditions. With many fuels, this offers no difficulties but for others it means that the gas and the ash, which accompanies the gas, will enter the convection heating surface of the boiler and superheater at so high a temperature that ash will be deposited on the boiler and the superheater tubes. This restricts the flow of gas, changes the superheat, and, to some extent, reduces the efficiency of the unit.

In order to obviate these difficulties the fuel may be burned in two or more furnaces, whereby the ratio of cooling surface to total furnace volume is increased and the temperature of the gas leaving the furnaces may be reduced to a temperature low enough to prevent the deposition of molten ash on the convection heating surface. The furnaces themselves may be cooled either by waterwalls or a radiant superheater. The waterwall and superheater tubes should be bare, in order to cool the gases to the greatest extent and as early in their passage from the burner as possible, since this not only reduces the likelihood of slag depositing on the walls but also, by decreasing the mean temperature in the furnace, increases the time taken by a particle of coal to pass through the furnace, thus giving it more time in which to burn.

The use of two furnaces instead of one presents: (*a*) An opportunity to install a superheater in one or more of the walls of one of the furnaces where, by regulating the relative rates of combustion, final temperature of the steam may be held constant or varied as required over a wide-range load; (*b*) less time is required in starting than where only one furnace, in which there is a superheater, is employed. This is accomplished by firing a small amount or no fuel in the superheater furnace until steam generated by fuel fired in the water-cooled furnace passes through the superheater.

Typical Modern Twin-Furnace Boiler

In some designs it has been found desirable to place the entire amount of superheating surface in one of the furnaces, while in others it has been found more desirable to place only a part of the superheating surface in the furnace and to superheat the steam further by means of a conventional convection superheater. This general principle of design has been adopted for several boilers now in use, one of which is in operation at the plant of the Ohio Power Company, at Power, West Va. This boiler has been in operation for more than a year and has proved its ability to deliver steam at a constant temperature over a load range of 325,000 to 800,000 lb of steam per hr with no undue difficulties from deposition of slag on the heat-absorbing surface.

Cross sections of this boiler are shown in Fig. 1 and some particulars of the design are given in Table 1.

TABLE 1 DESIGN DATA OF OHIO POWER COMPANY TWIN-FURNACE BOILER

Design pressure, psi	1555
Pressure leaving superheater, psi	1350
Final temperature of steam, F	925
Design steam capacity, lb per hr	750000
Design heat release, Btu per cu ft per hr	23700
Volume of furnace, cu ft	40000
Height of furnace, ft	72
Heating surface	
Boiler and waterwalls, sq ft	15500
Economizer, sq ft	34800
Convection superheater, sq ft	18000
Air heater, sq ft	47800

Distribution of projected surface in furnaces below screen tubes

	Left-hand side, sq ft	Right-hand side, sq ft
Front and rear walls	1040	1040
Division walls	780	780
Left-hand wall	920	..
Right-hand wall	..	..
Radiant superheater	..	940
Screen tubes	220	210
Total	2960	2970

As indicated in Fig. 1, the left-hand furnace is cooled entirely by waterwalls, while the right-hand furnace is cooled by water on all walls except the right-hand wall in which is a radiant superheater. The division wall between the two furnaces consists of two rows of tangent tubes, which effectively screen one furnace from the radiation of the other and substantially prevent any material passage of gas between the two furnaces.

The mixture of water and steam leaving the division-wall tubes enters the small drum at the top of the division wall and is conveyed through four rows of staggered tubes, spaced 28 in. horizontally and 14 in. vertically, to the side walls of each furnace. These widely spaced tubes form a slag screen through which the gases pass to an intervening chamber before entering the nest of boiler tubes.

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Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

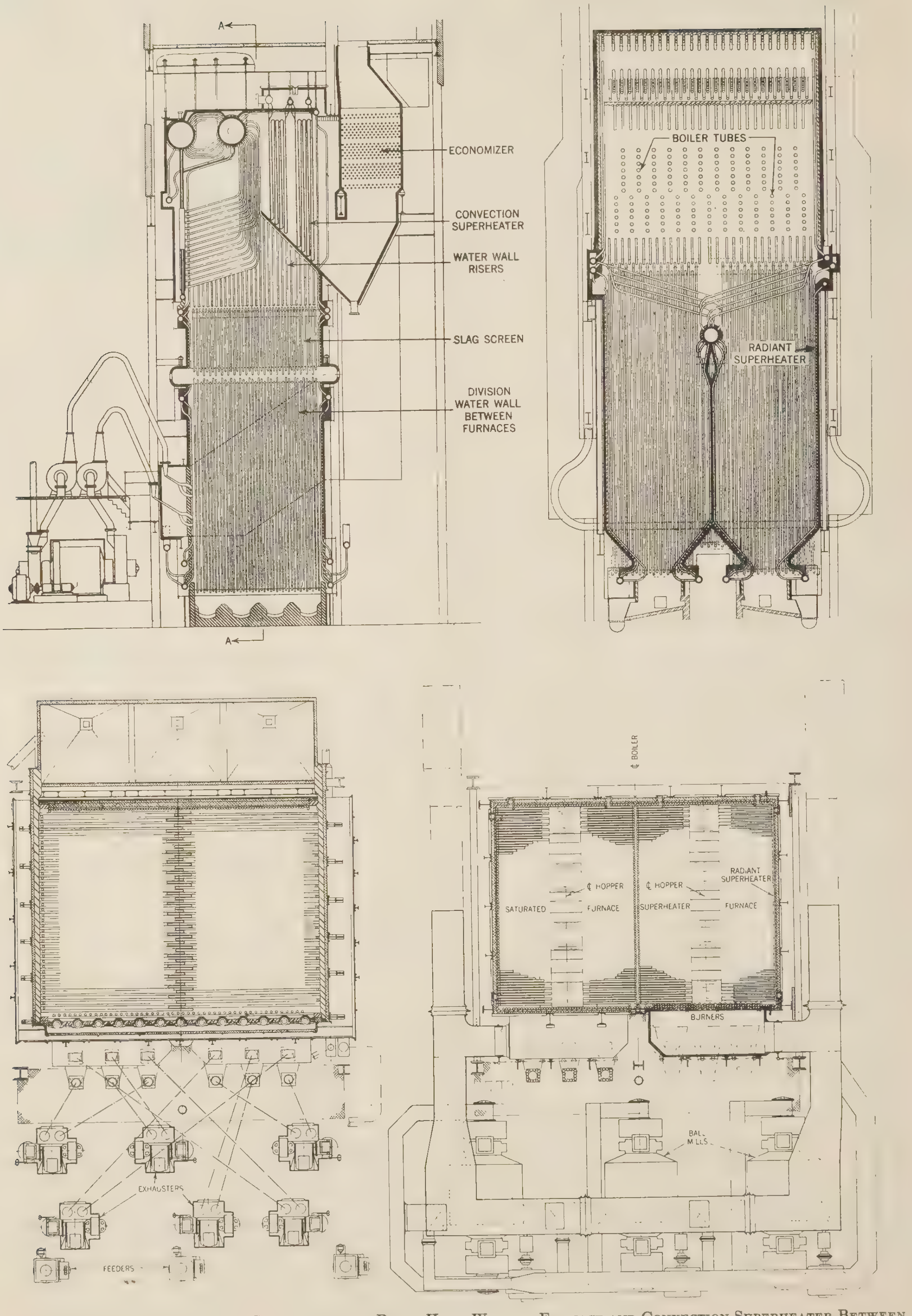

Fig. 1 Twin-Furnace Boiler With Superheater on Right-Hand Wall of Furnace and Convection Superheater Between Boiler and Economizer

The sloping portion of the boiler tubes also is widely spaced, i.e., 28 in. horizontally and 14 in. vertically. The upper bank is staggered with respect to the lower one but the tubes in each bank are in line. The boiler tubes in the vertical part of the tube bank are spaced 14 in. apart horizontally in the direction transverse to the flow of gas.

On leaving the boiler heating surface the gases pass over the convection superheater which consists of 2-in. tubes on $3^1/_2$-in. centers, then to an extended-surface economizer and two regenerative air heaters.

Two steam drums are provided. All the risers of the boiler and waterwalls enter one drum, called the turbulent drum. Some of the water and all of the steam then pass to a second drum in which a steam washer and drier are installed. From this drum, the steam passes first to the radiant superheater and thence to the convection superheater.

The boiler and waterwall downcomers are taken from both drums and are arranged so as to convey the water by a simple and direct path to the supply headers of the boiler and waterwall.

To maintain a uniform level of water throughout the length of the drums, the water leaves the drums through a series of connections spaced along the length of the drums, which eliminates longitudinal flow of water along the drums. Before leaving the turbulent drum, the water passes over a weir, which facilitates the removal of steam from the water before it enters the downcomers. The downcomers are not heated. This reduces the resistance to flow, gives a greater gravitational head at the entrance to the risers and a greater ratio of water to steam in the waterwall and boiler tubes.

The boiler is fired by coal which is pulverized in three ball mills, each having a capacity of $14^1/_2$ tons per hr. The mills are equipped with two exhausters apiece, each of which is arranged to fire two burners in each furnace. The twelve burners, of intertube type, are near the bottom of the front wall, in two rows of three burners in each furnace. With this arrangement, fuel is distributed uniformly across the full width of the furnace. The flame from the burners travels toward the rear and then upward through the slag screen to the boiler. The ash, which falls to the bottom of the furnace, is removed in dry form.

The temperature at which the gas reaches the convection superheater, as calculated from the temperature of the gas leaving the economizer where it was measured by a bare thermocouple and the heat balance on the superheater and economizer, is of the order of 1850 F at the designed load, so that no sticky ash is deposited on the convection superheater.

Fig. 2 shows how the temperature of the final steam varies with the rate of steaming. It will be seen that the final temperature remains practically constant at 925 F over a range of about 750,000 lb of steam per hr to 325,000 lb of steam per hr.

On the same chart is shown the temperature at which the steam leaves the radiant superheater and it will be observed that, as the rating decreases, this latter temperature increases which is brought about partly by this being an inherent characteristic of radiant superheaters and partly by firing at a somewhat greater rate in the superheater furnace than in the saturated furnace at the lower loads.

Had there been no radiant superheater and, if it were felt desirable to keep the temperature at which the gas enters the convection superheater down to about 1850 F at full rating, it would have been necessary to increase the surface of the convection superheater in order to obtain the final steam temperature of 925 F at the full output of the boiler.

Had this unit been of the usual design with a single furnace and convection superheater, but with gas entering the superheater at about 1850 F at full rating, it would have been neces-

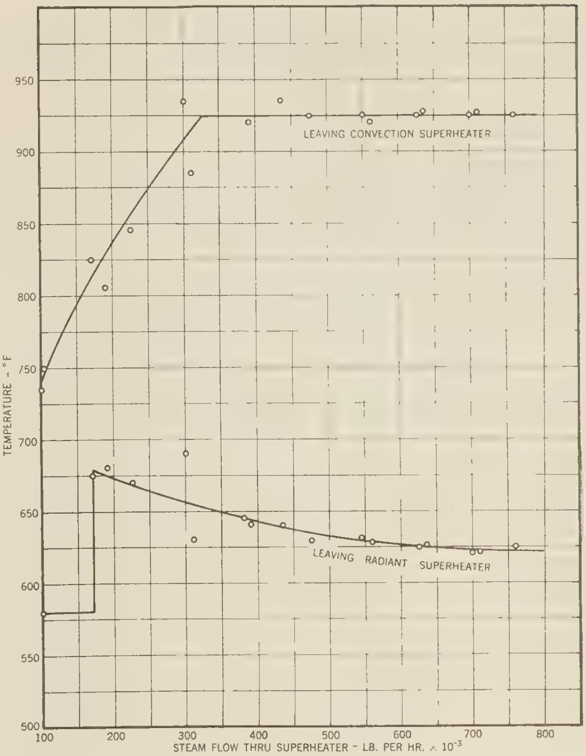

Fig. 2 Chart Showing How Final Steam Temperature Is Held Approximately Constant by Regulating Relative Rates of Firing in Furnaces of Twin-Furnace Boiler

(Radiant superheater shows saturated-steam temperature at low ratings because superheater furnace is not fired on starting up.)

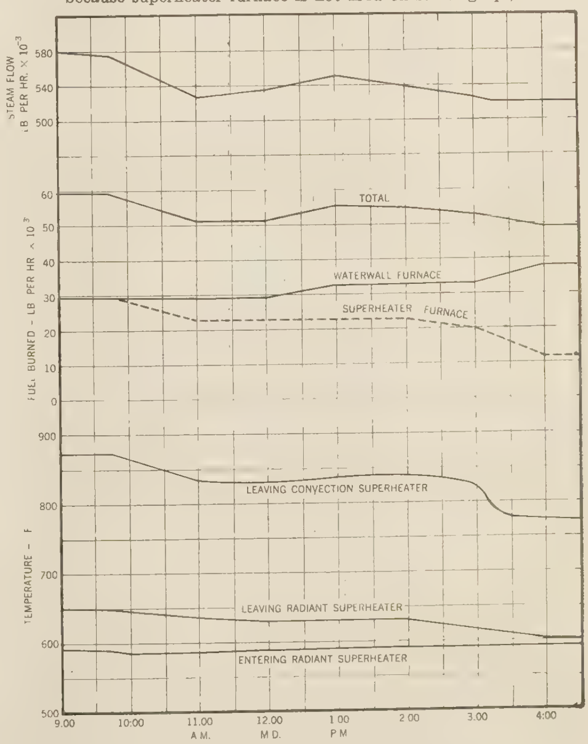

Fig. 3 Chart Showing How It Was Possible to Reduce Temperature of Steam in Twin-Furnace Boiler

sary to install a much larger convection superheater in order to maintain a constant steam temperature of 925 F over a range of 325,000 to 750,000 lb of steam per hr by by-passing the gases around the superheater at the higher rates of steaming.

Installation Has Great Flexibility

An interesting example of the flexibility of this boiler is shown in Fig. 3. The operators of the plant at one period asked to have a supply of about 500,000 lb of steam per hr at a temperature no higher than 800 F which is about 75 deg lower than the final temperature at which the boiler at that time was being operated. Fig. 3 shows the results of a test made to determine if this could be done. As the rate of firing in the superheater furnace relative to that in the waterwall furnace was decreased, the temperature of the steam leaving the radiant superheater decreased, resulting in a final steam temperature of about 775 F. This represents a drop of 100 deg below the normal temperature at which the unit was being operated at that time, and was 25 deg lower than had been requested by the operators.

Initially it took 24 hr to bring the unit up to design pressure, but now it is customary to take 12 hr to bring the boiler from zero to line pressure. It is believed that this can be accomplished in 8 hr without imposing any undue stress on the pressure parts. However, the longer period is preferred, since it is believed that care taken when starting minimizes troubles due to leakage, at rolled joints particularly.

The normal procedure of firing the boiler when starting is as follows: Two oil torches are ignited in the waterwall furnace and one in the superheater furnace. The latter torch is kept near the division wall and remote from the superheater. When a pressure of 230 psi is reached, the unit delivers steam to the low-pressure header and coal is then fired through the two center burners in the waterwall furnace. At this time about 100,000 lb of steam per hr is being made and the pressure is increased at a rate of 200 psi per hr until the pressure reaches 1200 psi.

Additional coal burners are then fired in the waterwall and superheater furnace and the pressure and temperature of the steam leaving the superheater adjusted to meet the requirements of the turbine. The boiler is then connected to the high-pressure header.

Great care was taken in putting this unit into service to avoid overheating any part of the unit and to avoid heating any part so rapidly as to set up undue stress. When putting the boiler on the line for the first time, the rate of firing was so adjusted that temperature rise of water in the boiler did not exceed 25 deg per hr.

Temperatures Taken of Structure and Parts

When starting, temperatures were measured by means of thermocouples at various parts of the unit. These temperatures included those of the boiler tubes entering the steam drum, waterwall tubes, temperatures on vertical boiler headers, temperatures of the beams supporting the economizer, temperatures on the

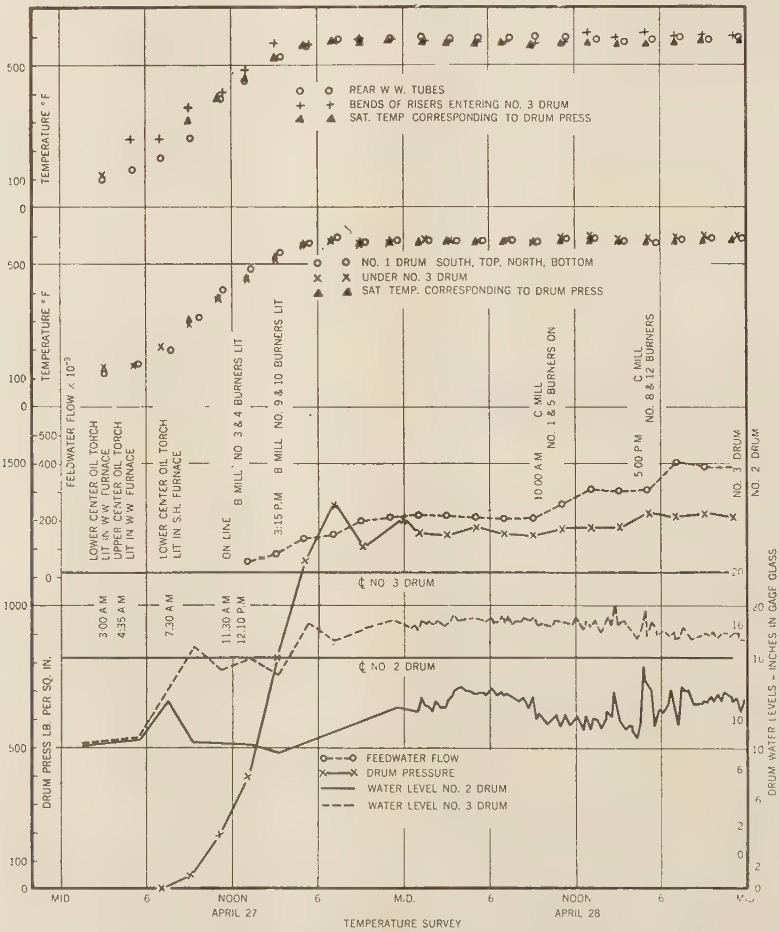

Fig. 4 Chart Showing How Levels of Water in Drums, Steam Pressure, Coal Rate, Feedwater, and Temperature of Tubes Varied on Starting

outside of the drum at the top of the division wall, temperatures on the No. 3 turbulent drum, which is the steam drum to which all riser tubes are connected, economizer-casing temperatures, temperature of girders and beams supporting the economizer, economizer tube sheet, temperature of radiant superheater tubes, convection superheater tubes, temperature of gas entering superheater, final steam temperature, etc. These temperatures were checked every hour and some of the more important temperatures oftener than this.

In Figs. 4 and 5 some of the readings taken, when starting in the usual manner, are shown graphically over the range of zero to 400,000 lb of steam per hr.

At the top of Fig. 4 are points showing the temperature on the bend of a riser entering the turbulent drum. A series of points, near these points, represents the temperature of the water in the boiler. The top of the bend of the riser is situated at a point where the ratio of the steam to water is highest in these risers. Its temperature was close to that of the saturation temperature throughout, although it is interesting to note that it exceeded the saturation temperature by more at the lower pressures and rates of firing than at the higher pressures and rates of firing. This temperature was measured at an elevation above that of the level of the water in the drum so that when starting there was no water in the tube at this point but, as the rate of firing increased, the circulation improved and the temperature of the tube approached that of the water.

Temperatures of the outer walls of No. 1 drum, which is the drum above the wall dividing the furnaces, and of No. 3 drum, which is the turbulent drum, are shown to be the same as the saturation temperature.

The levels of the water in the gage glasses on the turbulent (No. 3) and quiet drums (No. 2) are shown near the bottom of Fig. 4, and it will be seen that, when the load rises, the level in the turbulent drum rises above that in the quiet drum.

A further set of readings of the difference between the levels in the steam drums was taken during normal operation of the boiler. These readings are shown graphically by curve 3 in Fig. 6. Curve 2 shows the pressure required to force the steam from the turbulent to the quiet drum. Curve 1 is constructed by adding the ordinates of curve 2 to those of curve 3 and represents the pressure required to force water from the turbulent drum to the quiet drum. It will be seen that the circulation of water increased as the load increased.

Fig. 5 shows how the temperatures of a tube sheet supporting the economizer elements of the casing plate, surrounding one of the girders which supports the economizer, and of the gas to the superheater, increased as the boiler load was increased on starting.

At the bottom of the furnaces of this boiler are inclined walls beneath which are openings leading to an ash sluice. The distance between the burners and the bottom of the furnace is great enough to prevent any material circulation of the products of combustion immediately above the ash hoppers, which means that the gas in this space is comparatively cool, so that the ash itself in falling to the hoppers is cooled sufficiently to enable it to be removed without difficulty. It has never been necessary to use a furnace-cleaning tool or lance on the furnace hoppers.

The only soot blowers in the furnaces are those used to clean the radiant superheater. These are stationary nozzles placed in the front and rear wall of the furnace, by means of which the radiant superheater may be cleaned thoroughly.

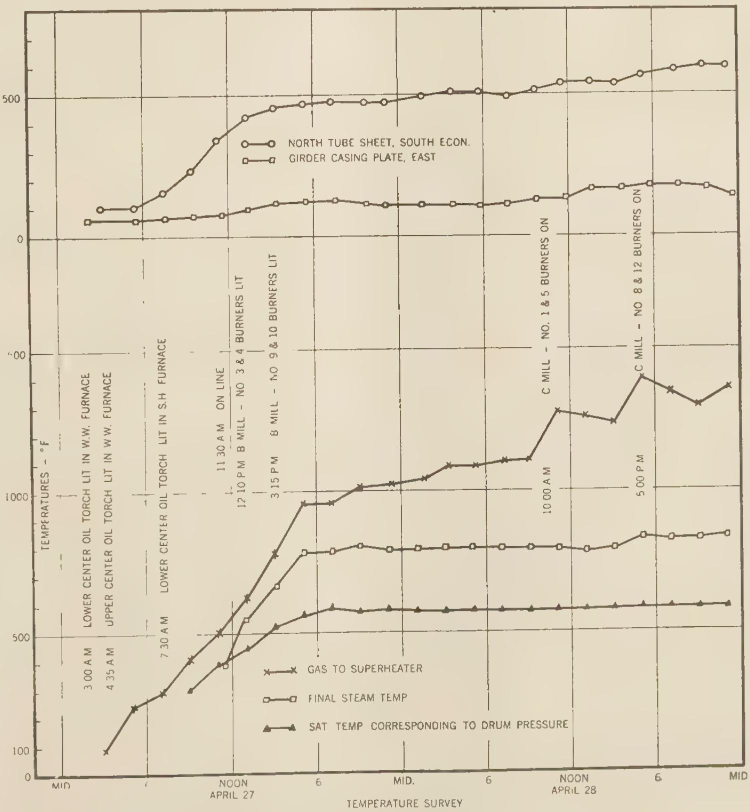

Fig. 5 Chart Showing How Temperature of Steam, Gas to Superheater, Economizer-Tube Sheet, and of Girder Casing Plate Beneath Economizer Varied on Starting

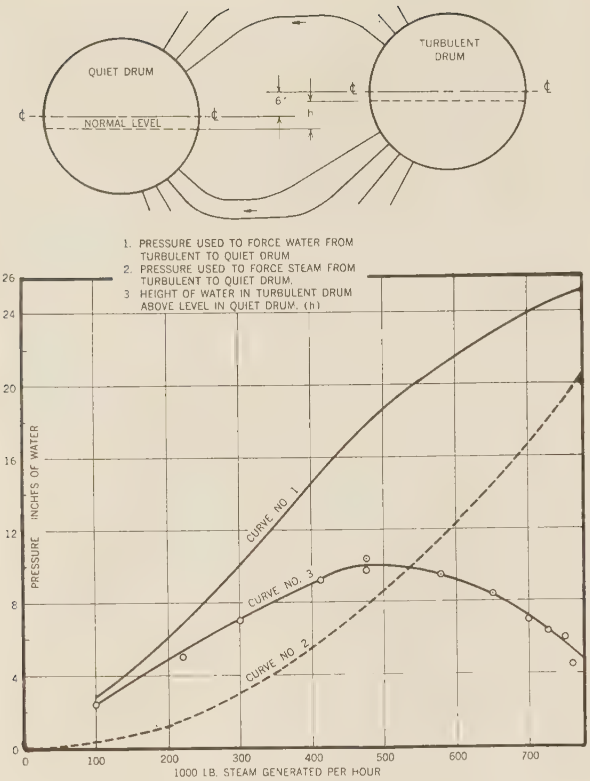

FIG. 6 CHART SHOWING DIFFERENCE BETWEEN LEVELS OF WATER IN TURBULENT AND QUIET DRUMS WHEN ON LOAD

The extent and frequency of cleaning the radiant superheater is under the control of the fireman who has the soot-blower nozzles operated, when the rate of firing in the superheater furnace to produce the final steam temperature of 925 F becomes unduly high for the particular load at which the boiler is operated.

The unit as a whole is kept clean by the use of soot blowers and a small amount of hand lancing. It is interesting to note that there has been no loss of capacity or outage of this unit due to slagging. One man per shift does all of the soot blowing and hand lancing and empties the soot hoppers. The slag-screen tubes above each furnace are lanced by hand as often as may be required which is between once or twice a shift, depending upon the kind of coal being burned.

The accumulations on these tubes are dislodged very easily when hand-lanced. The rate of accumulation is greatest at loads between 500,000 and 600,000 lb per hr. Below 500,000 lb per hr and above 600,000 lb per hr, the rate of accumulation is small and the time and labor of hand lancing correspondingly reduced.

TABLE 2 TYPICAL GAS AND WATER TEMPERATURES

Date of test (1940)...	July 4	June 22	June 16	June 15	July 3	July 2
Hours at rating......	6	6	4	5	5	4
Rating, 1000 lb per hr	440	500	640	700	700	735
Gas temperatures:						
To economizer.....	1008	1014	1120	1133	1130	1153
To preheaters......	518	518	561	575	569	581
Temperature drop..	490	496	559	558	561	572
From preheaters...	329	325	358	368	357	365
Temperature drop..	189	193	203	207	212	216
Air temperatures:						
From preheaters...	400	389	411	415	407	410
To preheaters......	113	96	100	100	99	103
Temperature rise...	287	293	311	315	308	307
Water temperatures:						
Economizer outlet.	562	554	566	570	567	569
Economizer inlet...	404	400	400	400	401	400
Temperature rise...	158	154	166	170	166	169

The amount of hand lancing on the convection heating surface of the boiler and superheater is small. In this region dry ash builds up on the downstream side of the heating surface until most of it falls from the tubes without the aid of steam jets.

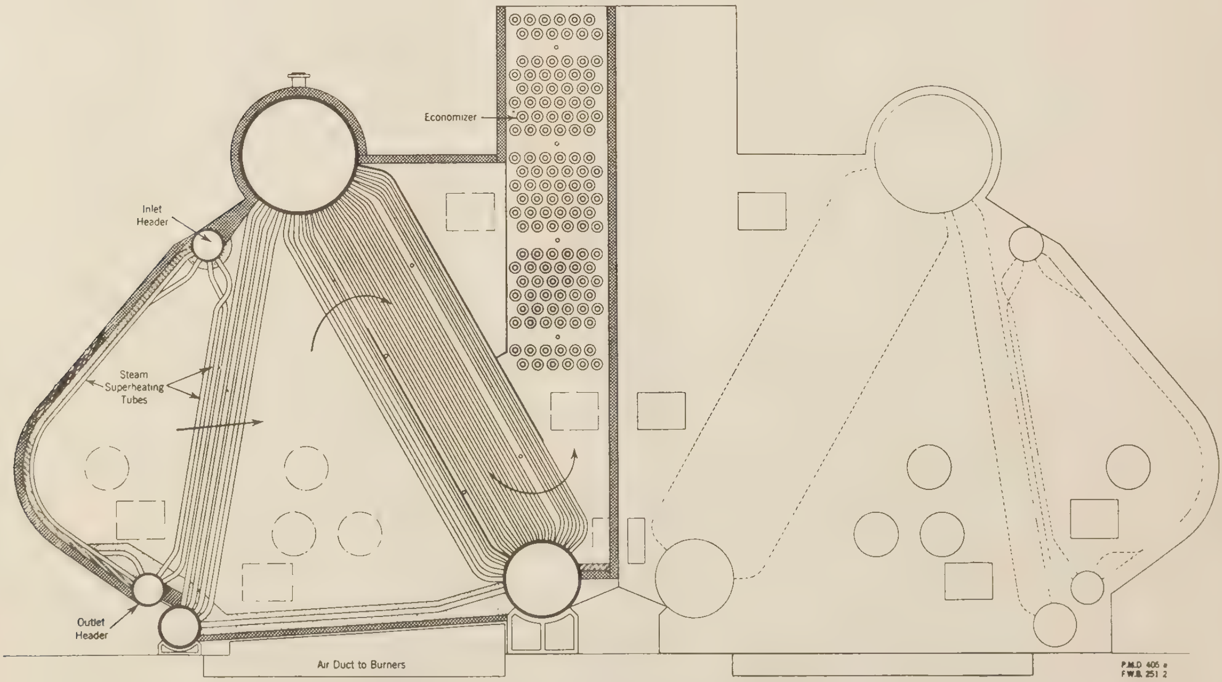

FIG. 7 MARINE BOILER WITH SEPARATELY FIRED RADIENT SUPERHEATER
(Gases from superheater furnace pass through main furnace.)

This boiler operates with an efficiency of about 86 to 88 per cent. The CO_2 in the gas going to the air heater is normally over 15 per cent. In Table 2, some gas and water temperatures, taken during normal operation, are tabulated.

Measurements of the impurities in the steam leaving the boiler indicate that the total solids varies from 0.5 to 0.6 ppm with total solids in the boiler water varying from 200 to 1400 ppm.

Separately Fired Superheater Marine Boiler

Another type of separately fired superheater boiler is shown in Fig. 7. Two of these boilers have been in use at sea since 1938, and were designed normally to generate 17,500 lb of steam per hr at 400 psi, 725 F, and at an overload to give twice this amount of steam at the same pressure and temperature. They are installed on a tanker where at times approximately 30,000 lb of saturated steam is required for pumping and 1500 lb of superheated steam for other purposes. It will be seen that there are two furnaces. In the furnace with the single oil burner is a superheater over which the gases from this burner pass and proceed through a screen of water tubes to the main furnace where there are three burners, and thence to the main-boiler heating surface. The temperature of the superheated steam is thus under complete control since no gases from the three main burners pass over the superheater, and comparatively little radiation finds its way to the superheater from the main furnace. The bulk of the heat absorbed by the superheater is by radiation from the oil flame which accounts for the comparatively small size of the superheater.

In order to avoid overheating the superheater tubes, an automatic device is provided which prevents the burner in the superheater furnace from receiving oil, unless steam is passing through the superheater. If at any time the temperature leaving the superheater is too high, the supply of oil to the superheater burner is automatically cut off.

No desuperheater is required with this boiler for supplying saturated steam since, by varying the rate of firing in the superheater furnace, the superheated steam can be held at the desired temperature, regardless of the proportion of the total steam which is being superheated. This is contrary to what occurs with a conventional boiler and superheater since, with this latter arrangement, if any steam is withdrawn from the drum the temperature of the superheated steam rises and, if the quantity withdrawn increases unduly, the steam in the superheater tubes may rise to too high a temperature.

In a recent paper,[3] a steam generator with two furnaces is shown which produces steam over a range of 30,000 to 120,000 lb per hr at a final temperature of 850 F and 600 psi when fired by oil or pulverized bituminous coal. It is required also to generate from 30,000 to 90,000 lb of steam per hr at the same pressure and temperature with pulverized lignite. Had the usual convection superheater been used on this boiler, it would have been impossible to obtain the same steam temperature with these widely divergent fuels without arranging for by-passing the gases around the superheater in quantities varying with the fuel used. However, this boiler has two furnaces, in one of which is a superheater so that by varying the relative rates of firing in the two furnaces, the required steam temperature is obtained without difficulty. The boiler is similar in principle to the marine boiler previously described but differs from it slightly in that the products of combustion from the superheater furnace, instead of passing through the screen of tubes separating the two furnaces for the full length of the furnace, pass through an aperture at the rear of the superheater furnace to the saturated furnace. Although, in this type of boiler the gases from the superheater furnace pass through the second furnace and thence to the boiler, no trouble has been experienced in burning the fuel in both furnaces with a slight amount of excess air.

A particular advantage with this type of boiler lies in the ease with which it may be started. When starting, fires are lighted only in that furnace in which there are waterwalls and the gases do not pass over the superheater. Thus, there is no danger of overheating the superheater when, in starting, no steam flows through it.

Fig. 7 of a previous paper[4] illustrates another type of boiler with two furnaces which is designed to generate 100,000 lb of steam per hr at 725 psi and 750 F final temperature when fired with pulverized coal. It has been in operation at Oil City, Pa., since 1935. This boiler is equipped with a convection superheater over which the gases from both furnaces pass. When operating the boiler below one-half load, only one furnace is used which causes the gases leaving the single furnace to be higher than they would be were both furnaces in operation; this in turn raises the rate of heat absorption of the convection superheater and so raises the temperature of the steam at the lower rates. In fact, with this mode of operation, there is but slight variation in the temperature of the steam over a range of evaporation of 30,000 to 100,000 lb of steam per hr. The variation of the temperature with the rating is shown in Fig. 8 of the paper[4] referred to.

Summary

Three types of boilers each with two furnaces and separate means of firing these furnaces have been described as follows:

1 A boiler with two furnaces, in one only of which is placed a radiant superheater. In addition to the radiant superheater, there is a convection superheater over which the gases from both furnaces pass.

2 A boiler with two furnaces, in one of which is placed a superheater. It differs from the first type in that the gases from the superheater furnace pass through the furnace in which there is no superheater. No convection superheater is provided. When, as on a ship, a separate supply of saturated steam is required, the steam may be taken directly from the steam drum.

3 A boiler with two furnaces from which the gases pass over a convection superheater. No radiant superheater is provided.

With these boilers, which have two furnaces instead of one, it is possible to increase the cooling surface surrounding the furnaces and, by differential firing of the two furnaces, to control the steam temperature over a wide range of steaming without installing a large convection superheater which has to be bypassed at the high loads if the temperature of the steam is to be controlled.

Discussion

R. S. Julsrud.[5] Several interesting features of the design of twin-boiler furnaces together with operating data of a large unit have been given in this paper. The writer would like to ask some questions concerning automatic control of steam output and steam temperature with this unit. In the conventional single-furnace and convection-type superheater, automatic control of steam output and steam temperature may be accomplished by any one of several automatic-control systems now available. However, with the twin-furnace boiler, a demand, say for in-

[3] "Superheat Control and Steam Purity in High-Pressure Boilers," by Martin Frisch, Trans. A.S.M.E., vol. 62, October, 1940, Fig. 4, p. 607

[4] "What Are Logical Trends in Design for Steam Generation?" by H. J. Kerr, John Van Brunt, and Martin Frisch, paper presented before Power Division of Metropolitan Section, A.S.M.E., New York, N. Y., Nov. 18, 1937

[5] Harmon-on-Hudson, N. Y. Mem. A.S.M.E.

creased steam output, would call for increased heat input to the saturated furnace. In order to maintain a constant total steam temperature, this might necessitate reducing the heat input to the superheater furnace. An explanation as to just how this is accomplished without a somewhat complicated system of automatic control would be of interest. The writer would further ask what means are employed to advise the operator when the rate of firing in the superheater furnace to produce a certain steam temperature has become unduly high for that particular load and prompts him to use the radiant-superheater soot blowers to remove accumulated ash and possible clinker from these surfaces.

The question of completely bare waterwall tubes or armor-clad surfaces in furnaces subjected to high heat inputs with consequent high furnace temperatures is still debatable. An increase in mean furnace temperature from 2000 F to 2500 F would increase the volume of the products of combustion in the ratio of 1 to 1.17 or 17 per cent and the velocity head approximately 0.037 in. water gage. Since complete combustion of a pulverized-coal particle depends largely upon rapid and intimate contact with oxygen, viz., turbulence, the particle size, volatile content, etc., and also temperature to a lesser extent, the increase in the time element in the furnace, which is due to even a large increase in furnace temperature would not materially affect the completeness of combustion of the particle.

Authors' Closure

Mr. Julsrud asks how the temperature of the steam, from the large boiler described in the paper is controlled.

The temperature of the steam is controlled automatically by a temperature controller that operates so as to vary the loading pressure on the fuel and air controllers for the saturated and superheater furnaces in accordance with the requirements. An increase in steam output is automatically met by an increase in heat input to both furnaces. Then if the steam temperature is low the loading pressure on the fuel controller regulating the fuel to the superheater furnace is increased which increases the heat input to the superheater furnace. This will increase the steam pressure since the total heat input is now more than the demand so the master pressure controller acts to reduce the input to both furnaces. This action is maintained until the steam temperature is as required. The opposite takes place if the steam temperature is high following a change in load.

Differential pressure gages are provided to indicate to the operators the extent by which the flow of fuel to one furnace exceeds that to the other furnace. At full load both furnaces are fired equally. From experience the operator knows about how much difference in differential there should be for the designed steam temperature at various loads. When the operating gages indicate too great a difference from the normal in fuel flow to the superheater furnace he has an indication that the radiant superheater needs cleaning. Another indication is obtained from the position indicators of the fuel controllers. In addition the desirability of cleaning the radiant superheater may be made by inspection.

Mr. Julsrud also discusses the effect of increasing the mean temperature of the gases in the furnace from 2000 to 2500 F on burning the coal. This increase in temperature would increase the specific volume of the gases by 20 per cent and accordingly would cause the gases at the lower temperature to take 20 per cent longer to pass through the furnace, which would thus give the coal 20 per cent longer time in which to burn. He appears to infer that since the velocity pressure will be higher at the higher temperature, the turbulence will be greater. But velocity pressure unless taken as a ratio of the pressure gradient is not a measure of turbulence. Actually, since the viscosity at 2500 F is 11 per cent greater than it is at 2000 F the turbulence presumably is less at the higher temperature.

Condenser Tubes and Their Corrosion

By CHARLES W. E. CLARKE,[1] A. E. WHITE,[2] and C. UPTHEGROVE[3]

The authors present a brief review of the development of condensers for steam prime movers, a discussion of some of the problems in connection with them that have been met and overcome, and a tabulation of returns from a questionnaire. They then describe and report a series of tests with a miniature condenser with tubes made of admiralty metal, aluminum brass, and cupronickel. Conditions were as closely as possible a duplication of those existing in a working condenser. In addition to corrosion tests, impingement tests were made. The authors present and discuss the results of these tests and grade the tubes tested in order of resistance to corrosion. Full information on the tubes tested is presented, with the results of metallographic examination of the materials of which the tubes were made. The conclusions arrived at as a result of a study of the tests are summarized at the end of the paper.

This paper has to do mainly with condenser tubes and their life as related to the material of which they are made. However, a brief review of the development of condensers, particularly as applied to stationary power-plant practices, and a discussion of some of the problems that have been met and overcome may be of interest.

The use of condensers to extend the heat range of steam-power units is old. In fact, Savery and others in the seventeenth century used the condensation of steam in a closed vessel to perform work. Surface condensers began to come into use along with steam navigation but there was little or no development until the advent of the steam turbine as an important prime mover. Prior to that development, high- or low-head jet condensers were the principal means of providing vacuum for engine-driven power units in land practice.

The original surface condenser consisted of a shell containing as many tubes as could be crowded in. Vacuum requirements were not severe, as these early installations were made in conjunction with reciprocating engines. One of the principal land uses originally was in connection with pumping engines which always provided a cooling-water flow many times the actual needs for steam condensation, so that what might be called the "surface efficiency" was not important.

The original shell full of tubes has given way to tube layouts which provide proper steam access to all of the cooling surface. This has permitted a great reduction in the surface requirement per pound of steam condensed. Pressure drops across the condenser have been reduced from about one half inch of mercury in the older designs to a tenth of an inch in present-day installations. This is no small item, especially for turbine operation where exhaust pressures of one inch of mercury absolute or less are the order of the day instead of the $2^1/_2$ to 3 in. of days gone by.

An interesting example of the early steps in the improvement of condenser performance occurred in 1908 at the Port Morris Station of the New York Central Railroad. The condensers consisted of a shell packed as full as possible with 4000 tubes. About 25 per cent of the tubes were removed in such a way as to leave three tapering steam lanes leading into the body of the tube bank and a fairly wide lane along the top. This change resulted in an increase of about one inch in vacuum.

Vacuum pumps were of the wet-vacuum type, usually driven from the shaft of the main prime mover. Later, separately driven dry-vacuum pumps were developed, to be superseded in present-day practice by steam-jet air pumps.

In order to secure some authentic information as to the detailed history of surface-condenser development, questionnaires were sent to several of the largest manufacturers of condensers and tubes. The answers to these questions are shown in Tables 1 and 2.

In the development of surface condensers, one of the main difficulties has been corrosion of tubes. The location of many of our large public utilities and industrial power plants is such that the condensing-water supply is contaminated by industrial waste and sewage, resulting in water which is corrosive. In many cases the water contains gases which are liberated in passing through the condenser, which add to the corrosive difficulties.

The liberation of corrosive gases within the condenser tubes owing to the increase in temperature and reduction of pressure is a principal cause of tube deterioration. Various methods have been tried to remove part of these gases before they can act on the cooling surfaces. One simple method consists of suitable pipe connections from the top of the inlet water boxes to the drop leg of the circulating system. The vacuum produced by the drop-leg siphon draws off a mixture of water and such gases as may be free at this point. However, this will not remove the dissolved gases which are occluded owing to the temperature rise and pressure drop within the tube itself.

Much has been written regarding these matters in the past. However, each location or water quality presents a different problem and results in actual service have more often than not been contradictory.

Collection of debris at the tube entrances (leaves, etc.) has always been a problem in surface-condenser operation. High-pressure water jets from nozzles which operate through ball-and-socket joints in the water-box cover plates have been effective in many cases. Divided water boxes, which allow tube replacement and cleaning of half the condenser while the unit operates, are common.

Hot-well temperatures in old-type condensers were often five degrees lower than the temperatures due to vacuum. Proper tube laning in modern condensers allows steam to come in contact with the condensate going to the hot well. This secures temperatures substantially the same as those due to vacuum and at the same time causes more complete deaeration of the condensate which is so important in modern high-pressure-boiler practice.

Tube end leakage was formerly a serious problem. Methods of packing have been gradually improved from the old corset lace to fiber, metallic, and finally to the expanding of tubes into the tube plates without the use of any packing material. In most of the older designs the tube ends protruded beyond the face of the tube sheet. This resulted in serious wear and deterioration of the tube at the inlet end. To correct this condition,

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Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 2–6, 1940, of The American Society of Mechanical Engineers.

TABLE 1 HISTORY OF CONDENSER DESIGN

	Westinghouse Elec. & Mfg. Co.	Foster Wheeler Corp.	Worthington Pump & Machinery Corp.	Ingersoll-Rand Co.
What was the date of your first installation of steam-jet air pumps?	1919	1917 (Alberger)	1918, Erie Works, General Electric Company	1920
(a) When did you first appreciate the necessity for proper steam laning?	1912	1916	1895, Nassau Electric Railway Company, 39th Street Station, 4500 sq ft	Prior to 1922
(b) What were the main phases you went through in connection with this development?	Only development has been radial steam inlet	1 Omitting tubes to form lanes 2 Radial design	1 Steam-inlet lanes 2 Steam-inlet and outlet lanes	Use full available area, spacing tubes to suit quantity of steam flowing Where necessary use "steam by-passes" which are shrouded ducts
What were some of the changing guarantees you made in drop across condenser, inches Hg?	Drop changed from 0.4 in Hg to 0.1 in Hg in large units	No answer	Drop changed from 0.5 in Hg in 1908 to 0.1 in Hg for present-day design	None
When did you first attempt to take gases out of circulating water to prevent cavitation at tube entrances?	Holes in horizontal partition 1916	No answer	1916, 35,000 sq ft condenser of Lowellville Station, Republic Railway & Light Company 1917, Connors Creek, Detroit Edison Company	1925
When did you first build a condenser with divided water box?	1913	1921	1913, Brunot Island, Duquesne Light Company, 2 condensers for one turbine 1920, Hell Gate Station, Consolidated Edison Company	1922
When did you first build a condenser with deaerating hot well?	1921	1923	1929, three 57,000 sq ft for Ashtabula Station, Cleveland Electric Illuminating Co.	1923
When did you first supply a condenser with expanded tubes?				
(a) One end	1922	1910 (Alberger)	1895, marine condensers, 1916 (35,000 sq ft)	1920
(b) Both ends	1927	1925	1890 (300 sq ft)	1929
When did you first equip a condenser with venturi or rounded-type inlets at the tubes?	1922	No answer	1888 (two 2120 sq ft, City of Cincinnati)	1920
When did you make your first all-steel welded-shell condenser of fairly large size?	1931	1935	1930, one 10,300 sq ft for Imperial Oil Company, five 12,492 sq ft for Baltimore Mail Line steamers	1929. 7000 sq ft condenser 1931, 26,000 sq ft condenser
What decrease in surface requirements per pound of steam, or increase in heat transfer?	Now they go as high as 850 Btu/sq ft/F/hr	Up to 1918. Used 350 Btu/sq ft/F/hr. Now they go as high as 850 Btu/sq ft/F/hr	Heat-transfer rate increased about 50 per cent since 1908	Have always used high transfer rate
Expansion joints between condenser and turbine?	Still used on some installations Copper Slip joints Rubber types Most installations use spring supports Counterweight Condenser bolted or welded to turbine exhaust	Still used extensively on small units Copper Rubber Most installations use spring supports Phillips slip-type joint, using rubber packing ring, water-sealed	First used in 1902, at Elyria, Ohio First spring supports, 1914, on six condensers for 74th Street Station of the Interborough Rapid Transit Company Spring support with hydraulic jacks, 1918, for three 50,000 sq ft, 59th Street Station of Interborough Rapid Transit Company First rubber joint, 1922, for Cahokia Station First Phillips rubber joint, 1923, Muskogee Station, Oklahoma Power Company	No answer
Internal air coolers?	......	......	First in 1904	No answer
Tangential drainage condensate tubes?	......	......	First in 1904, for 3000 sq ft condenser, Boston Navy Yard	No answer

the tube ends were kept within the sheet and an inlet ferrule was screwed into the tube sheet at the inlet end.

Corset-lace packing was always troublesome and difficult to install properly. If the tubes were soft, drawing up on the follower often crimped the tube, making a tight joint impossible. Raw tallow was quite generally used with corset lace and broke down into various acids, oleic in particular, which were injurious to the tube metal. Fiber packing was fairly satisfactory but subject to leaks. Alternate rings of fiber and metallic packing probably constituted the most efficient of its time. The straight metallic packing was quite satisfactory if expertly installed. However, this too was subject to crimping. Finally, the practice of expanding the tubes into both tube sheets came into quite general use. In some cases the tubes were bowed slightly so as more easily to take up expansion and contraction due to changes of temperature. Condensers have also been built with expansion joints in the shell, as well as with a floating head.

Various arrangements have been used from time to time to cool the air removed from the condenser before it reached the vacuum pump. Generally, the coolers are placed within the condenser and consist of baffled sections arranged so that the air passes over a section of the tube bank before it reaches the outlet pipe. External coolers have been used but require separate vacuum and cooling-water pipe connections and offer opportunity for leakage into the vacuum system. This leakage can, of course, be largely overcome by complete welding of all connections.

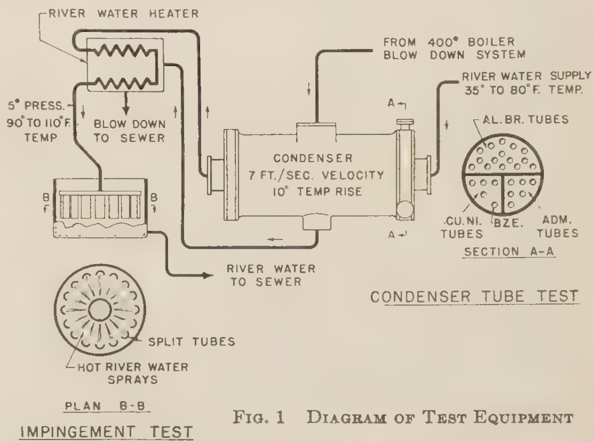


Fig. 1 Diagram of Test Equipment

TABLE 2 HISTORY OF CONDENSER TUBES

	Foster Wheeler Corp.	The American Brass Co.	Phelps Dodge Copper Products Corp.	Scovill Mfg. Co	Bridgeport Brass Co.
A What kind of tubes did you make at various times?	Muntz metal: 60% copper, 40% zinc 70–30: 70% copper, 30% zinc 2 & 1 mixture: $66^2/_3$% copper, $33^1/_3$% zinc Leaded admiralty: 68% copper, 0.75% lead, 1.00% tin, Rem. zinc Red brass: 85% copper, 15% zinc Copper: 99.90% copper Arsenical copper: 99.45% copper, 0.15–0.50% arsenic	Before 1890 Muntz metal 1889 Admiralty 1892 Mannesmann billet Piercing machine 1903 Benedict nickel 15% cupronickel 1909 Cup drawing process 1925 Ambrac 20% nickel alloy 1925 Ambrac B 30% nickel 1925 Red brass 85% copper 15% zinc 1930 Supernickel 70:30 cupronickel 1930 Aluminum brass 1931 Extrusion press	1900 Most tubes 2 & 1 mixture Some Muntz Some admiralty 1912 Muntz tubes had practically replaced 2 & 1 mixture. About 20% were tinned inside and out. Increased use of Admiralty 1925 About 75% of all tubes used were admiralty 1925 Start of aluminum brass 76% Cu, 22% Zn, 2% Al 1930 Copper-nickel tubes. Used mostly by Navy and Maritime Commission	Copper Phosphor copper Arsenical copper Muntz metal, 60% Cu, 40% Zn Scovill special Muntz 60–63% Cu, remainder zinc Brass, 70% Cu, 30% Zn Admiralty, 70% Cu, 29% Zn, 1% tin Phosphorized admiralty Red brass, 85% Cu, 15% Zn Red alloy, 85% Cu, 14% Zn, 1% tin Aluminum bronze, 95% Cu, 5% Al Alcunic Cu Al Ni Tin Zn 70 2 1 .. Bal. 80 2 1 .. Bal. 80 2 1 1 Bal. Aluminum brass, 76% Cu, 2% Al, 22% Zn, 0.01 to 0.05% As Adnic., 70% Cu, 29% Ni, 1% tin Cupronickel, 70% Cu, 30% Ni, 80% Cu, 20% Ni	Muntz metal Admiralty 70–30 Aluminum brass Aluminum bronze Cupronickel *80–20 *Arsenical copper
B When did you first begin to think grain-size control advisable?	Do not control grain size	1912, in conjunction with Mr. Sparrow of New York Edison Company	1910. There is no evidence that any relation exists between grain size and tube life	1905. Preceded our production of condenser tubes and has always been applied in the manufacture of condenser tubes since 1918	Never have thought that "grain-size control per se was advisable or useful" or had anything to do with tube life
C When did you first institute annealing and the proper controls for it?	Ever since condenser tubes were made and instruments for Phrometric control were available	1912	No definite date In universal use in 1920	For condenser tubes in 1918	About 1900 realized that final annealing was important
D What various kinds of packing have you used, corset lace, fiber, metallic, etc., and when?	Corset lace Fiber Lamp wick Metallic No definite dates	No answer	Cotton lace Fiber packing Metallic packing, about 1920 Alternate layers of fiber and lead-foil rings 1924, "Smoothflo" tube inlet ends	No definite answer	Type of packing not a responsibility of the tube manufacturer * Sometimes with small quantities of nickel and tin.

Condenser shells were originally made almost universally of cast iron with flanged joints which became rather complicated and hard to keep tight in large condensers. With the advent of welding the use of steel-plate shells came rapidly into vogue and today practically all condenser shells are of welded steel-plate construction. These are free of flanged joints and are much lighter and less costly than the old cast-iron shells.

The exhaust connection between turbine and condenser has always been a troublesome feature. Expansion joints of metal or rubber have been used but they involved two large flanged joints which were extremely difficult to make tight in the first place and to keep tight after installation. The joints themselves were more or less subject to cracking with age. In other cases the shell was bolted to the exhaust flange of the condenser and carefully designed systems of springs were installed to carry the weight of the condenser when there was no vacuum in the system. The most desirable arrangement is to hang the condenser from the turbine and weld the joint with suitable supports provided to take the weight when the condenser is filled with water for testing.

When the question of condensing equipment for the extension for The Narragansett Electric Company was under discussion, the condenser-tube problem was seen to be of major importance. The condensing water available is particularly bad and heretofore only jet-type condensers have been used, so no experience has been had with tubes.

In an attempt to assure the best material for tubes, it was decided to conduct an extensive test on various materials in a way that would as nearly as possible duplicate the conditions actually existing in a working condenser.

Based on such information as was available, Admiralty-metal, aluminum-brass, and cupronickel tubes were selected as the ones most likely to give good service. A miniature cast-iron condenser with Muntz metal tube plates was used, as shown diagrammatically in Fig. 1 and tubes of each of the three materials just mentioned, as well as a bronze tube, were installed as indicated in the cross section. Barriers were placed in the inlet water box as indicated, separating the sections containing the various kinds of tubes so there could be no possibility of water coming in contact with more than one kind of tube. No attempt was made to vent the water spaces. River water was passed through the tubes at a velocity corresponding to that expected in the actual condenser installation and blowdown water from the station 400-lb blowdown system was admitted to the steam space.

In addition to the condenser test, an impingement test was made. As will be seen from the diagram, Fig. 1, the river water from the test condenser was passed through a small heater where its temperature was raised to 90–110 F and then sprayed against the inner surface of split-tube samples. It was recognized that this test did not bear a direct relation to the conditions existing in a condenser, but it was thought that the information obtained might supplement that from the condenser test. It was also felt advisable to make an impingement test as it was a test which would produce deterioration in the tube metal more rapidly than would be the case from a condenser test. When the work was started, it was believed that the time element for the condenser test would be of the order of but a few months. Therefore, it was essential that as much supplementary information as possible be obtained. As a matter of fact, there proved

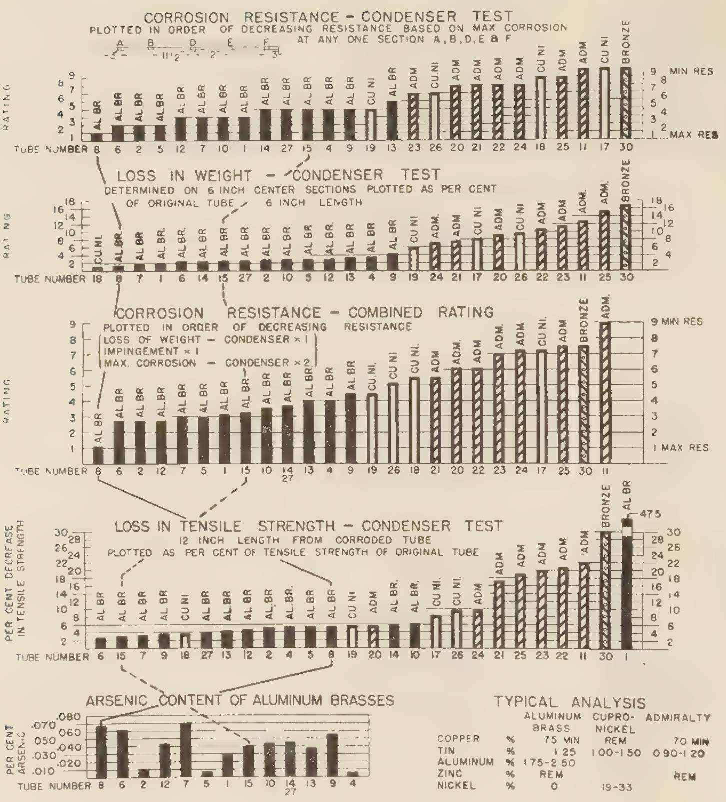


Fig. 2

to be reasonable agreement between the results obtained from the impingement test and the condenser test.

The work on The Narragansett Electric Company extension was delayed, making it possible to carry on this test fairly continuously for about 22 months. The only serious interruption was that due to the hurricane of September, 1938. The entire equipment had to be rebuilt and the test was interrupted from September 21 to October 26.

Fig. 2 shows graphically the comparative values of the various tubes as indicated by the tests. The first line shows the corrosion resistance in order of decreasing value as determined by the condenser test. The second line shows loss of weight in order of increasing values. The third line is a combination of the corrosion-resistance results as shown by both the condenser and impingement tests. The fourth line shows the loss in tensile strength after the condenser test. The lines connecting the various tests are drawn in an attempt to simplify interpretation of the figures. The solid line connects the various values for a tube which gave uniformly good results, and the dotted line connects the values for a tube which gave rather scattered results. Fig. 3 shows a comparison of the corrosion-resistance results from the condenser and impingement tests.

Fig. 4 is a cross section of a condenser showing steam laning and the arrangement for removing entrained gases from the inlet water boxes. This condenser has an external air-cooling section.

RESULTS OF INVESTIGATION

The details of the supplementary tests, the corrosion tests, including the impingement and condenser tests, and the metallographic examination are given in the following section of the paper.

Test Material

In carrying forward the program outlined in the preceding paragraphs, 27 different tubes were examined, although it would appear from Table 3 that there were 28. Specimens 14 and 27, however, were from the same tube. Tubes 28 and 29 are not included in the table as these tubes split in installation and no corrosion tests, therefore, were made upon them. The tubes which were tested were of the aluminum-brass, cupronickel, admiralty, and bronze compositions. They were furnished by various manufacturers and the question of grain size and physical properties resulting from the drawing and annealing operations was left largely in the hands of the companies producing the tubes.

The materials supplied are given in Table 3.

Chemical Composition

No specific analyses were made of the stock, although it was assumed that all the tubes furnished complied, from the chemical standpoint, with the A.S.T.M. standard specifications for the respective grades involved. The type of composition is given in Table 4.

Among condenser-tube manufacturers various claims are made for the respective advantages of slight additions of arsenic, phosphorus, and antimony. As the results of this investigation

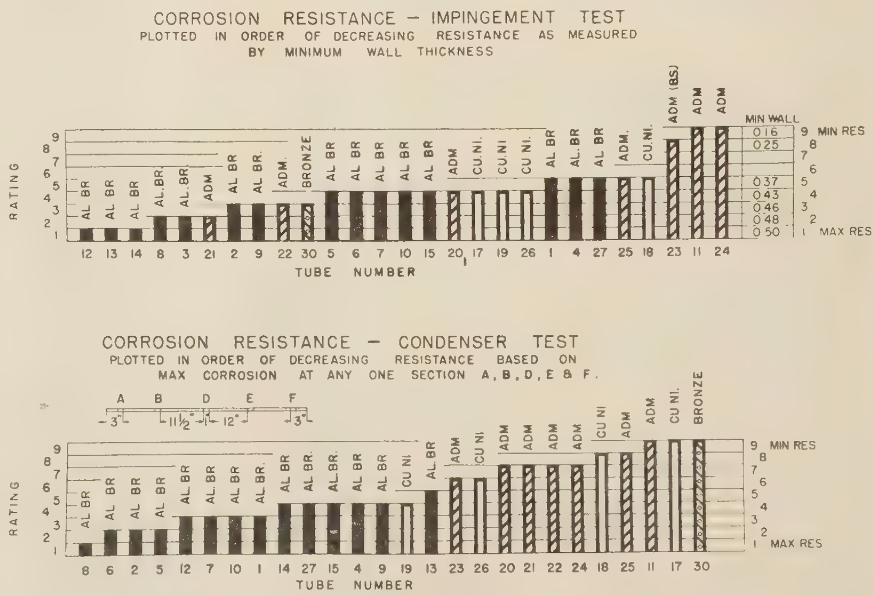


Fig. 3 Comparison of Impingement and Condenser Test

showed the aluminum-brass tubes to be superior to any of the other types of tubes, and as arsenic is said to be of importance in aluminum brass, an analysis was made of all of the different types of aluminum-brass tubes for arsenic. The results given in Table 5 and Fig. 2 indicate that arsenic was present in all of the tubes, ranging in amounts from 0.006 to 0.071 per cent. Though it would appear from current practice that arsenic additions are desirable, the findings do not point to any definite upper or lower limitations for the amount that should be present.

Claims are also made that phosphorus additions are desirable. A qualitative test was made, therefore, on six of the aluminum-brass tubes which made the best showing under the combined tests, and traces of phosphorus were found in all of these tubes.

DISCUSSION OF RESULTS

Before proceeding to the discussion of the results from the impingement and condenser corrosion tests, for these are the tests on which the major emphasis will be placed, a number of supplementary or additional tests were made. Among these were the mercurous-nitrate, flattening, expansion, and tensile tests.

No details with regard to the mercurous-nitrate, flattening, and expansion tests are included in this paper as the findings appear to be of no significance from the standpoint of resistance to corrosion. It is, of course, to be understood that any tubes which are to be installed in condensers should meet the requirements of the mercurous-nitrate and expansion tests. Flattening tests are not included as they are no longer a requirement in the standard A.S.T.M. specifications for condenser tubes.

Tensile Tests

Recognizing the probability that as a result of the corrosion attack the tensile properties of the tubes would be decreased, tensile tests were made on the tubes before and after being sub-

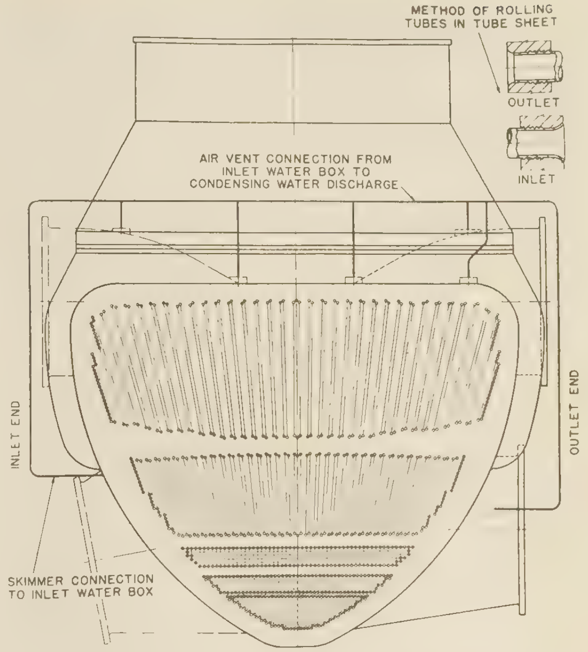


Fig. 4 Layout of Condenser Tubes Showing Steam Laning and Arrangement for Venting Gases Entrained in Circulating Water

TABLE 6 TENSILE TESTS ON 12-INCH LENGTHS OF CONDENSER TUBING BEFORE AND AFTER CORROSION IN EXPERIMENTAL CONDENSER

		Decrease in elongation (elongation, 8-in. gage)		
Tube no.	Loss in tensile strength, per cent	Before, per cent	After, per cent	Decrease, per cent
1	47.4	2.1	2.0	..
2	5.1	25.8	19.5	6.3
4	5.2	45.9	35.3	10.6
5	5.2	47.0	47.0	..
6	2.8	3.4	2.0	1.4
7	3.1	1.4	1.6	..
8	5.3	37.1	34.6	2.5
9	3.3	3.8	3.2	0.6
10	6.1	53.5	49.9	3.6
11	21.9	57.0	39.3	17.7
12	4.6	25.0	23.8	1.2
13	4.1	45.0	52.3	..
14	5.4	9.9	5.0	4.9
15	2.9	42.8	40.0	2.8
17	8.2	36.0	30.2	5.8
18	3.3	32.0	25.1	6.9
19	5.3	1.9	0.5	0.4
20	5.3	44.4	33.6	10.8
21	17.0	55.3	26.5	28.8
22	20.7	60.8	34.5	25.3
23	20.1	45.4	32.9	12.5
24	9.8	16.3	10.1	6.2
25	19.0	67.4	60.2	7.2
26	9.7	2.8	2.0	0.8
27	3.6	9.9	3.3	6.6
30	29.8	9.0	0.9	8.1

TABLE 3 LIST OF TEST MATERIAL

Material	No.	Condition furnished
Aluminum brass	1	Hard-drawn
Aluminum brass	2	Hard-drawn; ends annealed
Aluminum brass	3	Hard-drawn
Aluminum brass	4	Annealed
Aluminum brass	5	Annealed
Aluminum brass	6	Hard-drawn; tensile 110,000 psi; 0.035 mm grain size
Aluminum brass	7	Hard-drawn; tensile 118,000 psi; 0.025 mm grain size
Aluminum brass	8	Annealed; 0.17 mm grain size
Aluminum brass	9	Hard-drawn; Snead shock
Aluminum brass	10	Annealed
Aluminum brass	12	Relief-annealed; ends annealed
Aluminum brass	13	Full anneal A.S.T.M. specification
Aluminum brass	14	Hard-drawn
Aluminum brass	15	Annealed
Aluminum brass	27	Part of tube 14
Cupronickel	16	70/30 annealed
Cupronickel	17	70/30 annealed; Snead A No. 2
Cupronickel	18	80/20
Cupronickel	19	70/30 hard-drawn; ends annealed
Cupronickel	26	70/30 full anneal; A.S.T.M. specification
Admiralty	11	Annealed; Snead E
Admiralty	20	Annealed
Admiralty	21	Annealed; 0.015 mm grain size
Admiralty	22	Annealed
Admiralty	23	Annealed; black skin
Admiralty	24	Relief anneal; ends annealed
Admiralty	25	Full anneal; A.S.T.M. specification
Bronze	30	Hard-drawn

TABLE 4 COMPOSITION OF MATERIALS SUPPLIED

Material	Copper	Tin	Aluminum	Zinc	Nickel	Other elements
Aluminum brass	75% min	1.25% max	1.75–2.50%	Remainder	.....	Small
Cupronickel	Remainder	1.50% max	....	....	29–33%	Small
Cupronickel	Remainder	1.00% max	....	....	19–23	Small
Admiralty metal	70% min	0.90–1.20%	....	Remainder	.....	Small
Bronze	88%	4.00%	Lead 4.00%	4.00%	.....	Small

TABLE 5 ARSENIC CONTENT OF ALUMINUM-BRASS TUBES

Tube number......	1	2	3	4	5	6	7
Arsenic, per cent...	0.031	0.011	0.037	0.006	0.006	0.061	0.071
Tube number......	8	9	10	12	13	14	15
Arsenic, per cent...	0.067	0.050	0.044	0.044	0 037	0.045	0.040

jected to the corrosion tests. The results of these tests are given in Table 6 and Fig. 2.

In presenting the results of the tests in Table 6, percentage of elongation has been reported for each tube as submitted and after being subjected to service in the experimental condenser. Decrease in elongation, if any, is indicated for each tube. Tensile values are reported only as percentage loss in tensile strength, the percentage loss being based on the decrease in the breaking load for the corroded specimen as compared with the breaking load for the same tube but not subjected to the condenser test. The percentage decrease in tensile strength presumably represents the extent to which the effective cross section of the tube has been reduced by the corrosion. This may be due to a general or a local reduction in the cross-sectional area.

Although the tests were made on only one tube of each type before and after the corrosion tests, a check of the relative order of corrosion resistance, based on the amount of decrease in tensile strength, gives, with a few exceptions, a confirmation of the ratings obtained from the corrosion tests.

It is also interesting to note that with but one exception the percentage decrease in tensile strength of the aluminum brasses which showed so favorably in the corrosion tests was slight, ranging from 2.8 to 6.1 per cent. Excluding the No. 1 tube, the aluminum brass gives the lowest average loss for the different types of material which were tested.

Corrosion Tests

The corrosion tests were considered to be the tests of outstanding importance in the selection of the tube material for the proposed condenser installation. These tests were of two types, one, the impingement test, and the other what is known as the condenser test. Of these two tests, the condenser test is the one which is considered to be of primary importance, although the impingement test checks, to a surprising degree, the findings resulting from the condenser test.

Impingement Test

The impingement test was conducted by spraying water at between 90 and 110 F against the inner surface of split-tube samples. The diagram of the test equipment is shown in Fig. 1. On the completion of the test run, the tubes were given a visual examination. They were found to fall into three classes. The first class, and the one least resistant to corrosion, was made up of all those tubes in which the corrosion took the form of a localized deep-pitting attack. Tubes 18, cupronickel, and 24, admiralty, shown in Fig. 5, illustrate this type of attack. The second class, medium to fair corrosion resistance, consisted of those tubes on which the attack was more general or the pitting less pronounced than was the case with the tubes in the first class. Tube 6, aluminum brass, shown in Fig. 5, is illustrative of this group. The third class, or the group most resistant to the impingement attack, was made up of all those tubes in which the surface showed no appreciable roughening, or in which, if pits were present, they were very small. Tube 13, aluminum brass, is illustrative of this group.

As a means of arriving at a somewhat more definite rating of the resistance of the various tubes to the impingement attack, measurements were made of minimum wall thickness at the point of maximum corrosion for each tube. While micrometer measurements were made, the final basis was arrived at by sectioning and polishing cross sections through the corroded areas. These were then measured at 100 magnifications. The results are given in Table 7 and Fig. 3. Ratings have been set up on an arbitrary basis in which maximum resistance is indicated as 1, and minimum 9.

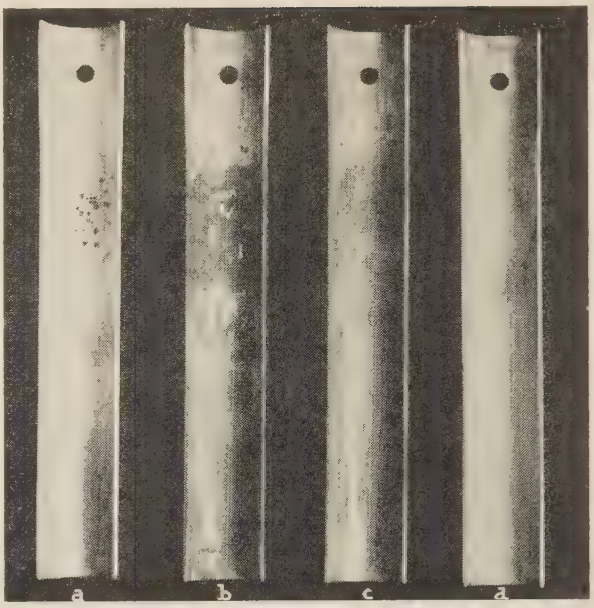


Fig. 5 Specimens From Impingement Test
a Tube 18, cupronickel; localized deep pitting
b Tube 24, admiralty; localized deep pitting
c Tube 6, aluminum brass; general roughening with shallow pitting
d Tube 13, aluminum brass; very slight attack

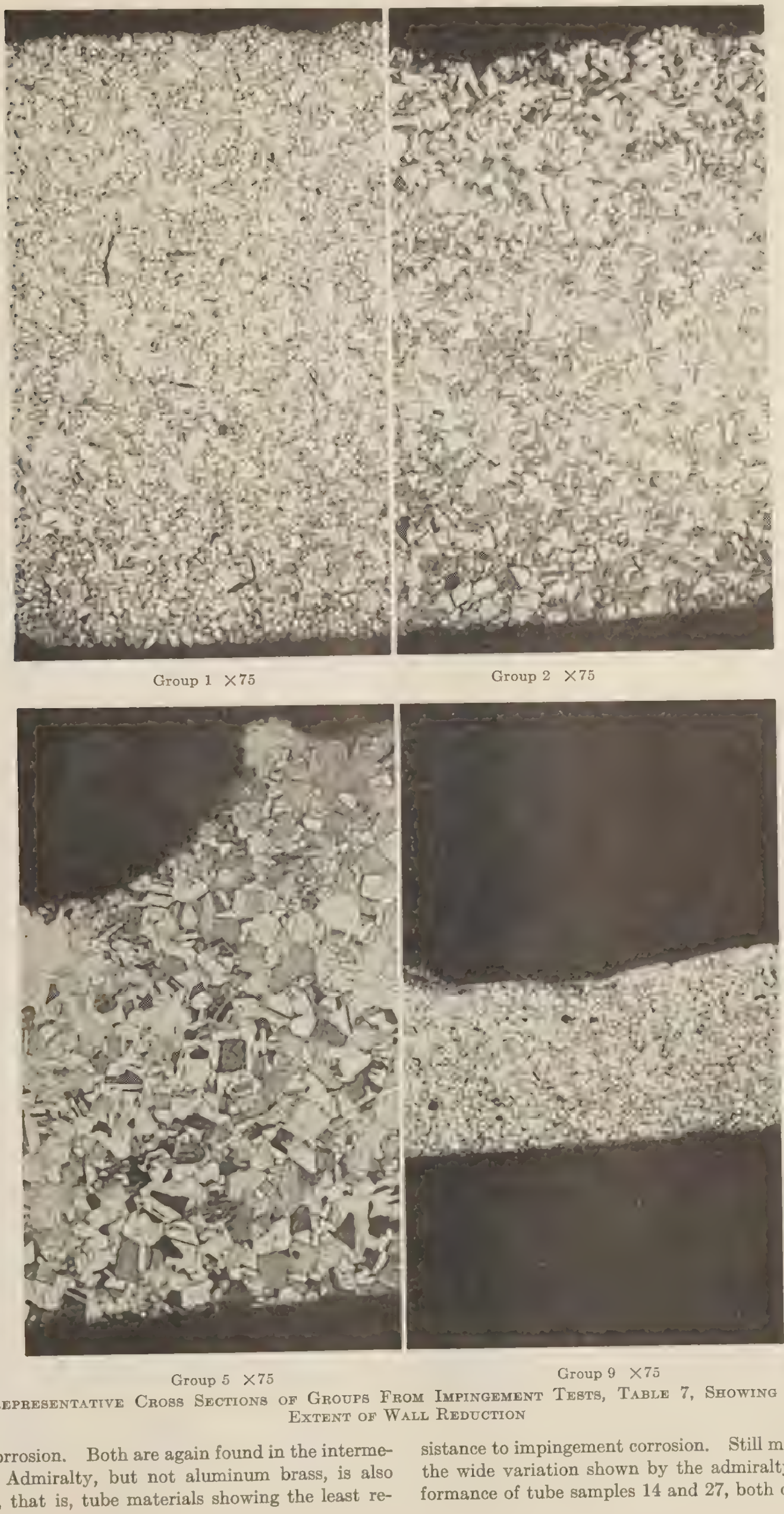
Representative minimum wall sections for groups 1 and 2, best corrosion resistance; group 5, intermediate corrosion resistance; and group 9, least corrosion resistance, are shown in Fig. 6.

An examination of the results, as given in Table 7, shows that no one material behaved consistently throughout the test unless it be the cupronickel, which did not show to any particular advantage. Aluminum-brass and admiralty tubes are both found in the one and two groupings, or those showing greatest resistance

TABLE 7 RESULTS OF IMPINGEMENT TESTS

Rating	Tube No.	Material	Condition	Min wall thickness, in.	Type of corrosion attack
1	12	Al brass	Relief and end anneal	0.051–0.049	Very slight; small pits
	13	Al brass	Relief and end anneal		
	*14	Al brass	Hard-drawn		
2	8	Al brass	Annealed	0.048–0.047	Very slight; small pits
	3	Al brass	Hard-drawn		
	21	Admiralty	Annealed		
3	2	Al brass	Hard-drawn End annealed	0.046–0.044	Slight; pits increasing
	9	Al brass	Hard-drawn		
	30	Bronze	Hard-drawn		
	22	Admiralty	Annealed		
4	5	Al brass	Annealed	0.043–0.040	Varies from general roughening to tendency for fairly large shallow pits
	6	Al brass	Hard-drawn		
	7	Al brass	Hard-drawn		
	10	Al brass	Annealed		
	15	Al brass	Annealed		
	20	Admiralty	Annealed		
	17	Cupronickel	Annealed		
	19	Cupronickel	Hard-drawn		
	26	Cupronickel	Hard-drawn		
5	1	Al brass	Hard-drawn	0.037–0.035	Attack more localized with increasing depth of pits
	4	Al brass	Annealed		
	*27	Al brass	Hard-drawn		
	25	Admiralty	Annealed		
	18	Cupronickel	Annealed		
8	23	Admiralty	Annealed	0.025–0.023	Attack more localized with increasing depth of pits
9	11	Admiralty	Annealed	0.016–0.014	Attack more localized with increasing depth of pits
	24	Admiralty	Relief and end annealed		

* 14 and 27 from same tube.



Group 1 ×75

Group 2 ×75

Group 5 ×75

Group 9 ×75

Fig. 6 Representative Cross Sections of Groups From Impingement Tests, Table 7, Showing Nature and Extent of Wall Reduction

to impingement corrosion. Both are again found in the intermediate groupings. Admiralty, but not aluminum brass, is also found in group 9, that is, tube materials showing the least resistance to impingement corrosion. Still more questionable than the wide variation shown by the admiralty tubes was the performance of tube samples 14 and 27, both of which are from the

same tube, aluminum brass. Section 14 showed no measurable wall reduction, while section 27 showed roughly a 25 per cent decrease at the point of maximum corrosion.

Aluminum brass was found to have performed about equally well in the impingement test, whether it was in the hard-drawn or in the annealed condition. Of the aluminum brasses in groups 1 and 2, good corrosion resistance, three are annealed and two are hard-drawn. In group 5, showing roughly a 25 per cent reduction in wall thickness by pitting, two of the aluminum brasses were hard-drawn and one was annealed. In the same way, the tube materials of the individual manufacturers showed no definite tendencies consistently favoring annealed or hard-drawn aluminum brass.

Conclusions that may be drawn from the results of the impingement tests are:

1 Under the conditions set up in the test procedure, aluminum brass, from a general average standpoint, is superior to all other materials.

2 The resistance of the aluminum brass to impingement corrosion is of the same order, whether in the hard-drawn or the annealed condition.

3 Admiralty metal, from a general average standpoint, is inferior to all of the other materials.

4 The black-skin treatment of the admiralty metal is not effective in increasing the resistance of the metal to impingement corrosion.

5 Maximum resistance to impingement corrosion is not dependent upon any single factor, but embodies the combined effects of composition and manufacturing procedure.

Condenser Tests

The condenser-corrosion tests were made on a miniature condenser which is shown diagrammatically in Fig. 1. Four different grades of tubes were placed in the condenser, namely, tubes of the type known as aluminum brass, cupronickel, admiralty, and one bronze tube consisting of 88 per cent copper, 4 per cent zinc, 4 per cent tin, and 4 per cent lead. The water velocity was 7 fps. The tubes were heated from the blowdown system of a 400-pound boiler. The temperature of the condensing water taken from the river, ranged from 35 F to 80 F. The test was of approximately 22 months' duration with but one serious interruption which was for a period of about one month.

On the completion of the test run, the tubes were removed from the condenser and examined for the purpose of determining the extent to which corrosion had taken place.

As it appeared desirable to run certain tests on the tubes from the condenser for comparison with the tube material which had not been subjected to the experimental run, it was not feasible to split the tubes longitudinally. In consequence, sections of each tube approximately one inch in length and from five different positions were obtained. This procedure not only made it possible to obtain a reasonable accurate survey of the extent of the corrosion, but also made it possible for material to be secured for the mercurous-nitrate and the various physical tests. The locations of these various sections are shown in the upper diagram in Fig. 2.

Corrosion-Resistance Condenser Test. The one-inch sections were examined under a low-power binocular to determine the nature and extent of the corrosion. Using the arbitrary arrangement scheme followed for the impingement-test rating, a comparison was made, first of the sections within any tube, and then on the basis of sections from corresponding positions in all tubes. As before, minimum corrosion was rated as 1 and maximum as 9. Depth of penetration of attack, rather than extent to which attack was general, was used as a basis for the ratings. Thus, a single pit of appreciable depth would rate the corrosion resistance lower than a general attack of little penetration. Micrometer readings of wall thickness, and also metallographic measurements, were made where any question existed as to the extent of a general corrosion attack. Examination was also made of the ends of the adjoining sections in checking final ratings. Inasmuch as the end sections A and F always showed lower degrees of corrosion than the sections from any of the other positions, they were eliminated in the final ratings and do not appear in any of the tabulations.

Table 8 and the upper-rating diagram in Fig. 2 give the results.

Loss-in-Weight Condenser Test. An analysis of the extent of the corrosion was also made on six-inch center sections from each of the tubes in question. The results are given in Table 9 and in Fig. 2.

Corrosion-Resistance Combined Rating

As a means of correlating the various ratings made on the basis of the results of the impingement test and the experimental condenser test, a combined rating was set up. The ratings determined by the impingement test have been given a rating of 1; the ratings from the loss of weight on the six-inch sections, a rating of 1; and the ratings from the corrosion-resistance condenser

TABLE 8 EXPERIMENTAL CONDENSER TEST—RATED ON BASIS MAXIMUM CORROSION SHOWN AT ANY SECTION

Rating	Tube	Material	Condition
1	8	Al brass	Annealed
2	6	Al brass	Hard-drawn
	2	Al brass	Hard-drawn; end annealed
	5	Al brass	Annealed
3	12	Al brass	Hard-drawn; relief anneal
	7	Al brass	Hard-drawn
	10	Al brass	Annealed; Snead E
	1	Al brass	Hard-drawn
4	14	Al brass	Hard-drawn
	27	Al brass	Hard-drawn
	15	Al brass	Annealed
	4	Al brass	Annealed
	9	Al brass	Hard-drawn
	19	Cupronickel	Hard-drawn; end annealed
5	13	Al brass	Annealed
6	23	Admiralty	Annealed; black skin
	26	Cupronickel	Annealed
7	20	Admiralty	Annealed
	21	Admiralty	Annealed
	22	Admiralty	Annealed
	25	Admiralty	Relief annealed; end annealed
8	18	Cupronickel	Annealed
	25	Admiralty	Annealed
9	11	Al brass	Annealed
	17	Cupronickel	Annealed
	30	Bronze	Hard-drawn

TABLE 9 WEIGHT LOSS IN PER CENT IN EXPERIMENTAL CONDENSER-TUBE TEST BASED ON 6-INCH SECTION FROM MIDDLE OF TUBE

Rating	Loss, per cent	Tube	Material	Condition
1	1.0	18	Cupronickel	
	1.2	8	Aluminum brass	Annealed
2	1.5	7	Aluminum brass	Hard-drawn
	1.6	1	Aluminum brass	Hard-drawn
3	2.2	6	Aluminum brass	Hard-drawn
	2.2	14	Aluminum brass	Hard-drawn
	2.3	15	Aluminum brass	Annealed
	2.3	27	Aluminnm brass	Hard-drawn
4	2.6	2	Aluminum brass	Hard-drawn; end annealed
	2.6	10	Aluminum brass	Annealed
	2.9	5	Aluminum brass	Annealed
	2.9	12	Aluminum brass	Relief and end annealed
5	3.0	13	Aluminum brass	Annealed
	3.4	4	Aluminum brass	Annealed
	4.2	9	Aluminum brass	Hard-drawn
6	5.9	19	Cupronickel	Hard-drawn; end annealed
	7.3	24	Admiralty	Relief annealed; end annealed
	7.5	21	Admiralty	Annealed
7	8.1	17	Cupronickel	Annealed; Snead A
	9.2	20	Admiralty	Annealed
	9.7	26	Cupronickel	Annealed
8	10.6	22	Admiralty	Annealed
	11.5	23	Admiralty	Annealed; black skin
9	12.8	11	Admiralty	Annealed
	15.3	25	Admiralty	Annealed
	16.7	30	Bronze	Hard-drawn

TABLE 10 COMBINED RATING OF CONDENSER-TUBE MATERIAL

Impingement Test............1(X)
Experimental condenser
Loss in weight (6-in. section)..1(Y)
Maximum corrosion..........2(Z)

Rating	$\frac{X + Y + 2Z}{4}$	Tube	Material	Microstructure
1	1.2	8	Aluminum brass	Fine grain; 90% recrystallized
2	2.7	6	Aluminum brass	Cold-worked; elongated grains
	2.7	2	Aluminum brass	Cold-worked; nonuniform grain size
	2.7	12	Aluminum brass	Nonuniform grain size
	3.0	7	Aluminum brass	Cold-worked; elongated grains
	3.0	5	Aluminum brass	Small to medium grain size
3	3.2	1	Aluminum brass	Cold-worked; elongated grains; nonuniform
	3.3	15	Aluminum brass	Similar to structure of 8
	3.5	10	Aluminum brass	Small grain size; uniform
	3.7	27	Aluminum brass	Cold-worked; elongated grains
	4.0	13	Aluminum brass	Nonuniform grain size
	4.0	9	Aluminum brass	Cold-worked; elongated grains
4	4.5	4	Aluminum brass	Small grain size
	4.5	19	Cupronickel	Small grain size
5	5.2	26	Cupronickel	Small grain size
	5.5	18	Cupronickel	Small grain size
	5.5	21	Admiralty	Small grain size
6	6.2	20	Admiralty	Medium grain size
	6.2	22	Admiralty	Small grain size
7	7.0	23	Admiralty	Small grain size
	7.2	24	Admiralty	Partial recrystallization 30%
	7.2	17	Cupronickel	Medium grain size
	7.5	25	Admiralty	Medium grain size
	7.5	30	Bronze	Cold-worked; elongated grains
9	9.0	11	Admiralty	Small grain size

Tubes 3 and 16 are not included in the rating in that both were removed in early stages of test. Tube 27 ran only 4713.5 hours, but made a poorer showing than tube 14, which ran the full test period. Tubes 27 and 14 were cut from the same original tube.

test, a rating of 2. The data are given in Table 10 and in Fig. 2, headed "Corrosion-Resistance Combined Rating."

Certain conclusions may be drawn on the basis of the combined rating for the materials tested and the particular conditions simulated by the test procedure:

1 Aluminum-brass tubes are superior to admiralty, cupronickel, and the bronze tubes.

2 Cupronickel tubes are on a par with, or superior to, the best of the admiralty tubes.

3 The condition in which the tube is furnished, that is whether it be in the hard-drawn or annealed condition, is not in itself the determining factor as to whether or not it has good or poor corrosion resistance.

Metallographic Examination

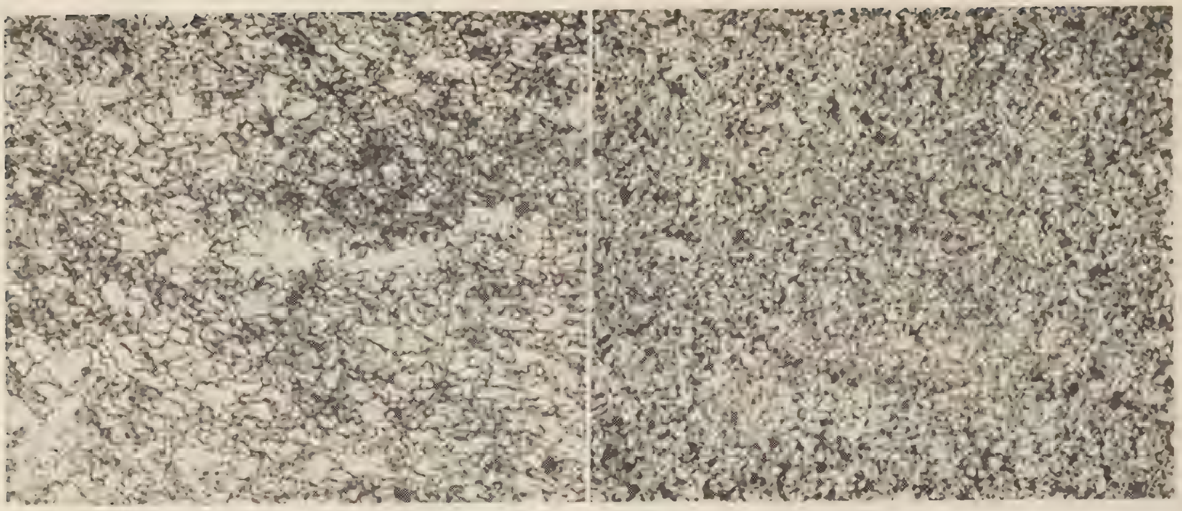
As indicated in the discussion of the results of the impingement and condenser-corrosion tests, there appears to be little, if any, definite tie-up between variations in microstructure and corrosion

Photomicrograph 1 ×250
Tube 14 (27) impingement test rating for tube 14 is 1 and for tube 27 is 5

Photomicrograph 2 ×250
Tube 1 impingement test rating 5

Photomicrograph 3 ×75
Tube 2 impingement test rating 3

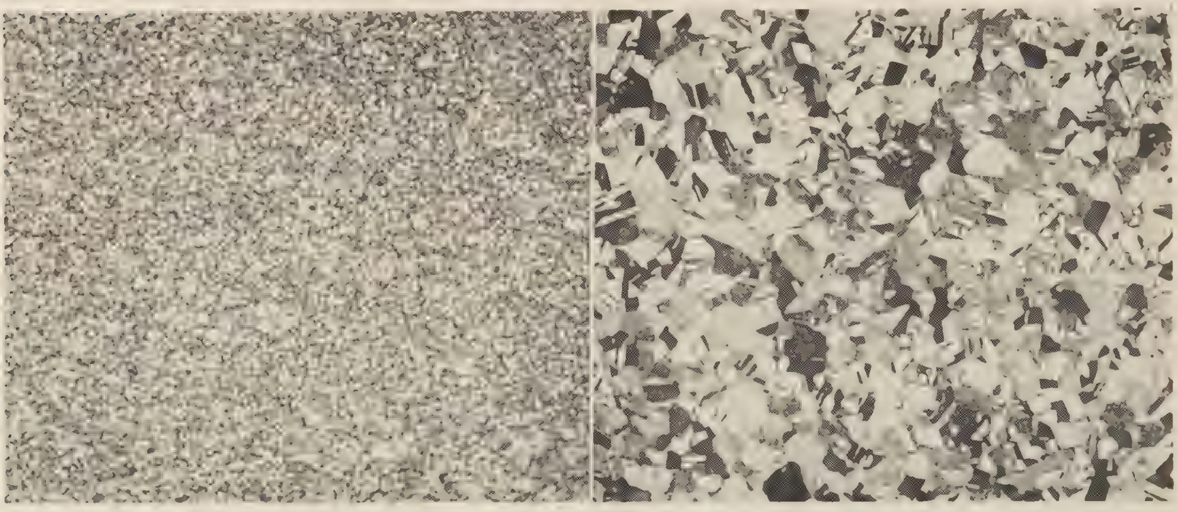
Fig. 7 Characteristic Microstructure for Cold-Drawn Aluminum Brass



Photomicrograph 4 ×250
Representative of tubes 8 and 15
Recrystallization not 100 per cent complete
Impingement test ratings 2 and 4

Photomicrograph 5 ×75
Tube 4, recrystallization complete
Small uniform grain size
Impingement rating 5

Fig. 8 Characteristic Structures for Annealed Fine-Grained Aluminum Brasses



Photomicrograph 6 ×75
Representative of tubes 21 and 11
Small uniform grain size
Impingement test ratings 2 and 9

Photomicrograph 7 ×75
Representative of tubes 20 and 25
Medium grain size
Impingement test ratings 4 and 5

Fig. 9 Characteristic Structures for Admiralty

resistance. As a check on this apparent lack of relationship between microstructure and corrosion resistance, a metallographic examination was made of sections from the original tubes, the impingement specimens, and the condenser test-tube specimens.

Of the 14 aluminum-brass tubes tested in the impingement test (27 is a part of 14), seven were furnished in the hard-drawn condition and seven in the annealed condition. Variations in the microstructure of the hard-drawn material appeared mainly as variations in the grain size or in the uniformity of grain size previous to the final drawing operation. Characteristic variations for the hard-drawn aluminum brass are shown in Fig. 7, photomicrographs 1, 2, and 3. Of the annealed aluminum brasses, variations in microstructure ranged from incomplete recrystallization to appreciable grain growth.

Photomicrograph 4 of Fig. 8 illustrates the slightly less than 100 per cent recrystallized condition of tubes 8 and 15. Impingement and condenser tests place tube 8 as one of the best, if not the best, and tube 15 as one of the poorer or low-ranking aluminum-brass tubes, yet microstructures are for all practical purposes identical. Photomicrograph 5, Fig. 8, shows a microstructure with recrystallization 100 per cent complete and a slight grain growth. This small uniform microstructure which is characteristic of tube 4 is definitely one of the low-ranking aluminum-brass tubes. A check of the corrosion resistance of this tube against tube 27 (the second half of tube 14) shows that this tube, possessing a small uniform annealed microstructure, gives the same corrosion resistance as tube 27, which possesses a characteristic cold-drawn microstructure as illustrated in photomicrograph 1, Fig. 7. Tubes 14 and 27, halves of the same tube and possessing the same cold-drawn microstructure, behave uniformly in

the condenser test, but vary widely in the impingement test, probably as a result of surface conditions.

Cold-drawn aluminum brasses of three different producers are represented by photomicrographs 1, 2, and 3, Fig. 7, showing microstructures for the original 14, 1, and 2 tubes, respectively. No apparent relation exists between the microstructure for these three tubes and their corrosion resistance.

Similar variations or lack of any direct relation between microstructure and the corrosion resistance were also found for the Admiralty metal which was represented in the tests by seven tubes. All admiralty tubes were furnished in an annealed condition and varied from the small uniform grain size of photomicrograph 6 of Fig. 9 to the medium grain size of photomicrograph 7, Fig. 9. The small uniform microstructure of photomicrograph 6 was found in tubes 21 and 11, or the tubes making the best and poorest showing in the impingement test for the tubes of this type. Tubes 21 and 11 were not made by the same producer. The maximum grain size for the admiralty tubes is shown in photomicrograph 7, Fig. 9, representative of tubes 20 and 25.

Owing to the small number of cupronickel tubes in the test and their generally unsatisfactory performance, no photomicrographs of these tubes are included in this paper. The same also holds for the bronze tube.

Conclusions

On the basis of the particular conditions pertaining to this specific investigation, the following conclusions may be drawn:

1 Aluminum brass, for the specific water conditions under consideration, is superior in its corrosion resistance to cupronickel, admiralty metal, and bronze.

2 Microstructure, as such, does not appear to be a controlling factor. A hard-drawn or an annealed material may show equally good corrosion-resistance properties.

3 Cupronickel, admiralty, or bronze tubes are not suitable for use under the proposed water conditions.

4 Internal stresses of an order to produce cracking, under the conditions of the standard A.S.T.M. mercurous-nitrate test, do not necessarily decrease the corrosion resistance of the tubes, nor does their absence necessarily increase the corrosion resistance. It should not be inferred from this conclusion, however, that tubes should be furnished under such conditions of internal stress that they will crack in the mercurous-nitrate test. It is realized that expanding tubes in the tube sheets produce local stresses at these points but it is felt to be the lesser of two evils.

5 Arsenic, while present in some degree in all aluminum-brass tubes, does not appear to be, in itself, a controlling factor. A content of 0.01 and one of 0.07 per cent, when properly associated with other factors, gives equally good results.

6 Phosphorus in traces was found in all of the top-rating aluminum tubes. Its presence or absence does not appear to be a controlling factor.

7 Proper manufacturing procedure is, beyond any question of a doubt, an important factor in the production of highly corrosion-resistant aluminum-brass tubes.

Acknowledgment

The authors wish to acknowledge the valuable assistance and cooperation of the operating department of The Narragansett Electric Company in collecting the basic data of this paper.

Discussion

F. L. LaQue[4] and C. A. Crawford.[4] It is always a problem to choose the proper material for condenser tubes which are to operate with a cooling water with which there has been no previous experience. The authors describe an interesting attack on this problem which presumably resulted in the choice of a satisfactory tube material for the particular installation in which they were interested.

[4] Development and Research Division, The International Nickel Company, Inc., New York, N. Y.

Since a similar technique might be applied in other cases, it would seem to be desirable to make a critical appraisal of the test methods in the light of available data from other sources and practical experience with the materials under consideration.

The most surprising feature of the test results was the relatively small difference in ratings between the 70-30 copper-nickel-alloy tubes and the admiralty tubes installed in the exerimental condensers. There have been fewer opportunities to compare the copper-nickel alloy with aluminum brass, but it may be said that test results and practical installations have shown that the copper-nickel alloy should not regularly be rated below aluminum brass.

The failure of these test results to coincide with such general experience may have been due to such causes as the following:

1 Some peculiar characteristic of the water. In this connection, it would increase the usefulness of the test results if the authors could supply a typical analysis of the water.

2 Effects of minor constituents in the tubes. Research abroad, and more recently in this country, has shown that the performance of the 70-30 copper-nickel alloy may be influenced to an important extent by such minor constituents as manganese, iron, zinc, and carbon. It is not suggested that this was an important factor in the present tests, nevertheless, in order to correlate these test results with other studies along the same line, we would be pleased if the complete analysis of the copper-nickel tubes could be reported. If such analyses are not available, we would appreciate the opportunity to have analyses made, using any portions of the test tubes which may still be available.

Through the kindness of the authors of the paper, an opportunity was provided to make chemical analyses of sections taken from the copper-nickel alloy tubes which were included in the tests. These analyses yielded the following results:

	Composition							
Tube no.	Cu	Ni[a]	Fe	Si	Mn	C	Zn	Sn
16	69.30	29.8	0.22	<0.01	0.42	0.045	0.13	<0.01
17	69.09	30.3	0.04	<0.01	0.38	0.050	0.05	<0.01
18	72.61	24.2	0.15	<0.01	0.42	0.045	2.55	<0.01
19	68.81	30.6	0.037	<0.01	0.48	0.045	<0.01	<0.01
26	68.62	30.8	0.032	<0.01	0.49	0.045	<0.01	<0.01

[a] By difference.

Consideration of these analyses with particular reference to iron contents shows that of the 30 per cent nickel alloys for which data were given in the paper all contained very small percentages of iron. This is especially significant in view of results of research both in this country and abroad which have shown that the resistance of the 70:30 copper nickel alloy to corrosion by salt water is improved to a very considerable extent when the iron content is raised to 0.3 per cent or more. Consequently, the results of tests on these very low-iron-content alloys are probably not indicative of what can be expected from similar alloys having substantially higher iron contents.

3 Nature of the corrosive attack. It is not clear from the numerical results of the test and the description of the nature of the attack, that the types of corrosion which occurred in the experimental condenser, and upon which the materials were rated, were characteristic of those which most frequently lead to failures of condenser tubes.

For example, there apparently was very little inlet-end erosion, even in the admiralty tubes, since it was reported that the end sections (*A* and *F*) of the tubes suffered the least attack. Likewise, there was no mention of dezincification, either general or of the plug type.

It is possible, therefore, that the ratings of the tube materials on the basis of the types of corrosion which occurred under the particular conditions associated with these tests would not apply to service in condensers where inlet-end erosion or dezincification are the principal factors determining the life of the tubes.

4 Duration of the test. So far as the 70-30 copper-nickel alloy and aluminum-brass tubes were concerned, it appeared that the test had been terminated before either material had approached the point of failure. In this connection, it would be interesting to learn the actual values of the maximum depths of attack of the various materials that formed the basis for the ratings in Table 8 of the paper. This point is especially important in so far as the comparison of copper nickel with aluminum brass is concerned. In reporting on British experience, R. May[5] stated:

"At present the cupronickel must be looked upon as the most reliable as regards pitting, and it achieves this distinction not by a complete resistance to pitting, but by the fact that any pits which start tend to widen rather than deepen, so that they develop into small areas of comparatively shallow attack which easily become stifled by the formation of a new protective film. Aluminum brass, on the other hand, probably resists the start of pitting rather better than the cupronickel but, in abnormally corrosive conditions, the pits tend to deepen without becoming wider."

These effects were illustrated by the results of the "impingement" tests reported in Table 7 where the 70-30 copper-nickel specimens apparently developed fairly large shallow pits, whereas, the pits on the aluminum brass varied from small pits to pits of increasing depth, which reached a maximum in the case of specimen No. 27 which, as noted by the authors, was pitted very much more than specimen No. 14 from the same tube.

It is possible that the shallow pitting on the 70-30 copper-nickel tubes was proceeding at a decreasing rate and that, if the test had been prolonged, the relative performances of these alloy tubes would have been better.

5 Nonuniform distribution of water in the experimental condenser. It is a well-known fact that the forces responsible for tube deterioration may vary from point to point in a condenser due to differences in water velocity, turbulence, and the release of dissolved gases. This subject has been discussed in considerable detail in recent papers.[6,7] Among the features of the condenser design that may influence tube performance are the design and location of the water-inlet nozzle and the water box. In the present case, the installation of the horizontal and vertical barriers in the water box could hardly be expected to favor uniform conditions of water flow through the several tubes. For example, the installation of only 5 tubes in the $^1/_4$-condenser-box section containing the copper-nickel tubes, as compared with 15 tubes in the $^1/_2$ section containing the aluminum-brass tubes, tended to favor a higher velocity of flow and greater turbulence in the copper-nickel tubes. It is difficult to estimate the probable extent of this effect without more detailed information as to the dimensions of the water box and inlet nozzle. Also, it seems likely that the vertical barrier between the copper-nickel and admiralty tubes was the source of extra turbulence in the lower half of the condenser.

The advantage of the barriers in preventing water from coming into contact with more than one tube material was probably not very great, since the test unit was a single-pass condenser and the water could not pass through different materials in series. In addition, the mixing of the waters in the inlet and outlet pipes completed the electrolytic circuit so that any electrolytic effects which may have been present were not prevented by the water-box barriers. It is suggested that, if these barriers were to be eliminated and the tubes distributed at random, the effect of variations in water flow would be reduced.

Presumably the condenser for which these tests were made was equipped with aluminum-brass tubes. If experimental lots of the other materials included in the preliminary tests were placed in the condenser, it would be interesting to compare the performance of all these materials in the actual condenser with their performance in the experimental condenser.

F. P. Fairchild.[8] The problem cited had to do with a condensing water of unknown characteristics so far as any actual experience with surface condensers is concerned. While some of us do not have a problem of condensing water as severe as is involved in this paper, water conditions sometimes change rapidly and we are often confronted with new problems due to local conditions.

The writer's company has used admiralty-metal tubes exclusively on all surface condensers of the system. Tube life is comparatively long, being 10 to 15 years on the Hackensack and Passaic River stations and at least 20 years on the Delaware River. Therefore, there has been little incentive to go to more expensive metals, at least until their longer life is definitely proved.

On one condenser at the Kearny generating station, on the Hackensack River, for some as yet unknown reason there has been a case of rapid failure of admiralty tubes by corrosion, possibly due to local turbulence or a high-velocity condition. Within the last year several makes of admiralty-metal and aluminum-brass tubes have been installed in this condenser to determine their relative life. Indications are that aluminum brass shows no pitting or corrosion in 1 year of service, whereas, all the Admiralty tubes are excessively corroded. It appears, therefore, that aluminum brass is a better metal than Admiralty for this particular condenser and water condition.

F. E. Foster.[9] In sharp contrast with England and Germany, the number of publications in this country dealing with the problem of condenser-tube corrosion has been relatively few. The problem of condenser-tube failure even now is far from being solved, although causes and effects of many forms of tube failure are quite well understood. It is gratifying, therefore, that a step has been taken by the authors in the form of an investigation on a semiplant scale into the behavior of certain common alloys used in surface condensers. They have recognized the necessity for treating a particular project as a unique problem. Experience has shown that in most cases the selection of a particular alloy is dependent to a large extent upon the quality of the circulating water to be used. Needless to say, improper condenser design can destroy the benefits of a high-grade expensive alloy.

After carefully considering this paper and bearing in mind the aims and reservations presented by the authors, it would seem desirable for them to give some further explanation of certain of their statements and of the conclusions which were deduced from the data.

Their statement: "The liberation of corrosive gases within condenser tubes, owing to the increase of temperature and the re-

[5] "Condenser Tube Corrosion—Some Trends of Recent Research," by R. May, Trans. Institute of Marine Engineers, vol. 49, 1937, pp. 171–176.

[6] "The Prevention of Failures of Surface Condenser Tubes," by R. E. Dillon, G. C. Eaton, and H. Peters, Trans. A.S.M.E., vol. 59, 1937, pp. 147–150.

[7] "Condenser Tube Life as Affected by Design and Mechanical Features of Operation," by A. J. German, Trans. A.S.M.E., vol. 61, 1939, pp. 125–132.

[8] Chief Engineer, Electric Engineering Department, Public Service Electric and Gas Company, Newark, N. J. Mem. A.S.M.E.

[9] Metallurgist, Consolidated Edison Company of New York, Inc., New York, N. Y.

duction of pressure, is a principal cause of tube deterioration," seems somewhat faulty. Corrosive gases which are no longer in solution in water would seem to be far less detrimental to a tube than when these gases, be they ammonia, hydrogen sulphide, or sulphur dioxide, are in solution. If the authors are referring, however, to the mechanical effects of entrained gases, then almost any gas, active or inert, would cause active tube destruction under certain conditions. It is pertinent to mention at this point that the general consensus of opinion is that the pitting of the inlet end of tubes as the result of entrained gases is the most prevalent cause of tube failure.

The authors endeavored to duplicate in their test apparatus the conditions actually existing in a working condenser. This would imply that, in their miniature condenser, the control of entrained air, water velocity, and flow conditions in the inlet water box had been effected. The apparatus as shown and described is an excellent attempt but appears to meet only partially these requirements. Although details respecting the appearance of the tubes after removal from the test condenser are scanty, the fact that the tube ends were in better condition than the center section bears out this contention. Also there is a possibility of a preferential action in certain parts of the test condenser as a result of the barriers which had been built into it. In addition, it would have been helpful if they had made a determination of the entrained air in the water supply.

The impingement test that has been described here can best be referred to as an accelerated erosion test and, therefore, is subject to all of the shortcomings of this kind of testing.

It is interesting to note that the authors conclude that 70-30 cupronickel tubing is excelled by aluminum brass with respect to impingement resistance. Cupronickel is generally considered, both on the basis of extensive laboratory work and upon service histories, to have the best impingement properties of any of the alloys studied. One of the unique characteristics of cupronickel is its tendency to form wide shallow pits rather than a deep penetrating type usually found. The erratic behavior of certain of the samples in the impingement test, referring to tubes Nos. 11, 21, 14, and 27, may have been the result of inadequate control of the jet velocities, the amount of entrained air, or the shape of the jets. The least likely cause is to be found in variations in the microstructure or composition of the alloys.

While the authors point out that their results obtain only for the conditions of their particular test setup, the conclusion that the degree of induced strain has no effect upon corrosion resistance of the alloys is contrary to all available information relating to the theory of corrosion.

G. C. Holder.[10] The authors of this paper have undertaken a most difficult problem, choosing, as they did for their experiment, a locality where the water is very bad and presenting evidence which has always been controversial.

Surface condensers are most important in attaining economical steam generation. Since the tubes are a vital part of this equipment, it is extremely essential that every effort be made to choose the proper material. This, the authors have attempted to accomplish.

There are many other factors in addition to the choice of material which have a definite bearing on the life of condenser tubes; such as retention of protective deposit on the surface, the amount of air and other gases entrained in the water. Fortunately the improvements in condenser water-box design and more efficient pumps have greatly reduced corrosive influences. All these are conditions over which the tube manufacturer has no control and no manufacturing procedure will overcome these cumulative effects.

[10] Chief Chemist and Metallurgist, Foster Wheeler Corporation, New York, N. Y.

The result has been for the tube manufacturer to produce various modified alloys for which are claimed certain improved qualities. It would now appear that conservatism is to be a thing of the past, as formerly a composition to be accepted had to undergo years of trial; aluminum brass underwent test for a period of some 7 years before being generally recognized by the trade.

It is most certain that cavitation had a decided bearing on tube failure at the inlet end, and that oxygen, carbon dioxide, or both must be present to enhance corrosion. Now it would seem that the logical thing to do would be to remove an annoyance, or at least to curb it, but little has been done along this line. The means of doing this is of no importance, whether it be done by baffling or other means. The writer has in mind an installation of admiralty tubes in a 20,500-sq ft condenser, the circulating water being highly polluted and brackish; the pollution consisting of sewage and heavy chemical industrial waste. The tubes showed signs of failure at the inlet end in about 9 months of service. An assembly to reduce cavitation and break up large-sized air bubbles was installed and, up to the present, no tubes have failed. The condenser has been operating satisfactorily since July, 1931. Means taken to overcome the condition might result in the use of lower priced tubes with satisfactory service life.

The data, given in Table 8 of the paper, indicate that, in ratings 3 and 4, of the aluminum-brass tubes, 6 were hard-drawn and 3 annealed. The hard-drawn tubes showed greater localized corrosion than the annealed tubes, and the weight loss was less for the hard drawn. However, the fact that pitting is predominant is more disturbing than the over-all weight loss. Therefore, annealed tubes would be preferable and would also reduce season-cracking hazards.

The paper, also, indicates that neither the actual size of the grain nor the so-called "uniformity" of size of the grain are controlling factors for corrosion inhibition. While this has been a widely discussed subject, to the writer's knowledge, no one has ever been able to show that regular grain size inhibits corrosion. Nevertheless, it is always a point at issue. Quoting one who has been a member of the brass industry for a great many years: "I know of no usage in the English language which would permit the use of the term 'regular' as descriptive of any grain-size photomicrograph that I ever saw."

In the course of contact with the industry for some years, the writer has examined a number of tube failures and found both coarse- and fine-grain structures, or a combination which have been in service for both long and short periods. Several years ago, a sample of arsenical copper was examined by a disinterested person; his report follows: "This tube shows a very definite lack of knowledge of proper treatment during the drawing process, but the tubes of this installation have a service record of 18 years with no renewals. The circulating water is highly contaminated fresh water with high suspended matter." This person was simply following the trend at that time. However, the actual service record proved him to be 100 per cent wrong.

The question of composition is also baffling. Some years ago, the writer examined a tube of English manufacture, taken from a ship condenser.

The composition was as follows:

	Per cent
Copper	68.00
Lead	0.73
Tin	0.94
Arsenic	0.15
Zinc	Remainder

The microstructure showed an extremely worked condition, nevertheless, the records indicated a service life for this tube of 11 years. It failed then only because of a slug of iron settling on the tube. Many more similar cases could be cited.

As to the addition of a small amount of some elements called "inhibitors" to the present composition to confer certain properties, their virtue is problematical as the authors have pointed out regarding arsenic content. That these alloys will have installations giving excellent service is acknowledged but, unless local conditions are favorable, failure too can be expected.

A point very rarely taken into account in condenser work is the installation of tubes, which is by no means perfunctory and should be well supervised.

For coastal installations, the writer personally prefers the use of ferrules instead of expanded tubes, the ferrules not to be bell-mouthed, but square-shouldered. The reason for this is that any floating object passing the ferrule will be small enough to pass through the tubes and not constitute a point for ensuing local corrosion or partial throttling.

J. T. Kemp.[11] The authors have made a notable contribution to American literature on the subject of condenser-tube evolution and usage. It is greatly to be desired that investigations and practical experience of this nature be reported generously. The experimental work at Providence described is more varied than most that has been done elsewhere. The observations have been analyzed ingeniously and a striking presentation of the relative values of the three alloy types is made.

The work at Providence confirms earlier and as yet unpublished experimental work done by other power engineers and serves to give a comparison of the three most prominent alloys on a series of arbitrarily numbered scales. The comparisons are significant when confined to tube types, especially in so far as the superiority of the aluminum-brass alloy is demonstrated. The conclusion that temper of aluminum brass, either hard drawn or annealed, is not of itself a determining factor in corrosion resistance is sound and within our experience. The weight of American experience in actual service, however, is definitely on the side of the annealed tube. Judged by usage, the annealed tube is favored by central-station operators on a ratio of better than 5 to 1. In part, this is due to considerations of condenser assembly and in part to a factor to which the behavior of hard-drawn sample No. 1 in the tensile test is possibly related, namely, an uncertain tendency for hard tubes to crack after a lapse of time. This tendency can be minimized by an appropriate manufacturing procedure but it persists nevertheless. A satisfactory mercurous-nitrate inspection test does not assure the user that a few of even the best made hard tubes will not crack eventually. This behavior is entirely aside from resistance to corrosion or impingement in its many forms.

In expressing this opinion in regard to sample No. 1, the writer is interpreting the report on a basis of metallurgical experience. In his opinion, there is little advantage in the hard temper. Hard-temper aluminum-brass condenser tubes were first introduced to American engineers by an aggressive English source. Even in the hard temper, the English tubes were so much better than admiralty that they were accepted as a standard. The earlier American aluminum-brass condenser tubes had to be finished similarly to obtain acceptance particularly among marine engineers. This condition, however, was not of long duration.

Comparative tests of different condenser-tube alloys have been made by the engineers of a number of power companies. The most frequent kind of test has been the insertion of a reasonable number of tubes of each of the kinds under consideration in a condenser already filled with admiralty and to watch the tubes under actual operating conditions. In the United States, tests of this kind, involving aluminum brass, date back to 1930 and 1931, in Baltimore and New York, respectively, and to 1931, on the West Coast. Not less than ten such tests have been run or are under way at the present time in central stations of major importance using tidewater for cooling. There have been others inland. So far as these tests have been completed or have been carried to a point of determining alloy policy, they have all shown a superiority of aluminum brass to admiralty. In a few instances, they have also shown aluminum brass to be superior to the cupronickels where the cooling water is seriously befouled.

Accelerated tests and tests of a large number of specimens, in which alloys, finishes, and sources have been compared, have also been run elsewhere. Much the same conclusions have been reached as are now recorded by the authors. Perhaps the earliest of these tests was that conducted by F. R. Knight at Seal Beach, beginning in 1931, in which service tests were paralleled by high-velocity flow through a chain of short samples. More recently, a very complete series of tests has been made in New York by Messrs. H. A. Kidder, H. B. Reynolds, and W. Welch, Jr., all of the Interboro Rapid Transit Company. We hope that the exceptionally complete data of the latter tests may be made public in due season.

It may be opportune to add a few notes on aluminum brass to the sketchy historical data in the paper under discussion. Aluminum brass, as we know it, the alloy of 76 per cent copper, 2 per cent aluminum, 22 per cent zinc was developed in England in the years following the first world war. A cooperative and remarkably thorough research was conducted by the British Non-Ferrous Metals Research Association and by the Corrosion Committee of British Institute of Metals, with the encouragement of the admiralty and the active cooperation of the several English condenser-tube manufacturers. A complete account of this work will be found in the Committee's Annual Reports.

The alloy quickly became established abroad. We in the United States were slower to take it up. At least we may say that certain manufacturing difficulties retarded the commercial production of aluminum-brass condenser tubes until 1931, when the first adequate tube-extruding press to be set up in this country went into operation, together with "tube-reducing rolls," the necessary heavy draw benches and up-to-date annealing equipment. Electric melting, also essential, was established in American brass mills prior to that date. The first commercial installation of American aluminum brass was made in the steamship *Lebore* late in 1930.

Aluminum-brass condenser tubes were given rigorous testing in the laboratories of the company with which the writer is associated from the time the alloy first began to attract important attention abroad. Our early observations on metal of the nominal composition were disappointing and led to some erroneous conclusions at the time. Continued work however brought to light the very important influence of small amounts of a fourth element, arsenic, on the behavior of the metal in contact with salt water. The English brass industry, by the way, is on a fire-refined-copper basis. British condenser tubes in consequence and as a rule are arsenical, both admiralty and aluminum brass. Once the significance of arsenic was appreciated on this side of the ocean, the production of aluminum brass was actively undertaken and has proceeded with increasing volume as the merits of the alloy have become more generally known. Aluminum brass has become the standard condenser tube for all but one of the stations on the East River. The first installation in New York was about 1000 of the English tubes at Sherman Creek in 1930. This was followed by the gradual replacement of admiralty by the English tubes at 14th Street over the next two years. Small lots of American aluminum brass were put in with the English tubes at this station. According to the writer's latest information both are credited with an equal number of operating hours today.

The use of arsenic was extended to admiralty-metal condenser

[11] Sales Engineer, The American Brass Company, Waterbury, Conn. Mem. A.S.M.E.

tubes by the same manufacturer in 1933, and at later dates by several other tube makers. Arsenic definitely improved the resistance of admiralty alloy to the form of corrosion known as dezincification.

It is a matter of great interest that the tests at Providence were run both in simulation of condenser operation in a small heat exchanger and in a jet type of impingement device. The real significance of the latter test may be open to some question. It is a difficult thing to maintain a large number of jets of equal intensity, particularly when the water passing is drawn from a city's harbor. There is, moreover, practically no such sharp erosive action as that caused by a jet in the open air in a surface condenser. In fact there is here something of a suggestion that the significance of the word "impingement" is misleading when applied to condenser-tube wear. Perhaps "erosion" would be a better word. Even then we must remark on the difference in action of a direct jet and the tangential flow of water entering a tube. Of late the thought on the erosive action of water entering a condenser tube has put more emphasis on the action of the air carried by the water, the formation and collapse of vapor bubbles and their influence on the formation and removal of the protective surface films on which the durability of a metal is really based. An interesting reference on this subject and one which goes a long way in explaining the relative behaviors of the several condenser-tube alloys is found in a report[12] by R. May.

So much has been said elsewhere than in A.S.M.E. circles in favor of the tensile test as a means of evaluating alloys exposed to corrosive operating conditions that it is interesting to read of its use in Providence and to be informed that the results of this test generally confirm the others.

From the metallurgical angle, it is our opinion that the paper leaves a great deal unsaid. There is only the most general reference to analyses, arsenic in the aluminum brasses alone being given and a passing reference made to the presence of phosphorus.

It may be that the general conclusions would not have been altered by consideration of the complete analyses. Some at least were in the investigators' hands. Other workers in this field have paid close attention to composition and have had reason to think that high, or low aluminum content was significant under the conditions of their tests. Space might have been found also for typical water analyses or at least the range in salinity, or whether there is a tendency toward the acidic side as might easily develop in such an industrial center as Providence.

A recent experience of the writer with the corrosive effect of over concentrated boiler blowdown leads also to a question regarding the condition of the exteriors of the tubes in the condenser test and whether action on the outside might have influenced the loss in weight and other figures.

N. W. Mitchell.[13] Interesting results are obtained by the authors of this paper using tests which include conditions approximating those of actual condenser-tube service. In general, the results confirm considerable previous knowledge and opinion on the behavior of the condenser-tube alloys tested. It appears that, under the conditions of the test in the experimental condenser, impingement attack is the dominating form of corrosion encountered. No dezincification or other form of corrosion is noted, and it appears likely that the pitting attack which takes place is the result of impingement. It would be interesting to ascertain whether the type of pitting is typical of impingement, i.e., characterized by horseshoe-shaped pits with nodules of metal projecting from their centers. Assuming that conditions in the condenser were favorable for impingement attack, the close correlation between the results of the corrosion test and the separately conducted impingement test is to be expected. It should be noted, also, that since dezincification apparently does not occur, the effect of the addition of elements, such as arsenic and phosphorus, would not be shown.

It is interesting to note that no difference is found in the behavior of annealed and hard-drawn aluminum-brass tubes. This confirms our opinions, derived from actual service experience and test results. The fact that, in the impingement test, two halves of the same tube showed widely divergent results illustrates the eccentricity of corrosion phenomena and the difficulty of obtaining completely convincing results with any limited corrosion test, however well conducted.

The behavior of cupronickel, as shown by the results in the paper, is surprising and contrary to the opinion of most people. It has been widely believed that cupronickel is a generally superior alloy for all-round corrosion resistance, and particularly for impingement attack. There have been a few reports indicating that impingement attack of cupronickel may take place in relatively short periods of time and, in view of the widespread use of this material by the Navy, it would appear that further experimental work should be conducted.

The authors conclude that proper manufacturing procedure is an important factor in the production of corrosion-resistant aluminum-brass tubes. They, however, offer no data whatever to substantiate this conclusion, and the method of manufacture of the various tubes is not given. If they have in mind that the tubes should be free from obvious defects, it is of course agreed that a sound tube is desirable. We do not agree, however, that the method of manufacturing plays any part in the corrosion behavior of the alloy. According to all our data and experience, tubes produced by any one of three or four commonly used methods behave identically with regard to corrosion and impingement attack, all other things being equal. It is well known, of course, that tubes started by any of the common manufacturing methods have identical finishing operations, giving no basis for any difference whatsoever in the final tube.

W. B. Price.[14] In contact, with salt or brackish circulating water, particularly under conditions which result in impingement attack on the tubes, aluminum brass, as a condenser-tube material, has been found to give excellent service. The superior performance of aluminum-brass tubes under the particular test conditions, cited in the paper, is therefore not surprising and appears to be fully substantiated by the test results. The authors also report that cupronickel tubes are on a par with, or superior to, the best of the admiralty tubes. In recent years, 70-30 copper-nickel tubes have come into increasing use where service conditions are particularly severe and, in many cases, they have given 5 times the service life of admiralty metal. In this particular case, the comparison between cupronickel and admiralty metal was based on tests of comparatively few tubes (three 70-30 copper nickel, one 80-20 copper nickel, and seven admiralty tubes) and might, therefore be misleading. In the writer's opinion, a similar test of a greater number of tubes of both alloys might show a definite superiority in the corrosion resistance of copper nickel. Incidentally, the single 80-20 copper-nickel tube was at least equal in performance to the 70-30 copper-nickel tubes in this test, which is contrary to the general experience with these two alloys under varied service conditions. Here again, a test of a greater number of tubes over a longer period of time would probably show a greater difference in the corrosion resistance of these two alloys.

[12] Eighth Annual Report of the Corrosion Committee, British Institute of Metals, 1928.

[13] Metallurgical Engineer, Chase Brass & Copper Company, Waterbury, Conn.

[14] Chief Chemist and Metallurgist, Scovill Manufacturing Company, Waterbury, Conn.

The authors have not given any information in regard to the type or form of corrosion which was found in their examination of the condenser-test specimens. It would be interesting to know in this connection, whether dezincification of the admiralty-metal specimens was experienced and to what degree. Some information was given in the paper concerning the arsenic and phosphorus content of the aluminum-brass samples but similar information was lacking in regard to the admiralty-metal specimens. The sole purpose of the addition of a small percentage of arsenic, antimony, or phosphorus to condenser-tube alloys is to inhibit dezincification of such alloys in service. The presence or absence of these elements in the Admiralty-metal tubes under test, as well as the extent of dezincification of these samples, would be of interest.

It would be helpful to a better understanding of the subject matter of this paper if the authors included an average or representative analysis of the circulating water which was used in the tests.

C. B. Reeves.[15] The author's conclusion that aluminum brass is a superior metal for condenser tubes which are to operate under the conditions of their experiment is supported and extended to include typical operating conditions on ocean-going merchant steamships by experience which the writer's company has had in supplying surface condensers for that service.

The first condensers which the company made and fitted with aluminum-brass tubes as original equipment were put into service in late 1931, and 1932. Twenty-one separate condensers, having a total cooling surface of 62,000 sq ft are included in that group. All twenty-one of these condensers are installed on modern steamships which have been in regular operation on long ocean voyages. Not one of these twenty-one condensers has yet been retubed, nor has any of the condensers which the company supplied with aluminum-brass tubes at later dates.

While our records contain instances in which condensers fitted with admiralty tubes have operated for the same length of time, they form a small minority of the total number of admiralty-tube installations, and the conclusion that aluminum brass is superior cannot be avoided.

With regard to the degree of anneal for aluminum-brass tubes and the method of securing them in the tube sheets, our practice has been to employ relatively hard tubes in accordance with the recommendation of the English manufacturers from whom we secured the greater part of the tubes used in our earlier installations, and to expand them into the tube sheets at both ends to give a positive metal-to-metal seating in order to eliminate all seepage of salt water into the condensate in accordance with the demands of the marine engineers for whom the condensers were built.

Realizing the possibility of setting up dangerous local stresses at the tube ends, the following installation methods were developed to minimize the amount of cold-working necessary to seat the tube tightly. As a first step, close tolerances were established to cover both the outside diameter of the tubes and the finished diameter of the reamed tube-sheet holes into which they were to be expanded, with the result that the clearance between tube and hole was held close to the practical minimum for initial placement of the tubes. Then the bore of each tube-sheet hole was serrated with shallow grooves for a part of its length so that sufficient holding power, against forces tending to pull or push the tubes out of the holes together with complete sealing against leakage, could be generated with a fraction of the amount of rolling required with a plain hole. Operators were given preliminary training in expanding test tubes into model tube sheets

[15] Condenser Department, Ingersoll-Rand Company, New York, N.Y.

TABLE 11 TESTS ON CONDENSER CIRCULATING WATER PROVIDENCE, R. I., OCTOBER 16, 1926

	Kick mv	1-min standard, mv
Intake Water, Temperature, 110 F		
Nickel, 85-15	8	$6^1/_2$
Plain admiralty	17	10
Blackskin admiralty	8	$6^1/_2$
Second test on nickel	12	$8^3/_4$
Discharge Water, Temperature, 80 F		
Nickel	7	4
Plain admiralty	8	5
Blackskin admiralty	$3^1/_2$	3 scant
Dead Low Tide — Discharge Water, Temperature, 110 F		
Nickel, 85-15	13	8
Plain admiralty	11	9
Blackskin admiralty	2	$1^1/_4$

Note: Using distilled water on three samples, needle movement hardly perceptible.
United Electric Railway Company, Winchester Street Station, Providence, R. I. Witnessed by J. C. Millard, F. L. Itschner, B. F. Keene, and C. O. Evans.

until examination of their work showed that they had mastered the feel and torque of the expanding tools to the extent that they would invariably stop the expanding operation as soon as the tube was seated, and well before a dangerous degree of cold-working was attained. The exceptional life that the tubes installed in this way have given in the twenty-one condensers mentioned will demonstrate that careful manufacture can be relied upon to prevent metal failures in the tube ends, resulting from expanding operations.

H. A. Staples.[16] While it is recognized that the conclusions drawn from an accelerated test on condenser tubes or from a model or experimental unit are not final, they are at least indicative and we commend the engineers responsible for this installation in attacking their problem in such an intelligent manner.

This test is of special interest to us for the reason that, in 1925, this original problem was taken up with us, as a general fear existed in an adjacent powerhouse, that surface condensers could not be used in Providence harbor because of bad circulating water.

On the basis of tests on this water dated October 16, 1926 (Table 11 of this discussion), we expressed the opinion that, with properly designed condensers and using the proper alloy, surface condensers could be used. We suggested the installation of a model condenser and submitted drawings for such a unit. The test condenser was built along the lines we suggested and preliminary tests were inaugurated.

It is not possible, in the course of a short discussion, to cover completely all aspects of a paper so filled with pertinent facts, and we may wish, at a later date, to amplify our remarks but at this time we would like to call attention to two significant points brought out in the report:

1 That neither the size of the grain nor the uniformity thereof are factors in the corrosion resistance of condenser tubes.

2 The variation in the percentage of arsenic did not appear to be a factor in the rate of corrosion.

It is a source of deep satisfaction to the writer that we have reached a point in the study of corrosion of condenser tubes when practically all engineers agree that there are three parts to the problem, i.e., the tube, the condenser itself, and the operating conditions.

Today, we find a widespread recognition of the fact that there are many factors affecting the life of condenser tubes which are not related to the tube itself. The tube manufacturer, however, has appreciated the fact that tube quality was one of the variables and therefore, has inaugurated improvements in his manufacturing technique which have resulted in a steady and consistent

[16] Phelps Dodge Copper Products Corporation, New York, N. Y. Mem. A.S.M.E.

improvement in the quality of tubes, metallurgically, physically, and mechanically, so that there are available today condenser tubes embodying a standard of quality impossible to obtain two or three decades ago.

Consider for instance the old practice: Admiralty tubes made from cast shells; the alloy melted in pit fires with natural draft; small unit melts which resulted in wide variations in analysis and quality; cast in iron molds with a built-up core of hay, sand, and clay; and, at one time, horse manure mixed with molasses as one of the core materials, and cold drawn to the finished size.

The next step in the improvement of admiralty was the turning or machining the inside and outside of the shell casting, also the use of the "cupping" process.

Today, this, as well as other tube alloys, are cast in large units from electric furnaces under accurate temperature control. Each "heat" is analyzed. The exterior surface of the billets is removed; the billets are then forged and extruded under high pressure and the resultant "extruded" tube carefully examined before receiving subsequent cold-working and drawing. Dies and tools are chrome-plated or made from tungsten-carbide steel.

Final inspection, as it exists today, on the resultant product is so severe, so searching, and of such a high quality, that it is doubtful if one tube, manufactured under the methods in vogue years ago, would be "accepted" today. Therefore, if tube life was wholly dependent upon quality alone, we would not be faced with the condenser-tube-corrosion problems which exist today.

We all recall that in the early days, in the minds of engineers, when tubes failed, it was not a prima facie case of bad tubes but a conclusive case.

We sincerely trust that the problem will be approached, in the future, to even a greater degree than now, on the basis of an analysis of all the facts; that engineers will continue to endeavor to design condensers and plan their installations in such a manner that the failure of condenser tubes will cease to be a matter of such grave concern as it is at present.

W. R. Webster.[17] There is probably no important engineering material concerning which there have been as many exparte claims, regarding the importance of this, that, or the other composition, treatment, or procedure, unaccompanied by any sufficient supporting evidence, as is the case with the condenser tube.

This paper is therefore welcome in that it reports the results of a scientific investigation intended to determine facts. It amply confirms one already well-established belief, namely, that aluminum brass is superior in corrosion resistance to admiralty metal under normal conditions. It does not, however, support another belief widely held that cupronickel is superior to both. The complete accuracy of these tests is, however, open to some question, due to wide variation in results from sample to sample. This may be because, in all probability, all tubes did not receive an identical exposure to the corroding media. On no other grounds can the differences observed in the case of tubes of approximately the same characteristics be explained, although the law of averages would correct this situation as between tubes of widely different characteristics.

It is regretted that the opportunity was not embraced to test the condenser. This could readily have been done since it would have been an easy matter to make sufficient tubes to equip it from one casting, all manufacturing operations being controlled with extreme accuracy. Should these tubes then have shown as wide variations in corrosion resistance, as occurred in the original test, it would be difficult to avoid the conclusion that each tube had been subjected to a different intensity of exposure. It is hoped that the authors will arrange to conduct such a test since it would add much to the value of the present paper and might help to controvert a widely held belief that the causes of unsatisfactory performance always reside in the tubes.

It is noted that the results give no support to the long-held grain-size theory which never had any evidence to substantiate it, although much to refute it. It is hoped that it is now dead beyond resurrection.

Certain omissions are noted, the addition of which would add greatly to the value of the paper, among which are chemical analyses of the various samples. This is particularly true of the cupronickels as the presence of manganese and iron are known substantially to increase corrosion resistance. In Table 4, of the paper, the presence of tin is indicated, although this ingredient is not normally found in the cupronickels. It is also noted that the 20 per cent nickel does not show the poorest performance of the nickel tubes although experience indicates that it is markedly inferior to the 30 per cent.

It is questionable whether the impingement test represents true impingement attack, as this requires the presence of air in quantity. Further, if air were present in the test it would be doubtful if it were uniformly distributed from jet to jet.

In the final summary, conclusion 7 is not supported by any data coordinating any manufacturing procedure with any variation in performance. It is, therefore, irrelevant to this discussion and in addition has no more application to condenser tubes than to any other high-grade material.

[17] Bridgeport Brass Company, Bridgeport, Conn. Mem. A.S.M.E.

Authors' Closure

The discussion of this paper as given by the various contributors is much appreciated.

The discussions by Messrs. LaQue and Crawford relate almost exclusively to the findings with respect to the copper-nickel tubes. They are at a loss to account for the relatively poor showing of the tubes of this type in comparison with the tubes of the aluminum-brass type. The authors of the paper were equally surprised at the results of the test. They expected copper-nickel tubes to show an outstanding superiority over the aluminum-brass and admiralty tubes.

Regret was expressed that the composition of the water was not given. This was purposely omitted for the reason that no one analysis would tell the true story. The cooling water used for condensing is virtually at head tide. This means that at high tide there would be a tendency for the water to be slightly alkaline, although possibly not greatly so because of the small rise and fall of the tide at that point, and at low tide the water would be slightly acid in character because of the fact that the water would be essentially that of the Providence River and its tributaries which collect sewage and waste from many of the textile plants and communities which lie along their shores.

The authors agree with the thought that these tests may not apply to service in condensers wherein an erosion or dezincification is the principal factor determining the life of the tubes. It must be remembered that, in initiating this investigation, it was the feeling of the authors that through an impingement and condenser test it might be possible to get some idea as to the relative merits of condenser tubes of the aluminum-brass and admiralty compositions. A few copper-nickel tubes were added, with the feeling on the part of all of those concerned in planning the test that there would be no question about the outstanding superiority of the copper-nickel tubes, though with some doubt in the minds of those engaged in the work as to whether or not the extra cost of these types of tubes could be justified, especially if the aluminum-brass tubes made a good showing. The authors are as much disturbed over these findings as are Messrs. LaQue and Crawford, for they agree that, in most instances of severe service, there is no better tube composition than the copper nickel.

The rating of the tubes, as given in Table 8 of the paper, was

based on actual measurements in most cases, although, where there was insufficient material for sectioning, it was based on observation. The ratings are as follows:

Tube no.	Depth of pit, in.	Tube no.	Depth of pit, in.
1	0.001	15	0.002
2	0.025	16	No test
3	Observation	17	0.012
4	0.003	18	Observation
5	0.001	19	Observation
6	0.002	20	0.004
7	less than 0.001	21	0.004
8	0.015	22	0.007
9	0.002	23	0.013
10	0.002	24	0.004
11	0.007	25	0.010
12	0.0Q1	26	0.006
13	0.002	27	Observation (part of 14)
14	0.002	30	0.018

Mr. Fairchild's comments concerning the experience of the Public Service Corporation with condenser tubes at their various stations are much appreciated.

With respect to the comments by Mr. Foster, the authors, in their statement, "the liberation of corrosive gases, etc.," referred mainly to the mechanical effects of entrained gases.

The authors recognize that, in the development of any apparatus, it is possible continually to make refinements, but these refinements take time and are costly. The authors feel that, for the purpose of the investigation, the apparatus used was satisfactory.

The authors agree that corrosion resistance should, within certain limits, be affected by the degree of induced strain. They recognize that the test was not of sufficiently long duration for such a finding to be effective and that was why they guarded their statement about the effect of strain with the statement that their results "obtain only for the conditions of this particular setup."

The comments of Mr. Holder, with inclusions as to his experience, are very helpful.

The discussion by Mr. Kemp has added materially, in the author's judgment, to the value of the paper. The authors have no preference as to whether the water-jet test be described as an "impingement" test or as an "erosion" test. They used the word "impingement," because they felt it more nearly described the type of test.

It is possible that some further light might have been thrown on the findings by complete chemical analysis of the materials used in the test. Personally, however, the authors question if such would have been the case.

Mr. Mitchell questions the propriety of one of the authors' conclusions to the effect that "proper manufacturing procedure is an important factor in the production of corrosion-resistant aluminum-brass tubes." The very fact that the aluminum-brass tubes as a class did not show the same rating would indicate that manufacturing procedure is an important factor, especially so since these tubes came from various concerns. Also, since the paper was written the authors can state from first-hand experience that proper manufacturing procedure is a most important factor.

The authors are in full agreement with the comments by Mr. Price with regard to the copper-nickel tubes. No dezincification was found in any of the tubes tested, though it is possible that, had the tests been continued over a period of several years instead of but 2 years, there would have been dezincification. The reason for not giving the analysis of the circulating water was covered in the discussion of the comments by Messrs. LaQue and Crawford.

Mr. Reeve's discussion of his own personal experience and the experience of his company with condenser tubes in general and aluminum-brass tubes in particular is most helpful.

Likewise, the comments of Mr. Staples are of significance and importance as he discusses in quite some detail manufacturing procedure.

Most of the matters brought up by Mr. Webster have been treated previously in this closure, especially under the comments on the discussion by Messrs. LaQue and Crawford. Particularly is this true with respect to the discussion on the copper-nickel findings. The compositions given in Table 4 of the paper are type compositions, as indicated in the section of the paper entitled "Chemical Composition." The authors have often wondered why maximum tin contents are included in these type compositions for, to the best of their knowledge, it is seldom found. In the analyses which were made under the direction of Messrs. LaQue and Crawford less than 0.01 per cent tin was found.

The support for our statement with regard to manufacturing procedure is given in our discussion of the comments by Mr. N. W. Mitchell.

In conclusion, the authors again wish to thank all those who contributed discussions to this paper, for it is their conviction that the value of a paper is greatly enhanced by a full and general discussion of its contents.

Thermometric Time Lag

By RUDOLF BECK,[1] BRIDGEPORT, CONN.

The purpose of this paper is to present facts and theory of thermometric time lag in a form suitable for industrial users of thermometers. The emphasis will be on the distant-reading type of thermometer, actuated by liquid expansion (mercury or other liquids), gas expansion, or vapor pressure, and mercury-in-glass therometers.

TWENTY years of experience in thermometer work has demonstrated to the author that the phenomenon of time lag is but little understood by industrial thermometer users, either as to character or quantitative value. Specifications written without knowledge of the character of time lags are likely to be meaningless, and cannot be checked accurately without additional explanations. One of the purposes of this paper is to propose a definite form of specification, which permits the use of a simple method of measuring time lag.

The most frequent form in which questions relating to time lag are put, is as follows: How long will the thermometer take to indicate a change of say 10 deg? Or, a specification will stipulate that a change of 10 deg is to be indicated within 30 sec.

No definite answer can be given to the question, nor could the specification be checked without an additional stipulation, for instance, that the thermometer show the changed temperature accurately to within 1 deg. The approach of the indication to the true temperature is asymptotic, i.e., requires theoretically infinite time. If a thermometer takes 30 sec to indicate a sudden change of 10 deg to within 1 deg, it will require another 30 sec to come to within 0.1 deg. This is the most fundamental characteristic of thermometer response, and should be kept clearly in mind.

Let us now consider the specification, "a change of 10 deg is to be indicated within 30 sec." If we set about checking this specification by means of a stop watch, we would get quite different results if we considered an approach of 1 deg as up to temperature, than we would if we considered 0.1 deg as up to temperature. In the second case, the time would be twice that of the first. It may be mentioned at this point that measurement of time lag by attainment of the final temperature is quite unsatisfactory, for the reason illustrated in the example given. The theory of time lag, as explained later, yields a method which permits taking time readings at two points of the scale, which are passed by the indicator fairly rapidly and, therefore, can be timed accurately.

A second form of specification encountered is as follows: "If the medium in which the bulb is inserted changes its temperature at a rate of 1 deg per sec, the thermometer must not lag behind more than 10 deg." This form is in itself complete, but is unsatisfactory, inasmuch as no means are available for determining the true temperature of the medium, as any reference thermometer used will in itself have a time lag. If this time lag is known, it may be added to the difference of the two thermometer readings to obtain the true time lag. However, in order to determine the absolute time lag of the reference thermometer, some method will have to be employed similar to the one to be proposed. Also, to obtain a uniform rate of temperature change in a medium for test purposes, as stipulated by the specification, is much more complicated than the method suggested later; therefore, this method of writing time-lag specifications is impractical.

[1] Research Engineer, Manning, Maxwell & Moore, Inc., American S. & B. Instrument Division. Mem. A.S.M.E.

Contributed by the Committee on Industrial Instruments and Regulators of the Process Industries Division, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

Theory of Thermometric Lag

The theory of thermometric lag is based on Newton's law of cooling, which states that the heat quantity, transferred from one body to another, is proportional to the temperature difference of the two bodies. The rate of temperature rise of a body, as, for instance, a thermometer bulb, will be proportional to the heat transferred to it; therefore, in accordance with Newton's law, also proportional to the temperature difference between the bulb and its surrounding medium.

Those interested in the mathematical development and further refinements of theory are referred to a report[2] by D. R. Harper, 3rd. The author proposes to adopt for industrial purposes the letter L for the time lag, as defined there.[2] The two definitions of L are as follows:

1 If a thermometer has been immersed for a long time in a bath whose temperature is rising at uniform rate, L is the number of seconds between the time when the bath attains any given temperature, and the time when the thermometer indicates this temperature. In other words, it is the number of seconds the thermometer "lags" behind such a temperature.

2 If a thermometer be plunged into a bath maintained at a constant temperature (the thermometer being initially at a different temperature), L is the number of seconds in which the difference between the thermometer reading and the bath temperature is reduced to $^1/e$ times its initial value, where e = basis of natural logarithms = 2.718; $^1/e$ = 0.4 (approx).

The first definition is the most useful one in predicting the behavior of thermometers under various conditions. The second

[2] "Thermometric Time Lag," by D. R. Harper, 3rd, U. S. Bureau of Standards, Bulletin No. 185, 1912, p. 659.

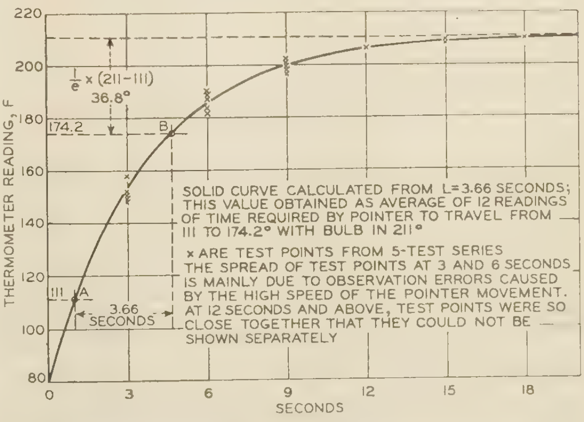


Fig. 1 Time-Temperature Curve of a Mercury-Actuated Dial Thermometer

definition provides a convenient method of measuring L. The time L in seconds is the same in both cases; the mathematical relationship is shown in the Harper reference.[2] Note also that L is independent of the temperature scale used.

Fig. 1 shows the typical behavior of a thermometer bulb. The points have been determined directly by experiment; the curve has been calculated from L, determined by the simplified test method described later on.

In the following sections, we will first discuss the use of L for predicting the behavior of a thermometer under various conditions; list various factors affecting L; then describe methods of determining L and, finally, give actual values of L for different types of thermometer systems, sizes of bulbs, etc.

Uses of Time-Lag Constant L

In the first definition, L represents the time the thermometer will lag behind the actual temperature. In the case of a constant rate of temperature change, this time is the same for any rate of temperature change. Let us suppose L is 10 sec, the temperature rising at a rate of 2 deg per sec, and the thermometer reads 100 deg at a certain instant. By definition, the actual temperature was 100 deg 10 sec previously and, as it is rising at a rate of 2 deg per sec, it is now $100 + (10 \times 2) = 120$ deg. In other words, the thermometer is lagging 20 deg. Generally, we can say that, if the temperature changes at a rate of n deg per sec, the thermometer will at any definite moment show $n \times L$ deg either less or more than the actual temperature, depending upon whether the temperature is rising or falling. Rate of change n may be expressed in any standard temperature scale, as Fahrenheit or centigrade; the result, of course, must be taken in the same scale.

The foregoing relation is true only if the rate of change n has been maintained for a certain length of time, which, for practical purposes, can be taken as 3 to 4 L. This will be explained later.

The second definition states that L represents the time it takes a thermometer, if transferred to a higher temperature T, to move from one indication T_1 to another one T_2, which is $^1/e$ of the original temperature difference $T - T_1$ below the final temperature T. In other words, L is the time for the thermometer indication to proceed from T_1 to T_2, if $T_2 = T - {}^1/e\,(T - T_1) = T - 0.368\,(T - T_1)$. Generally, it will be desired to determine the time required to come within smaller percentages of the final temperature. The factor $^1/e$ brings us to within 36.8 per cent (roughly 40 per cent) of the final temperature. The general formula brought into a convenient form is

$$S_a = L \times \frac{\log {}^1/a}{\log e}$$

where S_a = total number of seconds to come within a times the difference of initial and final temperature, and

$$\frac{1}{a} = \frac{T - T_1}{T - T_2}$$

The following table shows t for various values of a, and gives a fairly good picture of the behavior of a thermometer:

a =	0.50	0.368	0.20	0.10	0.05	0.01	0.001
S_a =	$0.7L$	L	$1.6L$	$2.3L$	$3L$	$4.6L$	$6.9L$

Suppose we transfer a thermometer suddenly from 100 deg to 200 deg, and want to know when the thermometer indication will have reached 199 F. In this case, $T = 200$, $T_1 = 100$, $T_2 = 199$, and $a = \dfrac{200 - 199}{200 - 100} = 0.01$. We will, therefore, take the value 4.6 L for $a = 0.01$. If the change were from 190 to 200 deg, and we again want to know the time of reaching the 199-deg mark, we will use the value $2.3L$ for $a = 0.1$.

To give a yet clearer picture of the thermometer response, actual values are inserted in the following table for a thermometer having a time lag $L = 10$ sec, which has been suddenly transferred from a 100-deg bath into a 200-deg bath:

Thermometer reading, F........	100	150	163.2	180	190	195	199	199.9
Time elapsed, sec............	0	7	10	16	23	30	46	69

It is possible to reduce practically all thermometer-response problems either to the case of a uniform rate of change, or of a sudden change of temperature at the bulb, or to a combination of both. For practical purposes, we can say that, in case of a sudden change, it requires $3L$ to get fairly close to the final temperature, or $5L$ if we want to be very accurate. For a temperature changing at a rate of n deg per sec, the thermometer will lag $n \times L$ deg. If we start from a steady temperature at which the thermometer shows the true temperature, and then change at a

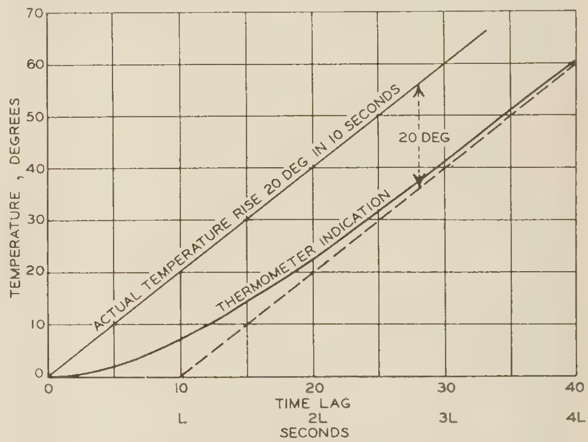


Fig. 2 Thermometer Response for Steady Temperature Changing Into a Uniform Temperature Rise

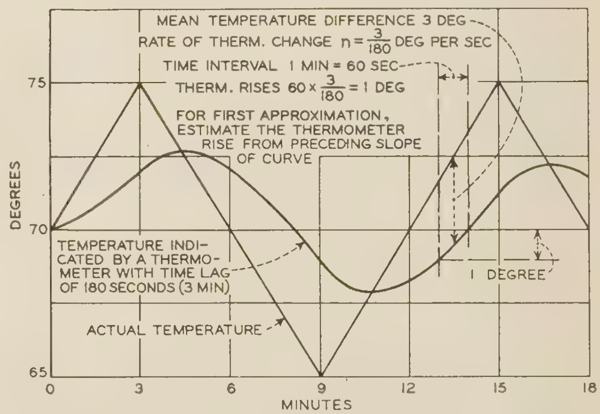


Fig. 3 Determination of Thermometer Response for Arbitrary Temperature Changes

rate of n deg per sec, we will not develop the full temperature lag of $n \times L$ deg until after a time of about $3L$. This type of behavior is shown in Fig. 2.

For cases of a more complicated nature, a graphic method can be used. This method is illustrated in Fig. 3. It is based on a reversal of the formula for temperature lag. We found that $n \times L = T_1$ represents the temperature lag T_1 for a rate of change of n deg. Inversely, if the temperature difference T_e between bath and bulb is known, we can draw the conclusion that the thermometer indication will change at the rate of n deg per sec. The

rate of change will be $n = T_1/L$. In this case, T_1 is the difference between actual temperature and thermometer reading at any time.

Fig. 3 shows the application of this relation to the study of an irregular temperature curve, as may occur in a thermostat-controlled room, and its reaction on the thermometer indications. The upper curve shows the actual temperature; the lower one starts with the initial bulb temperature. A zigzag curve has been chosen to show the relation between actual temperature swings and the swings shown by the thermometer. For calculation, the true temperature curve is divided into convenient elements, and the mean temperature assumed as being effective on the bulb. The corresponding rate of change for the bulb is calculated from the difference between bulb and true temperature.

Factors Affecting Time-Lag Constant L

In the discussion so far, L has been assumed to be a constant. It is constant for practical purposes only, in the same medium and for the same rate of agitation or speed with which the medium passes the bulb. It varies widely for different mediums, such as water and air, and for different speeds of the medium past the bulb. Actual comparative values will be given in the section on "Values of L."

For similar bulb constructions, and bulb lengths more than 3 to 4 times the bulb diameter, L will change somewhat faster than the bulb diameter. In other words, a bulb about twice the diameter will have an L slightly more than twice as large. The use of a separable socket will considerably increase L, because of the space between bulb and socket, and the size of this space will have quite an influence, also the medium used to fill up this space, as mercury, graphite, copper dust, etc.

The formulas derived from Newton's law apply to the heat transfer to the bulb only, and do not allow for any lag between bulb temperature and indication. In all thermometers actuated by liquid expansion, the amount of liquid which is displaced from the bulb is so small that there is almost no frictional resistance in the capillary connection. The interval between the time the bulb reaches a certain temperature and its indication on the dial is, therefore, practically negligible. Also, in liquid-filled distant-reading thermometers, the pressure differential available to push the liquid through the capillary is relatively high, as such thermometers employ pressure ranges of 800 to 2000 psi.

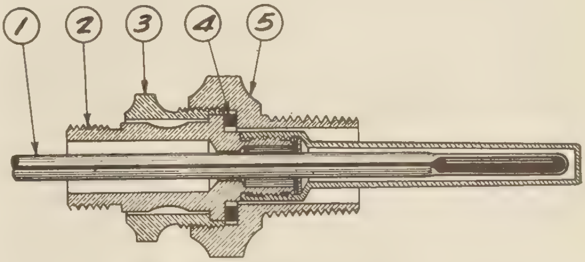
In other words, in a properly constructed liquid-expansion thermometer, time lag is caused almost entirely by the delay in bringing the bulb to the temperature of the surrounding medium, and not by any failure of the bulb to signal its temperature change to the dial or scale.

The conditions are somewhat different for gas-actuated thermometers. Here, the medium transmitting the pressure from the bulb to the indicator is compressible, and a relatively greater volume has to be moved through the capillary tubing. An appreciable pressure drop may occur so that, in addition to the thermal lag of the bulb, there would be a transmission lag of the indication to the dial.

In case of vapor-pressure-actuated thermometers, the conditions are yet more complicated. On a rising temperature, it is only necessary to bring the surface of the liquid in the bulb and the vapor space up to temperature, but not the entire mass of the liquid. On a falling temperature, the entire mass of the liquid must be cooled off. It might, therefore, be expected that the vapor-pressure-actuated thermometer would be faster on a rising temperature. This has been observed in some cases; in other cases the opposite behavior has been indicated by tests. This higher speed on a falling temperature is probably due to cold liquid from the capillary tubing being injected into the bulb which was at a higher temperature than the tubing. A jerky action can frequently be observed in these thermometers, sometimes attributable to the factor just mentioned, or to superheating or undercooling of the active liquid in the bulb. This erratic behavior occurs only where the bulb temperature is changed suddenly through a wide temperature range, a condition which practically never occurs in actual service.

It does make measurement of the time lag of vapor-pressure-actuated thermometers somewhat more difficult. In measuring time lag, it is of course essential to obtain figures which represent the behavior under actual service conditions. These will usually be slow temperature changes from a standard temperature. Occasionally, the statement has been made that time-lag measurements made over a large part of the instrument range do not represent the behavior under actual service conditions. This impression may have been due to a failure to take into account the various factors mentioned.

If time-lag tests are made in accordance with the suggestions in the next section on determining the time lag, figures representative of actual service conditions will be obtained.



6	1	SEPARABLE SOCKET	
5	1	3/4" UNION BUSHING	
4	1	PACKING RING	
3	1	SWIVEL NUT	
2	1	THERM. NECK	
1	1	GLASS TUBE	
NO.	REQ.	NAME	

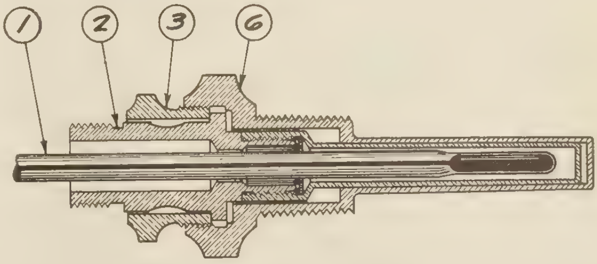


Fig. 4 (Above) Typical Bulb Construction of Industrial Mercury-in-Glass Thermometer

Fig. 5 (Below) Separable Socket Applied to Industrial Mercury-in-Glass-Thermometer Bulb

An exception is the condition in vapor-actuated-thermometers, where the bulb temperature goes from above to below the case or line temperature, as the liquid contained in spring and line must be evaporated and pushed back into the bulb. Some manufacturers now supply specially filled systems of this type in which this lag is eliminated, but without this special feature there is considerable time lag when the bulb temperature changes from above to below case temperature. This lag cannot be covered by any of the formulas cited, and varies so much with different designs that no figures can be presented which would be of any practical value.

For the condition of bulb temperature always above or always below the line and case temperatures, these formulas, however, give a sufficiently practical approximation of the average behavior of vapor-actuated-thermometer systems.

Mercury-in-glass thermometers for industrial applications are usually protected by a metallic well, especially where they are to be inserted in pressure vessels or pipe lines. A typical construction is shown in Fig. 4. As the glass bulbs cannot be made to fit the well accurately, some heat-transfer material has to be used to fill the space between bulb and well. For temperatures below the boiling point of mercury (about 670 F), that metal is the most convenient heat-transfer material. If there is enough mercury in the well to establish contact with the entire active bulb surface and the well at the same time, this type of thermometer will have about the same time lag as mercury-in-steel bulbs of the same over-all diameter. For temperatures above the boiling point of mercury, other heat-transfer materials, such as powdered graphite or copper are being used, but they have a poorer heat conductivity than the mercury. Their effectiveness also depends upon the care with which the heat-transfer material is applied.

Industrial mercury-in-glass thermometers for temperatures above 650 F, therefore, need particular checking as to time lag, when used for installations where quick temperature changes occur.

Whenever it is desirable to remove a thermometer from a pressure vessel while it is under pressure (or vacuum), a so-called "separable socket" is used which forms a pressure-tight pocket. The bulb can then be inserted or removed from the separable socket without interfering with the functioning of the apparatus. This construction is shown in Fig. 5.

The separable socket inevitably introduces an additional time lag (1) because the heat must travel through the metal of the socket wall, and (2) it has to pass the space between the socket and bulb. This space is by far the more important factor of the two.

In the case of industrial mercury-in-glass thermometers a tapered bulb is used extensively, fitting into a socket having the same internal taper. This produces at least a partial metal-to-metal contact. As a standard well size can be used for this type of thermometer, the tapered construction can be produced in reasonable quantities and therefore is commercially feasible. The bulb sizes of the other types of thermometers mentioned vary to such an extent that the tapered-bulb construction would add considerably to the cost. For this reason, the latter has not been commercially adopted. The bulbs of these thermometer types are generally made cylindrical in shape, and the hole in the separable socket is also cylindrical, of slightly larger diameter, in order to permit easy insertion or removal of the bulb. The gap between the two, if filled only with air, will offer a considerable resistance to heat transmission, as air is a poor heat conductor. A widely used method of improving the heat transfer is by application of graphite powder, either dry or mixed with oil so as to form a paste. The oil-graphite mixture should be used only for temperatures below the flash point of the oil, otherwise the oil will cake and make removal of the bulb very difficult.

Methods of Determining the Time Lag L

For determining the time lag by test, we refer to the second definition of the time-lag constant as the time required to come within $1/e$ of the final temperature. Suppose we wish to determine the time lag in water of a thermometer with range 0/300 deg. We provide a stop watch and boiling water (212 F). If the bulb was at room temperature (about 70 deg), we insert it in the boiling water and start the stop watch as the indicator passes the 112-deg mark. This is 100 deg below the final temperature. We want to determine the time required to come within $1/e$ or 0.368 of the difference between starting and end temperature, which is within 0.368 × 100 or 36.8 deg of 212 F. or 175.2 F. The stop watch therefore will be stopped as the indicator passes the 175.2-deg mark. This ends the test. We do not need to wait until the thermometer reaches 212 F. The time indicated by the stop watch is the desired L for water.

The exact starting point of the test is immaterial, as long as we calculate the end point correctly from the difference between the temperature into which the bulb is being immersed, and the starting point. The latter should, however, be at least 20 deg above the bulb temperature, as normal heat-transmitting conditions do not exist the first moment the bulb is immersed. In immersing the bulb of a liquid-expansion instrument into a much higher temperature, there occurs in fact a momentary lowering of the indicator due to the bulb walls expanding before the bulb filling is heated. By starting the test above the bulb temperature, we eliminate this effect which is due to a condition which practically never exists in actual use and therefore need not be considered further.

Fig. 1 shows a comparison between the results obtained by actual determination of the heating curve and a curve calculated from L obtained by the foregoing method. The agreement is very good. The test readings at 3 and 6 sec are somewhat dispersed, due to the difficulty of reading the temperature on the dial while the hand moves quite rapidly. From 9 sec on, the test points cluster very closely around the calculated points. In the abbreviated test, we only measure the time required to move from point A on the curve to point B. The temperature points having been determined beforehand, it is possible to clock quite accurately the times at which the pointer passes the two points.

The test may, of course, be made also with a falling temperature. Any source of heat, as for instance a gas flame, may be used to bring the temperature up nearly to the top of the range. In this case, the test is to be started at least 20 deg below the maximum bulb temperature and, if the bulb has not been heated evenly, a much larger interval should be allowed.

It is desirable to test the thermometer in the medium, and at conditions of circulation which approach actual use as nearly as possible. This is not always possible; to permit estimating time lag from tests in different mediums, and at different rates of circulation, comparative values will be given in the next section.

The most generally available mediums are water and air. Tests in water are quite satisfactory, provided the thermometer scale includes a sufficient part of the water range 32 to 212 F.

Tests in still air are likely to be very misleading, as the thermometer bulb itself will induce air currents which affect the cooling rate the same as changes in air speed. For instance, the bulb of a mercury-in-steel thermometer, $^3/_8$ in. diam, showed an L in air of 270 when 750 F above air temperature, changing to 600 sec upon approaching the air temperature. The air tests should, therefore, be made at a comparatively small temperature interval, or at a known air speed of at least 5 fps, which will eliminate the air currents induced by the bulb itself. Also, in testing with rising temperature, the bulb must be dry and not be cooled below the dew point of the surrounding air, otherwise quite erroneous results will be obtained.

Where calibrating pots with hot oil or lavite* are available, these may be used for thermometers with scales starting near or above the boiling point of water.

The following table gives a few examples of test temperatures, calculated as follows:

$$T_e = T_b - 0.368(T_b - T_s)$$

* Lavite is a salt mixture with low melting point.

Bulb temperature before test, T_a	Bath temperature, T_b	Timing: Starts at T_s	Timing: Ends at T_e	Medium
70	212	112	175	Water
212	70	170	107	Water
120	70	100	81	Air
70	450	350	413	Oil
500[a]	800	600	726	Lavite

[a] In the case of lavite, it is desirable to preheat the bulb, otherwise a crust of solidified lavite may be formed around the bulb immediately after immersion, which would disturb the test.

Molten tin is not recommended for time-lag tests as it does not "wet" the bulb, leaving an insulating layer of air around it.

In the case of liquid-expansion-actuated thermometers, the bath may be stirred with the bulb; this should not be done in

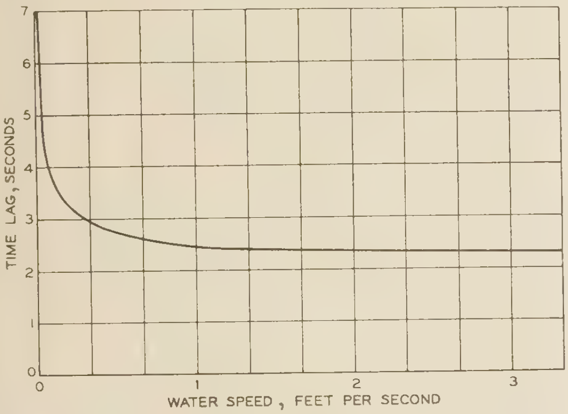


Fig. 6 Change of Time Lag in Water for Various Speeds of Water Past the Bulb

case of vapor-pressure-actuated thermometers, as shaking the bulb throws the liquid around in the bulb when it is only partially filled, and causes a much faster heat transfer than when the bulb is at rest. In other words, moving the bulb around gives results which are about twice as favorable as when the bulb is fixed to the apparatus and is operating under actual conditions.

For vapor-pressure-actuated thermometers, the test should be made over a comparatively short interval, and timing should not be begun until the thermometer indicator has traveled about half way between the original bulb temperature and the bath temperature. For instance, the bulb temperature should be at least 100 deg below the bath temperature, and the timing should be started 50 deg below bath temperature, and ended 18.4 deg below bath temperature.

It is always desirable to take at least three time-lag readings for consistent results. Comparatively short bulbs with considerable metal in the bulb connection will give a higher time-lag reading at the first test, due to heat conduction from the connected metal parts. The test should be repeated until subsequent readings duplicate each other within a few per cent. These readings will represent most actual service conditions in which the superstructure of the thermometer bulb will be at some thermal equilibrium corresponding to average service temperatures.

In every case, the calibration of the thermometer, which is to be tested for time lag, should be checked against the thermometer used to measure the bath temperature, and corrections applied if necessary.

If the industrial-type mercury-in-glass thermometers are used for temperatures considerably above room temperature, for instance 800 F, there will be some heating of the mercury in the scale tube due to heat conduction from the bulb to the metal case surrounding the scale tube. For instance, in one test of a thermometer with 7 sec time lag, it was found that in the first test the thermometer came only within 12 deg of the final temperature, instead of within 2 deg. The difference of 10 deg was due to the low temperature of the metal case, which had not yet reached its equilibrium temperature with the surrounding air. It takes 12 to 25 min to establish this equilibrium. For this reason it is desirable to maintain this type of thermometer for 25 to 30 minutes at the high test temperature, then cool off the bulb alone quickly below the low test point and immediately perform the time-lag test before the case has changed its temperature. This method will duplicate the conditions which obtain when a thermometer is used for a definite normal operating temperature, and its sensitivity to temperature changes from the normal must be determined.

Values of Time Lag L

The figures given in this section, especially the curves, represent average values for well-designed thermometers. Tests in liquids were made for a circulation or stirring effect of at least 1 fps. When using the bulb for stirring, this corresponds to a circular bulb movement 4 in. in diam, 1 turn per sec. The curve in Fig. 6 based on data in the Bureau of Standards paper,[2] shows the change in time lag of a mercury-in-glass thermometer for different speeds of the water, relative to the bulb. It will be noted that above a speed of 1 fps, the value of L changes but slightly. On the other hand, if the liquid is not stirred at all, very much higher values of L may be obtained.

The curves in Fig. 7 show the time lag of mercury-in-steel, vapor-pressure- and gas-pressure-actuated thermometers, for bare bulbs only, in various liquids.

There have been considerable arguments as to which of the three types of thermometers has the smallest time lag. The

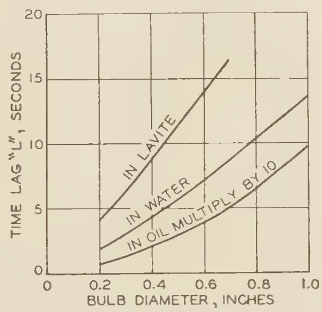


Fig. 7 Time Lag as Function of Bulb Diameter in Water, Oil, and Lavite

author has made a considerable number of experiments, and has found that a single curve quite fairly represents all three types. This may appear inconsistent with the recital of the various factors affecting time lag in vapor- and gas-pressure-actuated types. These factors are present, but it is the business of the thermometer designer to keep their effect as small as possible. For instance, tests with two gas-pressure-actuated thermometers, one with 10-ft and the other with 110-ft line, showed no difference in time lag. As to vapor-pressure-actuated types, the curve applies only where the bulb is either always hotter or always colder than the connecting tubing and the indicator housing.

The effect of separable sockets on the time lag in liquids depends upon the socket material, the thickness of its wall, the space between the bulb and socket, the material used to fill this space and the care with which this material has been applied. Roughly speaking, we can say that in water the socket will increase the time lag 3 to 4 times; in oil, 2 to 3 times. This ratio becomes smaller, as the heat conductivity of the liquid decreases, and as the bulb diameter increases.

Time Lag of Industrial Mercury-in-Glass Thermometers

The protecting well for the glass bulb usually has a diameter of $^3/_8$ to $^7/_{16}$ in. and, if the space between bulb and well is properly filled with mercury, the time lag should be between 3 and 4 sec in well-agitated water. The separable socket with tapered fit approximately doubles the time lag; figures of 7.5 to 8.5 sec have been observed in water. It should be noted that, due to the tapered fit, the time lag is only slightly more than doubled; whereas,

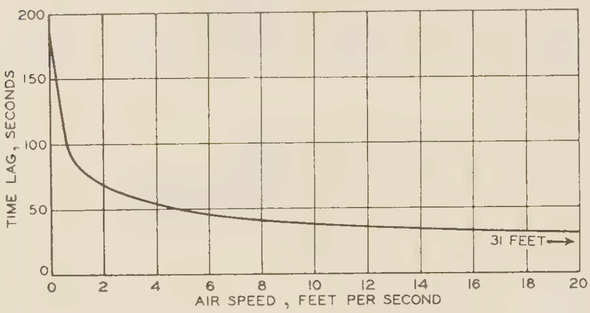


Fig. 8 Change of Time Lag in Air for Various Speeds of Air Past the Bulb

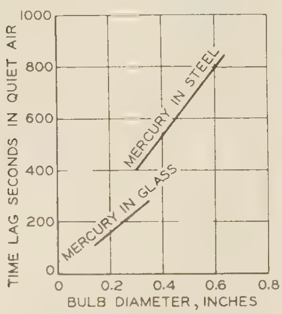


Fig. 9 Time Lag as Function of Bulb Diameter in Quiet Air

in case of the thermometers with straight bulbs described in the previous section, the socket increases the time lag 3 to 4 times in water.

The time lag in oil for the bare-stem thermometer is 15 to 20 sec; for the separable socket, 35 to 45 sec.

Fig. 8, based upon data in the Bureau of Standards paper, shows the change of time lag with changing air speed past the bulb of a mercury-in-glass thermometer. The rapid change in time lag between zero and 1 fps air speed indicates that time-lag figures for thermometer bulbs in quiet air will be extremely unreliable, as any slight change in air drafts may double or halve the time lag. Therefore, if we wish to control the time lag, we must control the air speed. For instance, if we are required to measure at a maximum rate of temperature change of 1 deg in 60 sec, and cannot allow an error of more than 1 deg, the time lag cannot be more than 60 sec. According to Fig. 8, we will need a minimum air speed of 3 fps for this particular thermometer. Fig. 8 shows some data on bulbs in quiet air; the figures correspond to the time lag of 190 sec for zero speed in Fig. 8. Time lags at other speeds can be estimated by using the same proportions of time lag for equal air speeds. For instance, at 20 fps, Fig. 8 shows a time lag of 30 sec; at zero, 190 sec. Fig. 9 shows for a $^3/_8$-in. mercury-in-steel bulb a time lag of about 500 sec. At 20 fps the $^3/_8$-in. bulb would have a time lag of $500 \times {}^{30}/_{190} = 79$ sec. This relation checks reasonably well with actual tests.

Conclusion

The preceding sections give the prospective user of a thermometer sufficient material to determine its behavior under actual service conditions for a definite time-lag figure. What this behavior should be for a certain process must, of course, be determined from the requirements of the process and the characteristics of the process apparatus, and is entirely beyond the scope of this paper.

Once the desired time lag has been determined, it is necessary to include it in the purchasing specifications in a form which is definite and permits easy checking by means generally available. The author proposes the following as a standard form of time-lag specification:

> The time lag of the thermometer in (medium) at a uniform rate of temperature rise must not exceed ( ) seconds.
>
> The time lag is to be determined by transferring bulb at temperature T_a into a (medium) bath at temperature T_b and measuring the time required for the indication to progress from T_s to $T_e = T_b - 0.368(T_b - T_s)$ with bulb at rest and bath stirred moderately. This figure will be accepted as representing the time lag in (medium) at a uniform rate of temperature rise.

Actual figures should be inserted for T_a, T_b, T_s, and T_e. By using specifications of this form, there cannot be any argument as to whether or not they have been fulfilled. Also, the figure given for the time lag permits the prospective user to predetermine the maximum deviations between actual and indicated temperatures which he may encounter in his process, using the formulas given in the section on "Uses of Time-Lag Constant L."

For industrial-type mercury-in-glass thermometers, it may be advisable to add that the bulb should be maintained at the temperature T_b for 30 min before performing the test. The expression "stirred moderately" is somewhat vague, but has been chosen because generally no special equipment is available by which the stirring speed could be measured. When such equipment is available, a definite speed of the liquid past the bulb may be inserted, for instance, 1 fps, or a speed corresponding to actual service conditions.

Frequently, it will not be possible to have the time-lag test performed in the medium in which the thermometer is actually used. In this case one of the mediums discussed previously with temperature characteristics nearest to the actual medium can be selected, or the figures for relations of time lags in different mediums can be used. If the time lag is to be determined in air or gas, the speed of the medium past the bulb in feet per second should be inserted in the specification in place of the wording "and bath stirred moderately."

Appendix—1

Those desiring to study further the theory of time lag should obtain a reprint from the Bureau of Standards bulletin[2] mentioned. Some of the formulas are given herewith in more detail than was justified in the main paper:

T_1 = Original temperature of the thermometer bulb
T_2 = Bulb temperature at time s
s = Time, sec
T_0 = Original temperature of bath
T = Temperature of bath at time s
L = Time lag, sec
r = Rate of temperature rise, deg per sec

Newton's law of cooling can be expressed as follows: The rate of heat transfer is directly proportional to the difference of temperature between two bodies, or

$$\frac{dT_2}{ds} = \frac{1}{L}(T - T_2) \quad [1]$$

The Bureau of Standards paper[2] gives the integration and solution of this equation for several special conditions, of which only two are of interest to us: The behavior of a body (bulb) immersed into another body (bath) of constant temperature, and of a temperature rising at a uniform rate.

The equation for the constant temperature is

$$T - T_2 = (T - T_1)e^{-s/L} \qquad [2]$$

Written in logarithmic form, Equation [2] becomes

$$s = L\,\frac{\log \dfrac{T - T_1}{T - T_2}}{\log e} \qquad [3]$$

which is the form used in the main paper. Solved for L, this formula permits the determination of L from any two timed temperature readings

$$L = s\,\frac{\log e}{\log \dfrac{T - T_1}{T - T_2}} \qquad [4]$$

It will be noted that with $\dfrac{T - T_1}{T - T_2} = e$ the formula becomes $L = s$, which is the relation on which the recommended method of measuring time lag is based.

The equations for the steady-rising bath are as follows

$$T = T_0 + rs \qquad [5]$$

$$T - T_2 = rL + (T_0 - T_1 - rs)\,e^{-s/L} \qquad [6]$$

The condition shown by the curve, Fig. 2, is based on $T_0 = T_1$, in which case Equation [6] becomes

$$T - T_2 = rL - rse^{-s/L} \qquad [7]$$

The term $rse^{-s/L}$ becomes increasingly smaller with time so that the thermometer finally approaches the condition

$$T - T_2 = rL \qquad [8]$$

In equation [8], the temperature lag $T - T_2$ eventually will equal the rate of temperature change times the time lag. Fig. 2 shows rL as a dash line and illustrates the gradual approach of the bulb temperature to this line.

Appendix—2

In the main part of the paper there has been no discussion of the events which occur immediately after transferring a thermometer bulb from one temperature medium into another one at higher or lower temperature. The author felt that they were of no concern to the practical engineer for whom the paper is intended. The methods proposed for determining the time lag were deliberately arranged to eliminate the effects caused by the sudden contact of the bulb with a medium at a considerably different temperature.

In a preliminary discussion[3] of the paper, the question was brought up whether in the theory of temperature regulators the time lag as discussed in the main paper, or that for "sudden bulb contact" would have to be considered. Fig. 1 shows the temperature curve without the "sudden contact" effect; Fig 10, curve A, shows an actual temperature-response type with a drop at the beginning, which is due to the sudden contact of the bulb with hot water. The reason for the drop is as follows: The heat proceeds like a wave from the bulb surface to the interior. During the time of traverse through the bulb wall, the latter expands, whereas, the mercury has not expanded as yet. Therefore, there is an increase in bulb volume without an increase in mercury volume, which causes the temperature indication to drop.

[3] Annual Meeting, Philadelphia, Pa., December 4–8, 1939, of The American Society of Mechanical Engineers.

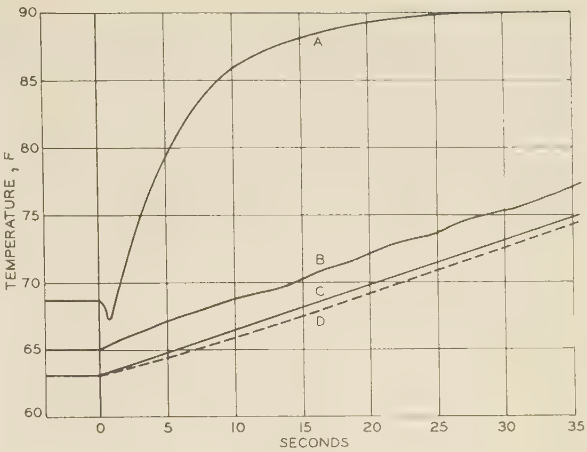


Fig. 10 Comparison of Thermometer Response for Sudden Bulb Transfer (A) and Uniform Rate of Temperature Rise (B)

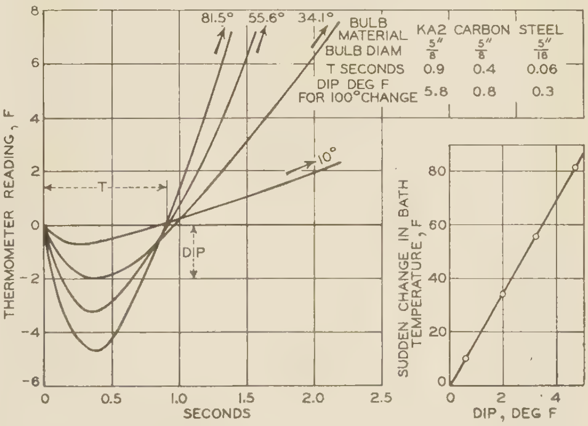


Fig. 11 Dip Effect for Sudden Bulb Transfer

To clear up the effect of this phenomenon on the response of thermometers, the author made some tests with apparatus which happened to be available. Due to lack of time and equipment these tests are by no means complete, but they yielded sufficient quantitative information to give an idea of the true effects, and may serve as a basis for further tests where more accurate information may be required.

The tests were made by means of a recording thermometer with a circular chart revolving once in 10 sec. One of the tests, transferred to cross-section paper, is shown in Fig. 10, curve A. This was made with a mercury-in-steel thermometer system, the bulb consisting of KA_2 stainless steel. Fig. 11 shows the start of the curve from four different tests. Each test was made with a different temperature interval between the bulb and the water bath in which the bulb was suddenly inserted. The result is interesting in several respects.

1 The time required for the reading to come back to the original bulb temperature is practically the same in each case. The variations shown are not greater than might be expected due to imperfections in the experimental equipment.

2 The maximum depression of the reading is directly proportional to the original difference between bulb and bath temperature. At the right of Fig. 11, the depressions have been plotted in relation to the temperature difference and are found to follow a straight-line law very accurately.

3 The time required for this particular bulb to resume a normal temperature rise is about 0.9 sec, and the depression is 5.8 per cent of the original temperature difference. This is for a bulb $^5/_8$-in. diam, heavy-walled, made of 18–8 stainless steel. This steel will show an exaggerated effect of the type discussed here,

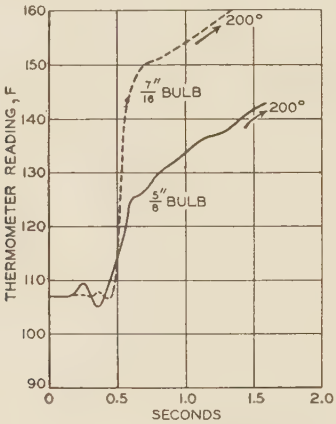


Fig. 12 Vapor-Pressure-Bulb Response

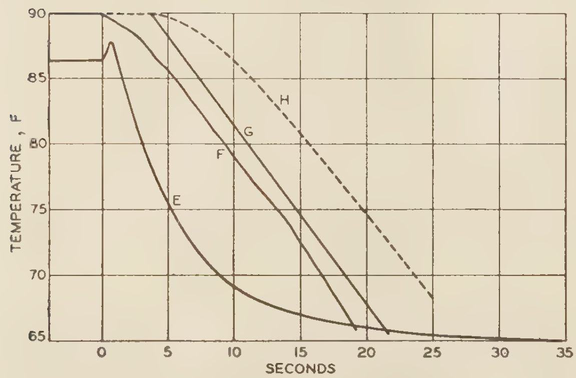


Fig. 13 Comparison of Thermometer Response for Sudden Bulb Transfer (E) and Uniform Rate of Temperature Drop (F)

because it combines low heat conductivity with high thermal expansion. A bulb of the same dimensions, made of carbon steel had corresponding values of 0.4 sec and only 0.8 per cent depression against the 0.9 sec and 5.8 per cent of the stainless steel. The latter was used for the test just because it would exaggerate the type of behavior on which information was desired. The same tests were made for temperature changing from high to low and gave very nearly the same numerical values.

Similar tests were made with vapor-pressure-thermometer systems as shown in Fig. 12. The peculiar behavior at the start is due to the fact that in the test setup the water bath is fixed and the bulb must be moved rapidly when inserting it. As the bulb of a vapor-pressure system is only partly filled, the liquid will be splashed around inside the bulb. The larger wetted surface of the $^5/_8$-in. bulb may account for the earlier initial rise of this indicator. Eventually, the $^7/_{16}$-in. bulb gets ahead of the $^5/_8$-in. bulb (both bulbs were tied together in this test).

The test shows the lack of a consistent depression as found in the mercury-actuated instrument, but it also shows that a certain time (a fraction of a second) is required to get going. The further discussion relating to the mercury-actuated instrument will also apply to the vapor-pressure type.

In control problems, in fact in most industrial applications, we practically never have to deal with instantaneous temperature changes of the magnitudes represented by the foregoing tests. The problems usually are in the form of a transition from a uniform temperature to one rising or falling at a definite rate. Additional tests were, therefore, made for a condition of change from uniform to uniformly rising or falling temperature. This condition was obtained by either opening a steam valve or a cold-water faucet suddenly a definite amount. The steam or water was discharged into the waterbath used for the tests; the water at the same time being stirred by a motor-driven propeller. Fig. 10, curve B, shows the result. There is quite a definite absence of anything resembling the downswing of curve A at the beginning.

Fig. 10 shows the test results for rising temperature; Fig. 13 for falling temperature. Curves A and E show the bulb behavior for sudden bulb transfer, with the very distinct kick in the wrong direction at the beginning. In curves B and F this kick is definitely absent. The straight lines C and G represent ideal test conditions of uniform rate of change; the actual temperature change probably deviated from a straight line due to imperfections in the test outfit, particularly the rate of water circulation was not fast enough to establish perfect mixing from second to second. The lines D and H are the theoretical bulb responses corresponding to the temperature-change lines C and G. The actual test curves C and F deviate from these theoretical curves due to the imperfections of the test setup, but it is quite obvious that there is no effect similar to the "kick" in the sudden-transfer curves A and F, and that they follow the general laws as explained in the first paper, with sufficient accuracy for all practical purposes.

This can be claimed especially because the tests were made with a type of bulb which shows extremes of irregular response in the sudden immersion test.

The only case, in which the characteristic shown in these tests must be considered, is in calculating the time required for the thermometer to reach its final reading. For instance, if a thermometer with a 6-sec time lag is suddenly immersed from 70 into 170 F, and we want to know in what time the thermometer will read 169 F, we found earlier that this will be $4.6 \times 6 = 27.6$ sec. In addition, we found that the "kick" requires about 0.9 sec in the worst case; evidently this period of 0.9 sec will be of no practical importance. If we calculate 27.8 sec, we will probably decide to wait "about" 30 sec to obtain the desired accuracy.

It will be noted that the method of measuring time lag described in the main part of the paper provides that the timing begin at some point above the original bulb temperature, which automatically eliminates the irregular bulb-behavior part from the time-lag test.

The author has not had occasion to make further tests with vapor-pressure- or gas-actuated thermometers. The one test with vapor-pressure bulbs, Fig. 12, showed irregular behavior for a somewhat shorter time element than the mercury-actuated thermometers, and for different reasons. The end effect, from the practical point of view, should be very much the same as in the case of the mercury- (or any liquid) expansion thermometers. Gas-actuated thermometers should show less irregularity than either of the other types.

It appears, therefore, that for all of the mechanical types of distant-reading thermometers, we can from the practical point of view disregard the irregular bulb behavior found in case of sudden immersion of the bulb, and can use the information given in the main part of the paper without applying corrections.

Discussion

M. F. Behar.[4] This paper will be particularly valuable to users of laboratory and of industrial instruments and to purchasing agents. By quoting actual specifications which do not specify, the author has performed a particularly noteworthy and praiseworthy service. In relations between manufacturers and buyers of engineering equipment, as in all other human relations, "Let there be light" should be the precept to be followed. In this competitive world, however, a manufacturer's representative cannot always be blamed for accepting specifications when he is tempted to tell his prospective customer, "Your specifications do not make sense; go back to engineering school!" He curbs himself and sends the absurdity to the home office. At the home office, such impossible or vague lag specifications are an old story and the order is put through. Later, the customer receives his thermometers and "tests" them for lag. His resultant pleasure or dissatisfaction, in most cases, has less to do with the actual lag characteristics (which, of course, remain undiscovered) than with the suitability of the instruments for their applications; with their workmanship; sometimes with their appearance; and in general with factors other than lag.

This is not intended to ridicule specification writers. They are no more blameworthy than the manufacturers' representatives. They merely reflect the present state of the art of writing lag specifications. They are not alone responsible for the present state of this art, nor do they constitute the only group to which the engineering world may look for its advancement.

All of us who have to do with temperature-measuring instruments are responsible for the backward state of the art. Chiefly responsible are those who are in possession of what might be called "advanced knowledge" but whose lips are sealed. It would be more accurate to assert that, between the light which could be shed and the prospective beneficiaries of this light, there is all too often interposed an opaque and almost unliftable veil customarily termed "confidential information."

Closely guarded trade secrets are the reason why the manufacture of thermometers and the specification of their time characteristics both remain arts rather than sciences. The manufacture of thermometers is carried on in hundreds of what the Census Bureau calls "establishments." These vary enormously in size, in modernity, in manufacturing facilities, in types of product, in types of customers, and in every other way, particularly in attitude toward research and development. Until well into the twentieth century the manufacture of thermometers was in the hands—literally in the hands—of expert craftsmen, then and now called glass blowers. Even today, in 1941, there probably are between fifteen and twenty one-man concerns, each headed by an old artisan who brought his skill from Europe, who left his American employer when the latter "went modern," who struck out for himself for love of his craft, who will competently fill your order for a high-grade calorimetric or A.S.T.M. thermometer, but who will not know what you are talking about if you mention "logarithmic decay" or "Newton's Law of Cooling."

There are also large modern mass-production establishments; but these likewise vary enormously in many ways, including attitude toward research and development. Suffice it to point out that a factory financed by chain stores or by an advertising-novelty firm naturally differs from a factory built by repeat orders from great engineering plants and laboratories; an A.S.M.E. pin is seldom seen in the former; in the latter we find many men like the author and other members of our Committee on Industrial Instruments and Regulators.

But the heterogeneity of the temperature-instrument "industry" (not even the Census Bureau has been able to define it as an industry), the enormous disparity of grades of products, of prices, of wages, and particularly of percentages of gross incomes allotted to research and development are factors prolonging the nineteenth-century "trade-secret" kind of competition; these factors tend to increase the opacity of the veil that hides the light, and make this veil more difficult to lift even part way.

The light is not thermometer research and development in general; it is precisely the subject under discussion. It is specific and detailed knowledge of lag characteristics. Most of the improvements in American-made temperature instruments during the last 20 years have been related to lag characteristics, particularly to differential-time effects. The odd aspect of it all, it seems, is that almost every firm thus improving its products believes itself to be the sole discoverer of these effects. Even odder are the instances of firms which have gradually improved the lag characteristics of their instruments by cut-and-try methods and deny the importance of lags other than primary or bulb lag, or remain unaware of the existence of these other lags.

The veil is lifted; not all the way, because the writer's research work has been too circumscribed for the purpose and because confidential information cannot be violated, but just enough to disclose the nature of the light. And now having gone this far, the writer may as well state explicitly the nature of the light. *There is no such thing as a thermometer which obeys exactly the simple law* (the one formulated by Harper in 1912 in his famous Bureau of Standards paper[5]). Every actual thermometer is a composite structure, the components of which not only have different temperature coefficients of expansion or of resistance or of pressure, etc., but also have different time responses to changes in temperature.

No confidences have been violated in making this explicit statement. It is true that this paper nowhere contains any such explicit disclosure, but the author's Appendix 2 reports a sensational departure from the simple lag law, definitely attributable to a metal bulb expanding quickly and appreciably before the contained mercury starts to expand appreciably. The publication of this presentation in the Transactions amounts to more than merely lifting the veil part way; it forces the writer to change his simile and to declare that it amounts to unplugging the hole in the dike, so that from now on we may expect an ever-increasing flood of information on the several components of actual lag curves, an ever-increasing flood of information on just why actual lag curves do not plot straight on semilog paper, as does the simple Newton-Harper law—and on what to do about it in order to minimize errors in temperature measurements.

It is to be regretted that the author worded the closing paragraph of his paper as he did. To be sure, "we can . . . disregard irregular bulb behavior found in case of sudden immersion," but it does not follow that we can "use the information given in the main part of the paper without applying corrections." The main part of the paper expounds the simple Newton-Harper law. The section on "Uses of Time-Lag Constant L" is the one which tells thermometer users how to apply this law. Not only does the author fail to mention that this law applies only to an ideally simple thermometer but his numerical examples, prepared for users of actual (hence, relatively complex) thermometers, include figures which seldom apply to actual thermometers and which are "way off" for most instruments of obvious structural complexity (such as the industrial thermometers

[4] Editor, *Instruments*, Instruments Publishing Company, Pittsburgh, Pa. Mem. A.S.M.E.

[5] "Thermometric Lag," by D. R. Harper, 3rd, Scientific Paper No. 185, U. S. Bureau of Standards, 1912.

illustrated in Figs. 4 and 5 of the paper) under most conditions of use. For instance, he gives the familiar 100 to 200 deg and L = 10-sec example:

Thermometer reading, F............	100	190	199	199.9
Time elapsed, seconds..............	0	23	46	69

as if it were true of all thermometers, his introductory sentence not only being unequivocal but beginning, "To give a clearer picture . . ." With all due respect to a courageous "veil-lifter"—and precisely because of the well-nigh historic importance of his paper—the writer challenges these figures.

Not only are these figures challenged, but the writer maintains the following:

1 The relation shown by the dominant line in Fig. 14 of this discussion (which is simply a semilog plot of the author's main thesis, the Newton-Harper law) is true only of an ideal thermometer in which only one element, say the mercury within the bulb, responds to temperature changes while all other elements remain unaffected by such changes.

2 The great majority of actual thermometers exhibit a composite lag (see secondary solid lines in Fig. 14).

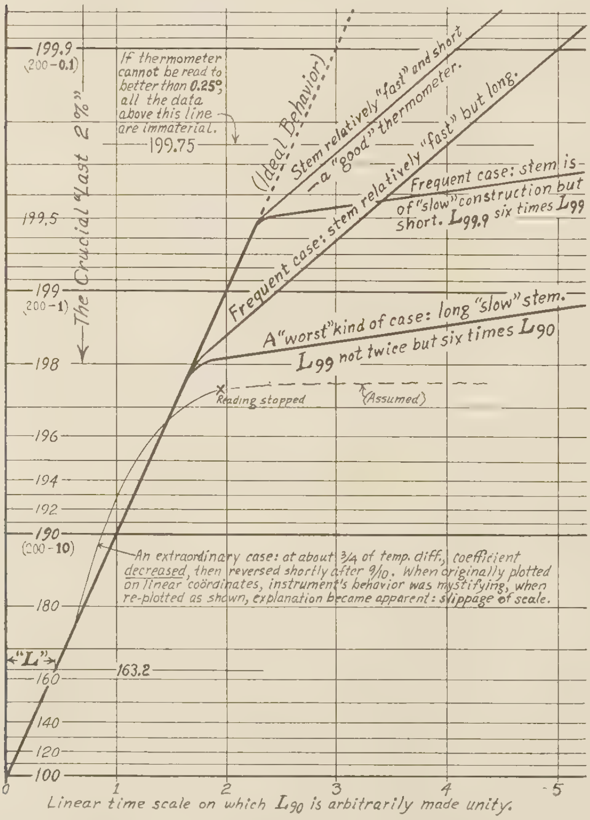


FIG. 14 SEMILOG PLOT DEMONSTRATING THE NEWTON-HARPER LAW AND DEPARTURES THEREFROM

3 In such cases the effect of secondary, tertiary, and other lags is to make the approach to the asymptote slower than that predicted by the "simple law."

4 In many such cases, the observer is deluded into thinking that the mercury or pointer has stopped before it has really stopped; inevitable result, a wrong reading.

5 This error of reading is proportional to the difference between initial and final temperatures (other factors being equal). Therefore, it is most serious in the use of testing thermometers which are taken from the pocket or hook or box at room temperature and immersed into the medium; examples: most laboratory thermometers and practically all "handled long-stem" thermometers used in various food industries, in varnish-making, etc. (Inferentially, then, it is less serious where thermometers are permanently installed. The writer admits this.)

6 The form of lag specification proposed by the author (and by others before him, including the writer when younger) should not be adopted as an American standard, however useful it may be as an educational device for the benefit of purchasing agents and others to whom the whole subject of thermometric lag is a recondite subject. Where precision is imperative, determination of L_{90} and L_{99} (even if magnifying glasses or cathetometric telescopes are thereby necessitated) should be written into the specifications, and tolerances should preferably be expressed in terms of the difference of slopes on semilog paper.

It is to be regretted in this connection that the author did not use logarithmic ordinates for any of his diagrams where temperatures are the ordinates. True enough, logarithmic scales cannot be understood by everybody, but the members of this Society, even the student members, constitute a mathematically minded audience. (However, it must be admitted that many graduate engineers seeing diagrams similar to Fig. 14 of this discussion are confused at first by the unfamiliar graduations. Therefore the writer has lettered in explanatory notations below 199.9, 199, and 190.)

It is on semilog plots that departures from exponential laws show up best; therefore semilog paper serves also as an excellent check on the accuracy of observations. Moreover, the semilog plot magnifies the really important region, i.e., the approach to the final reading of the instrument, which is nothing but a blur of merging lines on linear coordinates. Finally, and most importantly, there is the admirable simplicity of representing different lag coefficients by straight lines of different slopes, hence, conversely, being able to discover the existence of different components of the actual curve of the instrument.

The importance of this last-mentioned reason (straight lines) cannot be overemphasized in connection with writing lag specifications, running acceptance tests, assigning "figures of merit" to competitive instruments, and discussing papers on exponential-function phenomena.

As to how far to carry the magnification, that depends on the grade of the instruments. As a rule the "last 2 per cent" of the temperature difference is of chief interest.

In conclusion, a word as to how the four typical cases in Fig. 14 of this discussion happen to be paired so neatly; the writer purposely drew them that way, to bring out the effects of short-fast, short-slow, long-fast, and long-slow stems. This does not invalidate their actuality—they are truly "representative" cases, not hypothetical cases.

L. M. K. BOELTER[6] AND R. C. MARTINELLI.[7] A thermometer bulb may be represented, in a simple ideal system, as a lumped thermal capacitor C, its thermal resistance being small. The thermal circuit for a thermometer which is suddenly immersed in a fluid (or the fluid temperature is suddenly changed) is shown in Fig. 15 of this discussion.

The thermal conductivity of the ideal thermometer is zero axially (no heat transferred upward and thence to the surroundings) and infinite radially (the immersed portion is at a uniform temperature) and the temperature of the fluid is constant, that is, it possesses infinite thermal capacity ($C_0 = \infty$).

[6] Professor of Mechanical Engineering, University of California, Berkeley, Calif. Mem. A.S.M.E.

[7] Instructor of Mechanical Engineering, University of California, Berkeley, Calif. Jun. A.S.M.E.

The differential equations for the thermal circuit (and of the thermal system) and those of an analogous electrical circuit follow:

Thermal circuit	Analogous electrical circuit
$fA\,\Delta t = V\gamma C_p \dfrac{dt}{d\theta}$	$\dfrac{e}{R} = C\dfrac{de}{d\theta}$
$\dfrac{\Delta t}{\Delta t_0} = \epsilon^{-\frac{fA\theta}{V\gamma C_p}} = \epsilon^{-\frac{\theta}{RC}}$	$\dfrac{e}{e_0} = \epsilon^{-\frac{\theta}{RC}}$

where f = unit conductance between the fluid and the thermometer

Δt = difference in temperature between fluid and thermometer at any time θ after the sudden change in the fluid temperature (or immersion of the thermometer in a fluid of fixed temperature)

Δt_0 = initial difference in temperature between thermometer and fluid

A = area through which heat flows to immersed portion of thermometer

V = volume of immersed portion of thermometer

γ = weight per unit volume of immersed portion of thermometer

C_p = unit heat capacity of immersed portion of thermometer

e = voltage in electrical circuit corresponding to temperature in thermal circuit, i.e., voltage across resistor R

e_0 = initial voltage drop across resistor R

By definition:

$$L = \text{time constant (lag)} = RC = \frac{V\gamma C_p}{fA}$$

The quantity $\dfrac{V\gamma C_p}{A}$ depends only upon the construction of the thermometer (thermocouple) and the degree of immersion and may be tabulated for any instrument by the manufacturer. The unit conductance f is a function of the method of application of the instrument, i.e., the fluid properties and the character of the flow over the thermometric element. The unit conductance may vary a thousandfold for different applications. The manufacturer should tabulate typical magnitudes of the unit conductance for common applications. Users of thermometers can then readily compute the performance of any instrument under transient conditions.

More complex ideal systems, involving distributed capacities and resistances, may be devised more accurately to predict the behavior of a thermometric element. Further, more complicated thermometric elements may be treated analytically (the analogous electrical circuits for which solutions are available may be utilized).

A thermometric element may be employed to record a periodically fluctuating temperature. Concepts developed to solve the electrical circuit may be utilized. Again returning to Fig. 15, the condenser of infinite capacity being replaced by a sinusoidal generator

Thermal circuit	Electrical circuit
$q = \dfrac{\tau}{\dfrac{1}{fA} - j\dfrac{1}{\omega V\gamma C_p}}$	$i = \dfrac{E}{Z} = \dfrac{e}{R - j\dfrac{1}{\omega C}}$

τ = instantaneous temperature of the fluid (with respect to the mean) of angular frequency ω

q = thermal current, Btu per hr

e = instantaneous voltage of angular frequency ω applied to condenser C and resistor R in series

i = electrical current

j = $\sqrt{-1}$

The instantaneous temperature t, measured by the thermometric element (measured above the same mean as τ) corresponds to the voltage across the capacitor C. Thus

$$\frac{t}{\tau} = \frac{\dfrac{-j}{\omega V\gamma C_p}}{\dfrac{1}{fA} - j\left(\dfrac{1}{\omega V\gamma C_p}\right)}$$

or the ratio of the amplitudes of the thermometric element reading t and the fluid temperature variation τ

$$\frac{t_a}{\tau_a} = \frac{1}{\sqrt{1 + \left(\dfrac{\omega V\gamma C_p}{fA}\right)^2}}$$

and the angular lag ϕ of t with respect to τ is given by the expression

$$\tan^{-1}\phi = \frac{\omega V\gamma C_p}{fA}$$

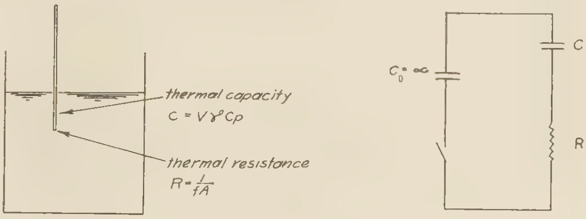


Fig. 15 Thermal Circuit for Thermometer Suddenly Immersed in a Fluid

Periodic thermal variations of complex shape may be solved by expansion in a Fourier series. Thus, it is seen that $V\gamma Cp/fA = CR$ = time constant L is a fundamental constant of the thermometric element but involves its mode of application.

The writers also wish to mention that the time lag for forced-convection applications (flow perpendicular to cylinders) varies inversely as the velocity to the 0.56 to 0.6 power. Thus, the curve illustrated as Fig. 8 by the author should not become horizontal.

W. G. Brombacher.[8] Data on two important classes of instruments have been omitted from the author's presentation of this subject, i.e., the thermocouple and the electric-resistance thermometer. These have been omitted for good reasons, since a full discussion would merit a separate paper. But it should be pointed out that these thermometers can be made with comparatively small lag, particularly the thermocouple thermometer.

The writer is in full agreement with the author on the method of testing which he recommends for inclusion in specifications. However, in instruments of small lag, it may be preferable to specify other readings during the test than those which give the constant L directly; for example, a thermometer originally at 80 F put into a bath at 211 F may be read at 111 F and 201 F. The value of L is easily computed; thus, in the example, using the equations in the paper, $L = 0.44\ S_a$, where S_a is the measured

[8] National Bureau of Standards, Washington, D. C.

time noted. This procedure is particularly useful in testing thermometers with low lag.

Another method which is useful in experimental testing, as contrasted with acceptance testing, is that described by Henrickson.[9] The formula connecting a series of readings made against time, such as given in Fig. 1 of the paper, is in his notation

$$S_a = L \log_e \frac{T - T_1}{T - T_2} \qquad [9]$$

It follows that, if S_a is plotted against $\log \dfrac{T - T_1}{T - T_2}$ on semilog paper, the points should fall on a straight line. The slope of this line is $L/\log_e$, from which L is easily computed. This procedure has the advantage that points which deviate greatly from the line are an indication of the extraneous phenomena which the author has so ably discussed.

It may be of interest to call attention to three papers on the lag of thermometers, used to measure air temperatures. The dependence of time lag of aircraft thermometers upon air speed and upon air density is discussed by Smolar.[10] This paper indicates (1) that the time lag varies inversely as the air speed, and (2) inversely as the square root of the air density. The conclusions are substantiated by experiments. These data are summarized by Peterson and Womack.[11]

If the data given in Fig. 6 of the paper are replotted, time lag against the reciprocal of water speed, as suggested by Smolar, they will be found to be in a straight line, probably within the accuracy of the original data.

H. A. Rolnick.[12] The time lag of distant-reading thermometers of the pressure-spring type is the sum of the time lag of the sensitive element proper and the time lag in the transmission of pressure changes from the sensitive element to the pressure-responsive element such as Bourdon spring. The former time lag will first be discussed.

The value of the time-lag constant given in the paper can be derived from a consideration of the bulb structure and heat-transfer rates. The following equation has been derived on the basis that the temperature throughout the entire bulb is the same.

For a cylindrical bulb

$$L = \frac{WCD}{4H} \qquad [10]$$

where

L = time-lag constant, as defined in the paper
W = density of bulb material
C = specific heat of the bulb (average value)
D = diameter of bulb
H = heat-transfer rate, from fluid to bulb

Equation [10] has been approximately verified by a considerable number of tests. From this equation, it is readily seen that the time lag of the bulb increases directly as the diameter of the bulb as shown in Fig. 7 of the paper.

Another interesting feature of this equation is the dependence of L on the heat-transfer rate of the bulb. For still air, H = 1 to 2 per Btu per sq ft per deg F per hr; for oils H = 10; for water H = up to 50, depending upon the velocity of the fluids past the bulb. So we can have a fiftyfold variation in time lag, depending upon the velocity and nature of the fluid in which the bulb is placed.

For the case where the bulb is placed in a separable socket, a similar derivation of L, the time lag, can be obtained. The following equation is based on a cylindrical bulb in a cylindrical socket.

$$L = \frac{w_3 c_3 d_3^2}{4}\left(\frac{1}{hd_0} + \frac{t_1}{k_1 d_1} + \frac{t_2}{k_2 d_2}\right) \qquad [11]$$

Where subscript 3 refers to the bulb proper A, Fig. 16 of this

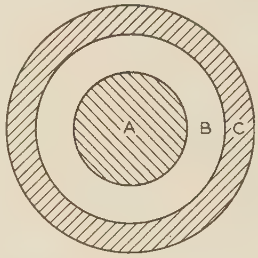


Fig. 16 X-Section of Bulb and Socket

discussion, subscript 2 refers to space B, and C refers to the separable socket.

w = density of bulb material
c = specific heat of bulb material
d = average diameter of section
h = heat-transfer rate from fluid to socket
k = thermal conductivity
t = thickness of space or socket material

Each of the terms in the parentheses of Equation [11] represents resistance to flow of heat. The first term $\left(\dfrac{1}{hd_0}\right)$ is the surface resistance, between the socket and the fluid, the temperature of which is being measured. The second term $\left(\dfrac{t_1}{k_1 d_1}\right)$ is the resistance to heat flow, through the socket or sheath material. The third term $\left(\dfrac{t_2}{k_2 d_2}\right)$ is the resistance to heat flow through the space between the bulb and socket.

Several interesting conclusions can be drawn from Equation [11].

1 The time lag increases with bulb diameter.

2 Where B is air space, this is the largest factor in Equation [11] and contributes most to time lag. A typical set of values for the equation is as follows:

For a $^1/_2$-in-diam mercury-in-steel bulb placed in a steel sheath, having an inside diameter of $^9/_{16}$ in. ($^1/_{32}$-in-thick air space) and a wall thickness of $^1/_{16}$ in.

$\left(\dfrac{1}{hd_0}\right)$ = 1250 (surface resistance) for bulb immersed in rapidly flowing water

$\left(\dfrac{t_1}{k_1 d_1}\right)$ = 14 (sheath resistance)

$\left(\dfrac{t_2}{k_2 d_2}\right)$ = 15,500 (air space resistance)

Filling the air space with oil reduces the air-space resistance to 2000.

[9] "Thermometric Lag of Aircraft Thermometers, Thermographs and Barographs," by H. B. Henrickson, Bureau of Standards, *Journal of Research*, vol. 5, September, 1930, p. 695.

[10] "Determination de la Temperature de l'Air Pendant les Essais en Vol," by V. Smolar, Aeronautical Research Institute, vol. 6, no. 18, Prague, Czechoslovakia, p. 37.

[11] "Electrical Thermometers for Aircraft," by J. B. Peterson and S. H. J. Womack, National Advisory Committee for Aeronautics, Technical Report No. 606, 1937.

[12] Director, Trent Engineering Laboratories, Philadelphia, Pa.

Filling it with mercury reduces the resistance of this space to 180.

A more complete analysis indicates that the time lag is only slightly affected by the superstructure of the bulb, such as the threaded portions of the upper part of the socket or sheath.

Both formulas are based on the assumption that the temperature of the fluid to be measured remains constant. But, when a cold bulb of considerable mass is placed in a hot fluid, the fluid adjacent to the bulb becomes cooler than that in the remainder of the fluid. This results in slower heat-transfer rates and, hence, an increase in time lag. This effect is greater for slow-moving fluids of low thermal conductivity.

In distant-indicating thermometers, the time lag in transmission from bulb to sensitive element is not always negligible. The magnitude of the delay depends upon the type of thermometer.

For vapor-filled thermometer systems, using a long length (100 ft) of capillary tubing, several seconds may elapse for the temperature-sensitive element to indicate completely a pressure change at the bulb. This time increases with the length of capillary, with decreasing capillary diameter, with decreasing absolute pressures in the thermometer system. For recording purposes, this effect may be of slight importance but, for control purposes, a knowledge of the magnitude of this delay is helpful.

For liquid-filled systems, the pressure differences between bulb and sensitive elements, due to sudden increase of temperature are quite high, and considerably greater than for vapor-filled systems. Hence, the delay in transmission will be less than with the vapor-filled systems.

For gas-filled systems, there should likewise be some delay, although no investigation has been made of this up to the present time.

Author's Closure

The discussion shows considerable interest in the problem of thermometric time lag and has added quite some valuable material to the original paper. Mr. Behar very vividly described the situation which was the impetus for the paper. This situation seemed to the author to call for a simplified exposition of the problem; therefore, the paper is mainly based upon the fundamental Newtonian law as applied to the bulb only, itself being considered as a simple body.

Further, the original paper was to be restricted to industrial appliances which, in most cases, use permanently installed thermometers. Such installations also do not require high accuracy of time-lag measurement.

From the point of view of the industrial producer, who has to work toward a minimum of cost, it may be just as wasteful to spend excessive effort on an accuracy beyond requirements as to waste material or to lower quality by insufficient accuracy.

Out of these considerations, the author omitted to discuss composite time lag and, in order to visualize the thermometer response, he intentionally used plain coordinates instead of the semilog type, which had been suggested from several sides.

Since the discussions have expanded the paper from a "primer" into a kind of symposium on time lag, the author is very glad to acknowledge the excellent and interesting chart, Fig. 14, contributed by Mr. Behar and also Mr. Rolnick's calculation of bulb time lag from the physical constants of the bulb.

Messrs. Boelter and Martinelli no doubt could expand their electrical analogy to the composite time lags. The bulb behavior, as described in Appendix 2, should also be susceptible to mathematical formulation based upon the physical constants of the bulb.

An investigation of the characteristics of "secondary" time lags would be a next step. Harper[5] discusses the time lag of a ("fully immersed") Beckman thermometer, in which the secondary time lag has substantially the same character as the bulb time lag. Industrial glass thermometers have secondary time lags of a different character, caused by heat conduction to the case and scale tube. In the first case, we have substantially radial heat flow, and in the second, longitudinal heat flow through variable cross sections, with the added complication of ambient-temperature effects.

A theoretical discussion and formulation of secondary time lags would help in the more exact measurement of time lag and should also suggest means of improving thermometer design. One general rule for design is obvious, namely, secondary time lags should be kept as small as possible.

The author agrees with both Dr. Brombacher and Mr. Behar, that plotting on semilog paper is the most suitable method for accurate investigation of time lags, especially where secondary effects are present. Fig. 14, in Mr. Behar's discussion, shows clearly the advantage of this method in indicating a secondary time lag.

For average industrial installations, however, the method proposed in the conclusion of the paper will be sufficiently accurate and has the advantage of requiring a minimum of time.

The author concludes with the hope that some fellow engineer, who is not too greatly occupied in defense work, will be able to consolidate and enlarge upon the material of this "symposium;" no doubt he will have the cooperation of all the foregoing contributors.

Vaporization Inside Horizontal Tubes

By W. H. McADAMS,[1] W. K. WOODS,[2] and R. L. BRYAN[3]

This paper reports an investigation carried out to determine the changes in the coefficient of heat transfer for the evaporation of a liquid flowing inside a heated horizontal tube. For this research, a semiworks apparatus was constructed, consisting of 48 ft of standard 1-in. copper pipe, provided with 12 individual steam jackets, steam traps, and condensate lines. In the benzene runs, the velocities ranged from 0.26 to 1 fps at the inlet and 80 to 240 fps at the outlet; in the water runs, the corresponding values were 0.27 to 0.85 and 205 to 540 fps. With moderate temperature differences, as the fluid is progressively vaporized, the local over-all coefficient at first increases, goes through a maximum, and then decreases sharply toward values typical of superheating dry vapor. Such "vapor-binding" is attributed to insufficient liquid to wet the wall, small droplets of liquid being carried down the center of the tube, as observed at the entrance to the glass return bend. With high temperature differences, the type of vapor-binding previously observed when boiling liquids outside submerged tubes, where (due to excessive temperature difference) a vapor film insulates the tube wall from the bulk of the liquid, was encountered.

Nomenclature

The following nomenclature is used in the paper:

h_{avg} = average film coefficient for entire boiling section, Btu per hr per sq ft of inside surface, divided by the length-mean temperature difference from inner wall to fluid inside of the tube

p = cumulative weight per cent of feed vaporized, based on cumulative heat transferred and feed rate

P = gage pressure on steam header, psi

q/A = local heat flux, Btu per hr transferred in an individual jacket, divided by 0.88 sq ft of inside surface of copper tube

U = local over-all coefficient q/A divided by difference (deg F) between saturation temperature of steam and temperature of fluid. In the boiling section, temperature of fluid was taken as saturation temperature

U_{avg} = average value of U for boiling section, based on length-mean temperature difference

W = feed rate, lb per hr

Introduction

Vaporization of liquids inside tubes is of such industrial importance that considerable experimental research has been devoted to measuring heat-transfer coefficients under such conditions. The usual method of reporting the results of such investigations has been to base the heat-transfer coefficient on the "apparent" temperature difference. For steam-heated apparatus, the apparent over-all temperature difference involves the condensing temperature of the steam and either the outlet temperature of the partially vaporized liquid or an average of the inlet and outlet temperatures. Where tube-wall temperatures have been measured by thermocouples, the length-mean wall temperature may be substituted for the condensing-steam temperature in order to obtain apparent "film temperature differences." Although apparent over-all or film coefficients are of great value to the designer, who usually knows only the apparent temperature difference, they can safely be used only when design conditions are almost identical with those used in obtaining the data. Thus, the use of a longer or shorter tube might cause considerable variation in the effective temperature difference and capacity without affecting the apparent temperature difference.

Some investigators (1, 2, 3, 4, 5)[4] have reported "true" temperature differences obtained by means of a traveling thermocouple which measures the temperature of the fluid at various distances along the inside of the tube; the true temperature difference being taken as the length-mean average of the local temperature differences between the inside wall and the fluid.[5]

Since the fluid velocity may vary by several hundredfold during passage through the tube, a large variation in the local heat-transfer coefficient throughout the length of the tube would not be unexpected. Even if heat-transfer coefficients based upon true temperature differences and total heat flux in the boiling section are known, the designer still does not know whether these same coefficients would prevail with a different heated area.

The principal object of the work to be described in this paper was to study the variation in local heat-transfer coefficients in a semiworks apparatus in which large percentages of the liquid feed were vaporized. The results obtained when boiling pure benzene and pure water are given. An analysis of the pressure drops, and the results obtained when boiling mixtures of benzene and lubricating oil will be published subsequently.

Apparatus and Experimental Procedure

The apparatus used in this investigation was a special semicommercial evaporator, Fig. 1, consisting of four horizontal 12-ft lengths of copper pipe, 1-in. standard pipe size, connected in series by glass return bends. Each copper pipe carried three separate steam jackets, 3 ft 2 in. long. Condensate was collected separately from each of the twelve steam jackets in order that changes in the rate of heat transfer along the pipe could be measured. Dry steam from a cyclone separator was supplied to the jackets, as shown in Fig. 1. The vapor-liquid mixture leaving the last pass was separated, the vapor condensed at atmospheric pressure, and the two liquid streams continuously mixed and returned by a pump through an orifice to the first pass. An over-all heat balance could be obtained from the rate of condensation of steam and the rate of flow and temperature rise of the water in the con-

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Contributed by the Process Industries Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

[5] T. B. Drew has reported that insertion of a metal thermocouple with a rough surface inside of a vertical steam-heated glass tube resulted in radical changes in the boiling action, minimizing superheat and causing boiling to commence earlier in the tube than when the couple was absent. Such a phenomenon should not be encountered in commercial tubes where the additional nuclei for bubble formation offered by the thermocouple are small in number compared with the numerous nuclei along the metal wall.

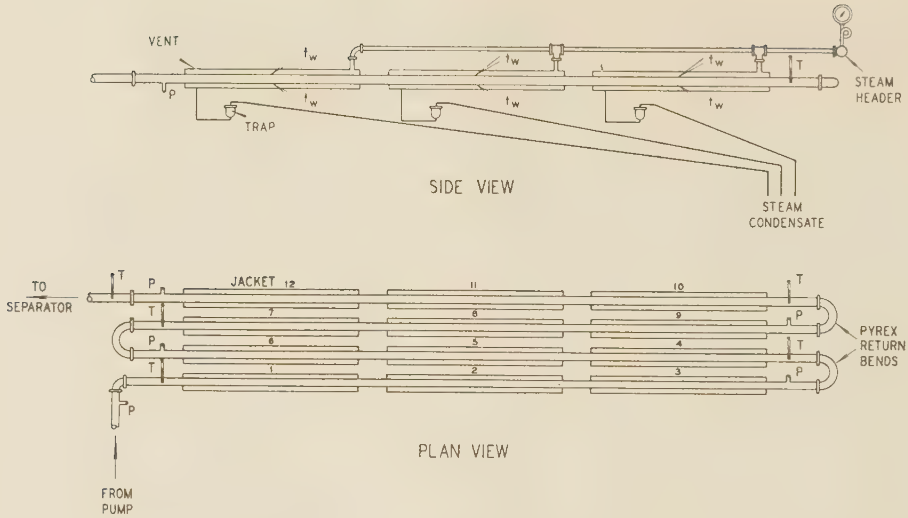

Fig. 1 Diagram of Four-Pass Evaporator
(This diagram shows pressure taps P, thermometers T, tube-wall thermocouples t_w, Pyrex-glass return bends, and twelve steam jackets.)

denser. The feed rate was determined by means of an orifice and checked by means of a heat balance in those runs where the liquid was completely vaporized. Fluid temperatures were measured at the entrance to each pass. Pressure taps were provided near the end of each pass, and pressure drops across each pass and the preceding bend were measured by mercury-water differential manometers. At the midpoint of each jacket, two thermocouples were embedded in the top and bottom of the outer wall of the copper pipe. Dropwise condensation of steam was promoted by the use of octylthiocyanate. To permit removal of noncondensable gases, a steam vent was provided at the top of each jacket, near the condensate outlet.

A series of runs was made on commercial benzene, followed by a series of runs on benzene-oil mixtures and, finally, a series of runs on distilled water. High-speed photographs of the glass return bends were taken by stroboscopic light during some of the runs on water.

During the initial runs on benzene, the thermocouples in the tube wall read so nearly like the condensing-steam temperature that the film heat-transfer coefficients were almost identical with the over-all heat-transfer coefficients. Consequently, the thermocouple readings were dispensed with during the remainder of the runs on benzene, in order to reduce the length of the run and the possibility of variation in steam pressure during the run. The runs on boiling water were started 3 months after the initial runs on boiling benzene; during the intervening time the thermocouples were impaired to such an extent that their operation during the runs on boiling water was unsatisfactory. One observer could record all temperatures, pressures, and orifice readings, while a second operator collected the steam condensates. The twelve condensate streams were collected almost simultaneously by starting the collection from each line at 5-sec intervals. After steady operating conditions had been maintained for 20 min, the actual test period lasted less than 10 min.

The operation of the apparatus was characterized by fluctuating flow in the glass return bends, accompanied by fluctuations of all readings, except the rate of flow of water through the condenser. Reported readings represented a visual time average over the space of several seconds. The saturation temperature of the fluid, corresponding to the observed fluid pressure, varied over smaller limits than did the observed fluid temperature. Hence, all temperature differences in the boiling section were based upon the saturation temperature of the liquid, although the observed temperatures were usually within 1 C of the saturation temperature. The observed temperatures fluctuated over a range of as much as 3 C or more.

From the steam-condensate readings and the known temperature and pressure of the fluid, the cumulative weight per cent p of the feed vaporized, up to and including any specified jacket, was calculated. The local over-all coefficients U in Btu per hr per sq ft per deg F, were calculated by dividing the heat flux in each jacket by the difference between the saturation temperature of the steam and that of the liquid.

Results and Discussion of Runs on Boiling Benzene

The results of three representative runs on boiling benzene are shown in Figs. 2, 3, and 4. In Fig. 2, it is noted that the rate of heat transfer prior to boiling is high, due to the high temperature difference. In the boiling section, the heat-transfer coefficient starts at 290 in jacket No. 3, passes through a maximum of 710 in jacket No. 9 (in which the cumulative weight per cent of the feed vaporized p had increased from 55 to 70), and then decreases to about 60 in the last jacket, where p was 95 per cent. The initial increase in the heat-transfer coefficient might be ascribed (a) to the increasing temperature difference in the later jackets, due to pressure drop, or (b) to the increase in the cumulative weight per cent of the feed vaporized.[6]

The run illustrated in Fig. 3, was taken at a higher steam pressure than that of the run illustrated in Fig. 2, at approximately the same feed rate. Consequently, 67 per cent of the feed was vaporized in the first three jackets and the fluid leaving jacket No. 8 was superheated vapor. It is noted that jackets Nos. 4 and 7, which immediately follow a return bend, gave rates of heat transfer and heat-transfer coefficients considerably higher than obtained in the adjacent jackets. It is also noted that jackets Nos. 5 and 6 gave heat-transfer coefficients substantially the same as obtained when dry benzene vapor was being superheated in jackets Nos. 8 to 12, inclusive.

[6] That the increase in the heat-transfer coefficient is not due solely to increasing temperature difference is best proved by reference to data (6) on boiling benzene-oil mixtures, in which a similar increase in the heat-transfer coefficient with increase in the cumulative per cent vaporization occurred despite a decreasing temperature difference as the benzene was boiled out of the oil.

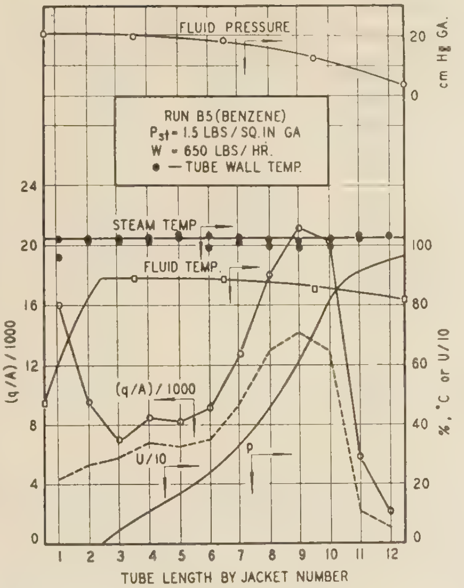

Fig. 2 Data and Calculated Values for Run No. B-5 on Benzene, With Steam at 1.5 Psi Gage

(Decrease in U beyond p of 70 is caused by vapor-binding, due to insufficient liquid to wet the wall.)

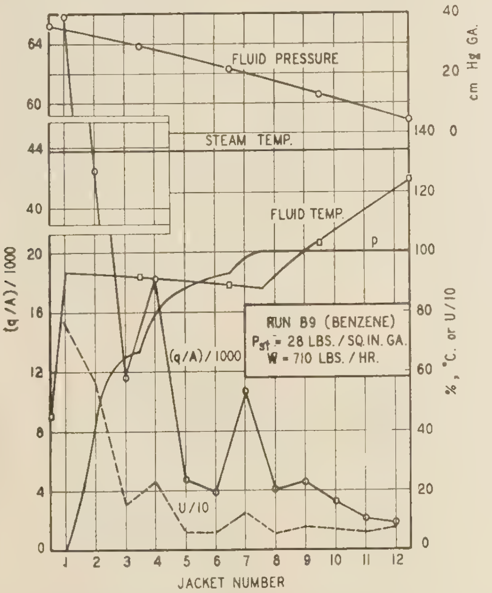

Fig. 3 Data and Calculated Values for Run No. B-9 on Benzene, With Steam at 28 Psi Gage

(Return bends preceding jackets Nos. 4 and 7 result in increased rates of heat transfer in these jackets.)

These two runs are typical. It was generally observed that, when the cumulative per cent vaporization became too high, the heat-transfer coefficients decreased rapidly in magnitude to values approximating those for warming vapor, despite the presence of small amounts of liquid in the stream, Fig. 2. This phenomenon represents a form of "vapor-binding," due to the presence of insufficient liquid to wet the walls of the pipe. It was also generally observed that a jacket immediately following a return bend gave abnormally high coefficients under those circumstances where the tube wall would otherwise be vapor-bound, Fig. 3.

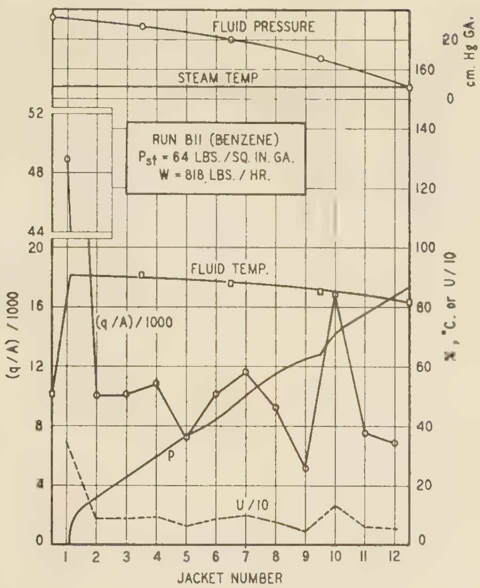

Fig. 4 Data and Calculated Values for Run No. B-11 on Benzene, With Steam at 64 Psi Gage

(Low values of U are caused by vapor-binding due to excessive temperature difference. Capacity of apparatus is less than when using steam at 28 psi gage, Fig. 3.)

This phenomenon apparently results from the whirling of liquid droplets against the tube wall as the fluid passes around the bend. The operation of jackets Nos. 4, 7, and 10 was not abnormal when (*a*) only small cumulative percentages of the feed had been vaporized, or (*b*) when the feed had been completely vaporized. This beneficial effect of return bends indicates that the insertion of twisted metal ribbons (known as "whirlers") inside the tube should serve to minimize vapor-binding as large cumulative percentages of the feed were evaporated.

As shown in Fig. 4, when high steam pressures were used (64 psi gage, or greater) the heat-transfer coefficients were reduced to an average value of less than 100 for the entire boiling section. This is the type of vapor-binding encountered when boiling under natural convection outside of horizontal tubes at high temperature differences, where a vapor film insulates the tube wall from the bulk of the liquid. Vapor-binding when boiling inside of tubes can be due (*a*) to excessive cumulative per cent vaporization, or (*b*) excessive temperature differences.

The operating variables and the over-all results of all of the runs on benzene are summarized in Table 1. The work with benzene was intended primarily to high light the field. In order to check the reproducibility of the data, a large portion of the runs were taken at low steam pressures. There are five runs at high steam pressures and only four runs at intermediate steam pressures (10 to 50 psi gage).

The over-all heat balances, corrected for losses, showed an average deviation of $3^1/_2$ per cent for all of the runs on benzene. The heat balances were less satisfactory during those runs where the total heat transfer was small; a maximum deviation of 19 per cent occurred in run No. B-13 where the total heat transferred was only 60,000 Btu per hr, or an average of only 5700 Btu per hr per sq ft of heated area.

Fig. 5 presents the coefficients for the boiling section for all runs taken at steam pressures of from 1.2 to 2.1 psi gage. At the beginning of boiling, the heat-transfer coefficients are approximately 300, independent of feed rate. However, the maximum heat-transfer coefficient, and the corresponding cumulative per

TABLE 1 RUNS ON BOILING BENZENE[a]

(Steam used in all twelve jackets in all runs)[b]

Run no.	Feed rate, lb per hr	Feed temp, C	Steam pressure, psi gage	Heat[c] transfer, Btu per hr	Inlet velocity, fps	Outlet velocity, fps	Weight per cent of feed vaporized	Start of boiling section, per cent of jacket	Start of superheating section, per cent of jacket	Pressure drop, cm Hg
B-6	1010	54.5	114.0	136000	0.94	175	66	20 of J 2	.....	20.1
B-3	990	52.5	108.0	161000	0.92	220	82	10 of J 2	.....	24.4
B-2	906	45.0	12.7	183000	0.84	240	100	10 of J 2	50 of J 12	37.4
B-4	1080	55.8	1.9	149000	1.00	185	66	90 of J 3	.....	20.8
B-7A	1030	58.5	2.1	146000	0.95	190	73	70 of J 3	.....	...
B-11	818	50.5	64.0	141000	0.76	190	87	60 of J 1	.....	23.0
B-10	689	44.5	34.0	152000	0.64	200	100	50 of J 1	60 of J 9	26.7
B-9	710	45.0	28.0	157000	0.66	210	100	50 of J 1	10 of J 8	32.0
B-8	750	45.7	13.4	157000	0.69	205	100	End of J 1	50 of J 10	33.2
B-1	615	42.5	1.5	125000	0.57	170	100	40 of J 2	50 of J 12	20.2
B-5	650	47.0	1.5	122000	0.60	165	96	95 of J 2	.....	17.4
B-7	700	45.5	1.2	127000	0.65	170	91	10 of J 3	.....	17.4
B-14	713	44.0	1.1	124000	0.66	160	85	60 of J 3	.....	18.0
B-1A	704	47.5	1.6	129000	0.65	165	91	10 of J 3	.....	20.9
B-4A	674	51.5	1.7	127000	0.62	180	100	80 of J 2	90 of J 12	19.6
B-6A	720	48.0	2.1	131000	0.67	175	92	End of J 2	.....	19.1
B-12	448	45.5	74.0	87000	0.42	120	98	70 of J 1	.....	9.3
B-2A	434	41.0	1.6	91000	0.40	120	100	20 of J 2	40 of J 9	11.7
B-5A	453	46.0	1.8	92000	0.42	130	100	30 of J 2	70 of J 9	12.3
B-13	289	37.5	70.0	60000	0.27	80	100	70 of J 1	70 of J 12	4.6
B-3A	282	36.0	1.8	60000	0.26	81	100	80 of J 1	10 of J 6	5.7

[a] Refer to Bibliography (6).
[b] Inside heated area of copper pipe = 0.88 sq ft per jacket.
[c] Based on steam-condensate measurements.

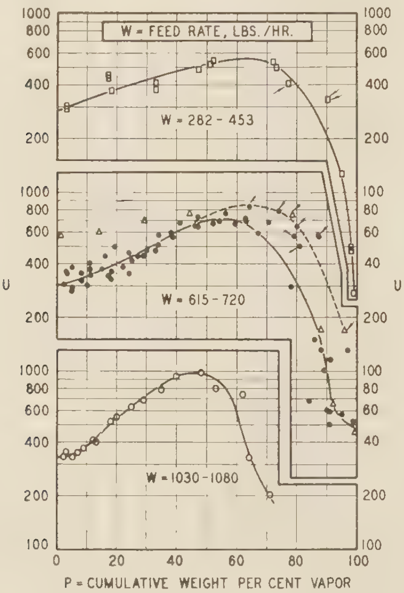

FIG. 5 PLOT OF LOCAL OVER-ALL HEAT-TRANSFER COEFFICIENT VERSUS CUMULATIVE WEIGHT PER CENT VAPOR FOR ALL BENZENE RUNS AT LOW STEAM PRESSURE; 1.2 TO 2.1 PSI GAGE

(Maximum local coefficients are obtained at critical values of p, but size of maximum coefficient and critical p are affected by feed rate.)

cent vaporization, depended upon the feed rate, being 940 and 45 per cent, 700 and 55 per cent, and 540 and 60 per cent when the feed rates were 1050, 680, and 390 lb per hr, respectively. The first run taken in the apparatus (run No. B-1, shown by triangles in Fig. 5) yielded abnormally high coefficients which could not be duplicated in subsequent runs. In Fig. 5, arrows are used to designate data taken from jackets Nos. 4, 7, and 10 (immediately following a return bend) when the cumulative per cent vapor was greater than 60 per cent.

Fig. 6 shows the coefficients of Fig. 5 plotted against the average linear velocity, arbitrarily taken as the geometric mean of the calculated velocities entering and leaving that section, deleting the data shown by triangles and arrows in Fig. 5, and adding data for sections where vaporization started. If the curves of Fig. 5 are compared, it will be found that, in the range of p from 55 to 70 per cent, the coefficients are complicated functions of the feed rate, the highest coefficients being obtained at the intermediate feed rate. This difficulty is overcome in Fig. 6, where, at the lower velocities the coefficients are independent of the feed rate and, at the higher velocities, higher coefficients are consistently

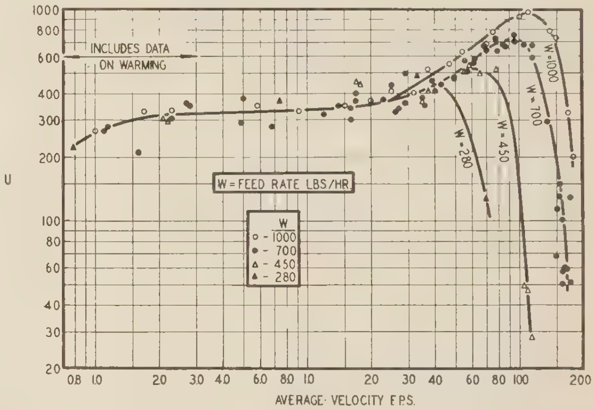

FIG. 6 RESULTS OF FIG. 5 REPLOTTED AS U VERSUS AVERAGE VELOCITY

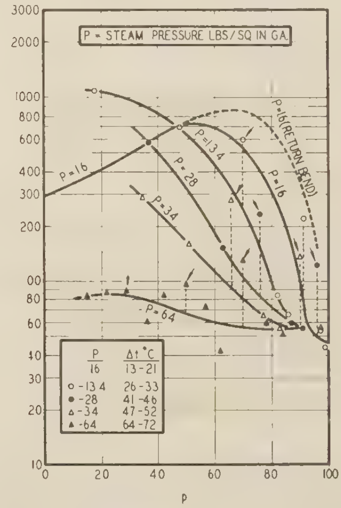

FIG. 7 EFFECT OF STEAM PRESSURE AND PER CENT VAPORIZED FOR BENZENE RUNS WITH FEED RATES RANGING FROM 615 TO 818 LB PER HR; CURVES FOR P OF 1.6 FROM FIG. 5

obtained with the higher feed rates. However, Fig. 6 sheds no light on the point where vapor-binding, due to excessive cumulative vaporization commences, i.e., there does not seem to be a critical velocity, independent of feed rate, which is sufficient to sweep the liquid from the wall of the tube.

The results of all runs taken at feed rates of approximately 700 lb per hr and various steam pressures are presented in Fig. 7. At low cumulative per cent vaporization the highest heat-transfer coefficient was obtained when using a steam pressure of 13.4 psi gage, corresponding to an over-all temperature difference of 26 C. Critical temperature differences of this same magnitude have been obtained (7) when boiling benzene in natural convection evaporators heated by submerged horizontal tubes. At high cumulative per cent vaporization, higher temperature differences (resulting from the use of higher steam pressures) increase the extent of vapor-binding. The beneficial effect of return bends is strikingly illustrated by those points which are designated by arrows in Fig. 7.

At low rates of heat transfer, the precision of measurement becomes poor because the condensate resulting from heat losses from the apparatus becomes a substantial fraction of the total steam condensate. At feed rates of about 700 lb per hr, the heat-transfer coefficients averaged 70 in the superheating section, although the actual coefficients in the superheating region varied from 57 to 107. Predicted coefficients (8) for superheating benzene vapor at this feed rate are about 80.

Heat-transfer coefficients obtained for preheating the benzene ran from 2 to 4 times as large as those predicted (8) for the warming of liquids in turbulent flow inside of pipes. These abnormally high coefficients may be due (*a*) to increased turbulence resulting from temporary vaporization in the superheated film adjacent to the hot wall, followed by condensation in the bulk of the stream of liquid, and (*b*) to the turbulence resulting from the surging flow inside the tubes, superimposed upon that which would normally exist in steady flow.

Results and Discussion of Runs on Boiling Water

The operating variables and the over-all results of all of the runs on water are summarized in Table 2. The first five runs were made using water which had been left in the apparatus 48 hr and was turbid and red with rust. The system was then drained and flushed with distilled water, the inside of the copper pipes was thoroughly swabbed with pieces of cloth wrapped around a wire brush, and two check runs were taken using fresh distilled water. Use of rust-free water and freshly cleaned tubes (run No. W-6) gave the same results as run No. W-2 with rusty water; run No. W-7 with clean water gave somewhat higher coefficients than run No. W-5 with rusty water.

The over-all heat balances, corrected for losses, showed a maximum deviation of 8 per cent and an average deviation of 3 per cent; the heat flux and the coefficients were based on the steam-condensate readings. The local over-all temperature difference for each jacket in the boiling section was taken as the difference between the saturation temperature of the steam and that of the liquid. During the final water runs, the steam supply was cut off from the first three to six jackets; this technique provided a check on the operation of the individual steam jackets by varying the number of heated jackets preceding any one jacket.

Of special interest in Table 2 are the first four runs, all taken at substantially the same feed rate and steam pressure, but with a varying number of heated jackets. Six heated jackets (run No. W-10) transferred 308,000 Btu per hr; increasing the heated area

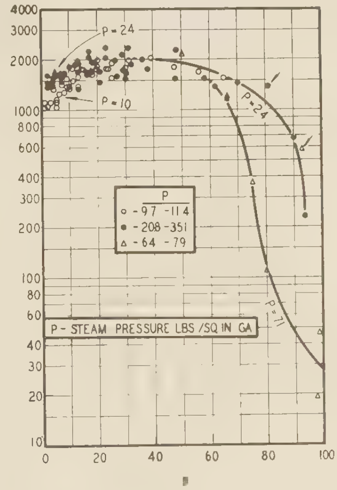

Fig. 8 Plot of Local Heat-Transfer Coefficient Versus Cumulative Weight Per Cent Vaporized for All Runs on Water (Vapor-binding occurs at high values of p due to insufficient liquid to wet walls.)

by 50 per cent (run No. W-9) resulted in the transfer of 376,000 Btu per hr, an increase of only 22 per cent in the heat transfer. Increasing the heated area from six to twelve jackets increased the heat transfer from 308,000 to 425,000 Btu per hr, an increase of only 38 per cent. As can be seen from Table 2, in these four runs, an increase in heated length increased the pressure drop and decreased the temperature difference; the average coefficient for the boiling section changed but little.

During the runs on water the amount of condensate collected

TABLE 2 RUNS ON BOILING WATER[a]

Run no.[b]	Feed rate, lb per hr	Feed temp, C	Number heated jackets	Steam pressure, psi gage	Heat[c] transfer, Btu per hr	Inlet velocity, fps	Outlet velocity, fps	Weight per cent of feed vaporized	Average boiling Δt, C	Average boiling coefficient, Btu per sq ft per deg F per hr	Pressure drop, cm Hg
W-5 (r)	984	88.3	12	23.4	423000	0.75	410	41	14.3	1630	44.3
W-7 (c)	1022	81.5	12	21.7	426000	0.78	400	39	12.6	1940	49.1
W-9 (r)	1110	87.0	9	21.2	376000	0.85	380	32	16.1	1760	31.8
W-10(c)	935	85.9	6	22.1	308000	0.71	310	31	20.5	1700	19.8
W-3 (r)	1037	88.2	12	11.4	227000	0.79	230	20	9.5	1400	20.4
W-13(c)	696	74.4	6	(64)[d]	552000	0.53	540	77	(42)	(1370)	42.8
W-11(c)	672	79.3	6	35.1	465000	0.51	470	67	26.6	1910	33.4
W-6 (c)	685	78.5	12	10.3	245000	0.52	250	33	9.2	1450	17.7
W-2 (r)	690	80.8	12	10.5	244000	0.53	260	33	9.2	1450	18.5
W-8 (r)	674	82.4	9	10.1	200000	0.51	205	27	10.3	1560	12.5
W-12(c)	348	63.1	6	(79)[d]	358000	0.27	400	99	(57)	(620)	22.2
W-4 (r)	425	66.7	12	20.8	417000	0.32	460	95	18.2	1220	32.5
W-1 (r)	372	65.5	12	9.7	273000	0.28	300	69	9.4	1600	16.8

[a] Refer to Bibliography (6).
[b] (r) and (c) refer to rusty and clean feedwater, respectively.
[c] Based on steam-condensate measurements.
[d] During the two runs at steam pressures above 60 psi gage, the operation of the pressure gage was unsatisfactory and the condensing-steam temperature, as estimated from the thermocouple readings, may be in error by as much as 5 C; but this error is only about 10 per cent of the over-all temperature difference in these two runs.

from jacket No. 6 was consistently lower than that in either of the adjacent jackets. Even when steam was not supplied to the first three jackets, jacket No. 6 (which was then the third heated jacket) gave less condensate than jackets Nos. 5 or 7. It was concluded that the conditions for heat transfer in jacket No. 6 were not as favorable as in the adjacent jackets, possibly due (*a*) to a partial blocking of the air vent, or (*b*) to a partly clogged condensate drain line which caused the lower part of the tube to be immersed in condensate. Accordingly, all data from jacket No. 6 have been deleted from the subsequent correlations, except that the actual rate of heat transfer in jacket No. 6 was used to calculate the cumulative vaporization in subsequent jackets.

The local over-all coefficient of heat transfer in the boiling section is plotted in Fig. 8 versus the cumulative weight per cent of the feed vaporized, for all of the runs on water. The feed rate ranges from 58,000 to 185,000 lb per hr per sq ft of cross section, corresponding to inlet velocities of 0.27 to 0.85 fps. Arrows are used to designate data taken on a jacket immediately following a

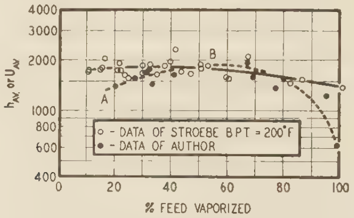

Fig. 9 Plot of Average Heat-Transfer Coefficient in Entire Boiling Section Versus Total Per Cent of Feed Vaporized, and Comparison With Similar Data Taken at University of Michigan (5) in a Vertical Tube

return bend when the cumulative vaporization was in excess of 60 per cent.

In Fig. 8, over a range of cumulative per cent vaporization from 2 to 70 per cent, the coefficients fall in a band of points having a maximum deviation of about 25 per cent from the average coefficient at any given value of p. At vaporizations of less than 10 per cent, the higher steam pressures (about 24 psi gage) give coefficients higher than the lower steam pressures (about 10 psi). This beneficial effect of increasing temperature difference fades out as higher cumulative vaporization is encountered, and the heat-transfer coefficients increase with increase in p and go through a flat maximum at about 40 per cent vaporized. At high values of p the heat-transfer coefficients decrease to well below 1000. Based on meager data in this range, the coefficients obtained when using higher temperature differences (resulting from the use of higher steam pressures) are lower than when using lower temperature differences. This increase in vapor-binding at high cumulative vaporization is similar to that obtained with benzene, Fig. 7.

In confirmation of the results obtained on water are the data of Stroebe, Baker, and Badger (5) who boiled water under various pressures inside a 1.76-in. vertical copper tube, 20 ft long. Mass velocities ranged from 14,400 to 126,000 lb per hr per sq ft of cross section, corresponding to inlet velocities of 0.065 to 0.58 fps. However, the heat-transfer coefficients (based on an average temperature difference as measured with a traveling thermocouple) were correlated in terms of physical properties of the liquid and the temperature difference, independent of feed rate, except as varying the feed rate varied the mean temperature difference and, hence, the average liquid temperature. Film heat-transfer coefficients for boiling water varied from 1160 to 2640.

When Stroebe plotted his heat-transfer coefficients versus the temperature difference, for a given feed rate and discharge pressure, he observed that high coefficients were obtained at the lowest temperature differences, that the heat-transfer coefficient went through a minimum at 8 to 10 F, and that further increases in temperature difference resulted in an increase in the coefficient. The high coefficients at low temperature differences may be specific to low temperature differences, or may possibly result from (*a*) small errors in measuring the temperatures, which would cause a large percentage error in the small temperature difference, or (*b*) the use of a length-mean temperature difference, despite a variation in local heat-transfer coefficient along the tube. For purposes of comparison with the data of this paper, in which temperature differences below 14 F were never employed, the coefficients obtained by Stroebe when boiling water at 200 F have been plotted in Fig. 9, versus the total per cent of the feed vaporized, arbitrarily deleting all coefficients based upon temperature differences of less than 3 F. On the same plot are shown the data of this paper for the boiling section of all runs on water, calculated as average coefficients (cf. Table 2) by dividing the total heat flux in the boiling section by the length-mean temperature difference.

Both sets of data plotted in Fig. 9, indicate a decrease in the heat-transfer coefficient as the discharge end of the tube becomes vapor-bound, due to excessive cumulative vaporization. In fact, Stroebe observed that as the fluid left the tube, under some conditions (particularly high over-all temperature difference), all of the liquid seemed to be carried as a spray up the center of

Fig. 10 High-Speed Photograph of Return Bends at End of First and Third Passes

(Taken between surges at a feed rate of 530 lb of water per hr and a steam pressure of 20 psi gage. Calculated cumulative vaporization equals 8 per cent of feed [by weight], in first return bend [at the left] and 47 per cent in third. Note quiet liquid layer in bottom of first return bend. A cardboard background was placed behind return bends. Photograph taken by Professor H. E. Edgerton with exposure of 1/100,000 sec.)

Fig. 11 High-Speed Photograph Taken a Few Seconds After Photograph Shown in Fig. 10

(Photograph taken during a surge in first return bend by Professor H. E. Edgerton.)

the tube, "suggesting that the film became thinner near the top and. . .disappeared entirely, leaving part of the tube surface dry." It is important to note that, although the last part of the tube surface may be badly vapor-bound, the average heat-transfer coefficient for the entire boiling section may still be quite high.

Stroebe's coefficients at low vaporization run higher than those obtained in the horizontal tubes. While this may in part be due to the comparison of film versus over-all coefficients, it is more probably due to the difference in vertical and horizontal tubes. When only small percentages of the feed had been vaporized in the horizontal-tube apparatus, observation of the glass return bends showed that the liquid phase was carried as a separate

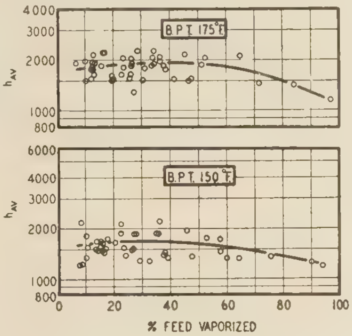

Fig. 12 Data Taken at the University of Michigan (5) When Boiling Water at Fluid Discharge Pressure Other Than Atmospheric

(Vapor-binding at high cumulative vaporization causes decrease in average coefficient at high total vaporization.)

layer, moving at low velocity and filling only a fraction of the horizontal tube. Occasional surges resulted in an intimate mixture of liquid and vapor which completely filled the tube. This intimate mixture prevailed steadily at higher vaporization, eventually giving way to a spray of liquid drops suspended in a stream of vapor. This phenomenon is illustrated in Figs. 10 and 11. In the range *AB* of Fig. 9, it is quite possible that the beneficial effect of increasing the per cent vaporized in horizontal tubes is largely due to an increase in the fraction of the wall wetted, as an intimate mixture of liquid and vapor replaces the liquid layer which in earlier jackets filled only a fraction of the tube.

The data of Stroebe (5) at pressures other than atmospheric, Fig. 12, also suggest the existence of vapor-binding at high cumulative vaporization, for no coefficient greater than 1500 was obtained when the total vaporization in the tube was greater than 70 per cent.

Tube diameter apparently is not important when boiling liquids outside of horizontal pipes (9), but it may be very critical when boiling liquids inside of horizontal pipes because of its possible effect on the character of flow. Thus, with a low entering velocity, in a 4-in-diam horizontal pipe, stratification of liquid and vapor might persist to far greater cumulative vaporization, whereas, in $^1/_2$-in-diam horizontal pipe, appreciable stratification might never occur.

Conclusions

For pure liquids entering at low velocities and boiling inside horizontal tubes the following conclusions are drawn:

1 Vapor-binding, accompanied by a decrease in the local over-all coefficient of heat transfer, can be caused by excessive cumulative per cent vaporization when using moderate temperature difference (Figs. 2, 5, and 8) or by excessive temperature difference at moderate per cent vaporization, Fig. 4.

2 With moderate temperature difference, as the cumulative per cent vaporization increases, the coefficient at first rises, then goes through a maximum and decreases sharply; in the latter range the effect of a return bend is beneficial in increasing the coefficient in the jacket immediately following.

3 With excessive temperature difference, equal to or somewhat greater than that required to produce the maximum coefficient, the surface becomes vapor-bound, and an increase in cumulative weight per cent vaporized is not beneficial and may be detrimental, Fig. 7.

4 In the boiling section, the fluid stream, immediately following the return bend, is at substantially the saturation temperature corresponding to the pressure.

5 In the superheating section, the coefficients are of the same order as those predicted from the accepted equation for heating gases.

6 In the preheating section the coefficients are considerably higher than those predicted for warming liquids; this may be due to local boiling followed by condensation in the bulk of the stream of liquid, and to the surging flow inside the tube.

Acknowledgment

The authors acknowledge their indebtedness to Prof. H. E. Edgerton for use of the photographs shown in Figs. 10 and 11, and to Mr. L. C. Heroman for his participation in runs B-1A to B-7A.

BIBLIOGRAPHY

1 "Temperature Drops and Liquid-Film Heat-Transfer Coefficients in Vertical Tube," by R. M. Boarts, W. L. Badger, and S. J. Meisenburg, Trans. American Institute of Chemical Engineers, vol. 33, 1937, pp. 363–389; same title, *Industrial and Engineering Chemistry*, vol. 29, 1937, pp. 912–918.

2 "Heat-Transfer Coefficients in the Boiling Section of a Long-Tube Natural-Circulation Evaporator," by C. H. Brooks and W. L. Badger, Trans. American Institute of Chemical Engineers, vol. 33, 1937, pp. 392–413; same title, *Industrial and Engineering Chemistry*, vol. 29, 1937, pp. 918–923.

3 "Liquid Velocity and Coefficients of Heat Transfer in a Natural-Circulation Evaporator," by A. S. Foust, E. M. Baker, and W. L. Badger, Trans. American Institute of Chemical Engineers, vol. 35, 1939, pp. 45–71; same title, *Industrial and Engineering Chemistry*, vol. 31, 1939, pp. 206–214.

4 "Film-Heat-Transfer Coefficients for SO_2 in a Vertical Evaporator," by F. C. Stewart and F. G. Hechler, *Refrigeration Engineering*, vol. 31, Feb., 1936, pp. 107–111; same title, *Ice and Cold Storage*, vol. 39, 1936, pp. 126–127; same title, *Ice and Refrigeration*, vol. 90, Jan., 1936, p. 3.

5 "Boiling-Film Heat-Transfer Coefficients in a Long-Tube Vertical Evaporator," by G. W. Stroebe, E. M. Baker, and W. L. Badger, Trans. American Institute of Chemical Engineers, vol. 35, 1939, pp. 17–41; same title, *Industrial and Engineering Chemistry*, vol. 31, 1939, pp. 200–206.

6 "Heat Transfer for Boiling Inside Horizontal Tubes," by W. K. Woods, D.Sc. thesis, chemical engineering, Massachusetts Institute of Technology, Cambridge, Mass., 1940.

7 "Boiling: Heat Transfer in Natural Convection Evaporator," by G. A. Akin and W. H. McAdams, Trans. American Institute of Chemical Engineers, vol. 35, 1939, pp. 137–155; same title, *Industrial and Engineering Chemistry*, vol. 31, 1939, pp. 487–491; "Heat Transfer to Boiling Liquids," by E. T. Sauer, H. B. H. Cooper, G. A. Akin, and W. H. McAdams, *Mechanical Engineering*, vol. 60, 1938, pp. 669–675.

8 "Principles of Chemical Engineering," by W. H. Walker, W. K. Lewis, W. H. McAdams, and E. R. Gilliland, third edition, McGraw-Hill Book Company, Inc., New York, N. Y., 1937, pp. 111–113.

9 "Heat Transfer to Boiling Liquids Outside of Horizontal Tubes," by G. A. Akin, D.Sc. thesis, chemical engineering, Massachusetts Institute of Technology, Cambridge, Mass., 1940.

Discussion

Victor J. Skoglund.[7] The authors analyze pressure-drop data under the assumption that the pressure drop has only two

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components: (*a*) wall friction, and (*b*) change in kinetic energy. In the flow of media of more than one phase of different densities, the different phases will have different velocities, and therefore, there will be a transfer of momentum between phases. A transfer of momentum between particles, having different velocities, results in a loss of mechanical energy. This loss of energy adds another component to the pressure drop.

The equation of motion of a liquid particle evaporating in a pipe is very complex. The equation should include the following terms:

1 A force term due to the pressure gradient.

2 A force term due to the relative velocity between liquid and vapor phases. This is the component discussed by the writer.

3 A momentum term due to the changing velocity of the particle.

4 A momentum term due to the changing mass of the particle.

Authors' Closure

The analysis of the pressure drops is to appear in a paper to be presented at a subsequent meeting of the Society. The friction losses were calculated from the observed pressure drops in each of the last three passes, allowing for the changes in kinetic energy due to changes in the mass and velocity of both vapor and liquid. It was assumed that the friction loss could be correlated by an equation of the Fanning type, without allowance for the unknown slip between the vapor and liquid.

In the range where the cumulative vapor generation exceeded 20 per cent by weight, the friction factors so obtained were intermediate between the usual friction factors for one-phase isothermal flow, corresponding to Reynolds' numbers for all liquid, and all vapor, respectively. The reasonable values of these apparent friction factors suggest that friction, arising from the transfer of momentum between vapor and liquid phases, is of minor importance under the conditions of the experiments described.

In the range where the percentage of feed vaporized was small, and where separation by gravity into two continuous phases sometimes occurred (as shown in Fig. 10 of the paper, at the end of the first pass), the pressure drops were too small to warrant any conclusions.

Recommended Code of Procedure for Fatigue Testing of Hot-Wound Helical Compression Springs

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This paper gives the recommended procedure for fatigue testing hot-wound helical compression springs in complete detail, so that any competent research technician, with only an elementary knowledge of spring theory, can successfully carry out a program of fatigue tests on helical springs, with practically no other guidance than the instructions laid down in the Code. This has been proved by actual trial.

The test procedure is planned to develop a complete *S-N* diagram for each group of variables, instead of merely an endurance limit. In the actual plotting of the *S-N* diagrams, the Committee has developed a technique based on probability theory, which has been included in the Code recommendations. The type of formula used is assumed. The precision measurements and test methods prescribed in the Code permit the computation of relative stresses in the test springs to a high order of accuracy.

It would seem reasonable to assume that there is a mathematical equation representing the *S-N* relation, the formulation of which awaits an exact determination of the statistical laws controlling the formation and propagation of fatigue cracks. The Committee feels that a studied guess at the correct form of equation is a constructive move. If results indicate that the guess is approximately correct, the use of such a formula has the important advantage that without it the research consists of a series of successive approximations to the correct value of the endurance limit, a quantity obviously immune to exact measurement. In testing helical springs, this procedure is very tedious and expensive, on account of the slow speeds to which we are limited. On the other hand, if the test data are fitted to a probable *S-N* diagram, every observation makes a considerable contribution to the correctness of the graph, and only one or two of the runs need be prolonged beyond 1,000,000 cycles of stress.

This code has been prepared for the guidance of the Special Research Committee on Mechanical Springs in the conduct of fatigue tests on heavy helical springs. It is the result of experience gained from a considerable number of such tests which have been conducted in recent years, under the sponsorship of the Committee.

Purpose of Tests

As the test procedure laid down in the code is quite definitely standardized, it cannot be considered as applicable to every variety of investigation into helical-spring endurance. In general, it is intended to cover tests on such variables as grade of material, heat-treatment, amount of cold-working, effect of surface finish and (to some extent) variations in the absolute and relative dimensions of the springs.

Size of Test Springs

Except where the variable investigated is a dimensional one, the recommended size of the test springs is as follows:

	Dimensions	Design *A*	Design *B*
D =	outside diameter	6 in.	$4^1/_2$ in.
d =	size of bar	1-in. rd	$^3/_4$-in. rd
n =	number of coils	$7^1/_2$	$7^1/_2$
p_i =	pitch as coiled (max)	$2^1/_2$ in.	$1^7/_8$ in.
	pitch as coiled (min)	$1^1/_2$ in.	$1^1/_8$ in.

Design *A* is preferred, provided the fatigue machine available is of sufficient size and strength to handle it; otherwise use design *B*.

It is recommended that the endurance tests be planned to develop a complete stress-endurance curve, rather than merely a value for the endurance limit. Therefore, the free height of the test springs should be varied so that a suitable range of test stresses can be applied. The simplest practical method to obtain this range of stress in the test springs is to specify the desired pitch to which the individual springs in the test set should be wound. The maximum and minimum values for pitch previously given are satisfactory for most grades of material.

Selection of Material

The standard test set consists of fourteen bars 1 in. rd by 111 in. long, for design *A*; $^3/_4$ in. rd by $83^1/_2$ in. long, for design *B*. The chemical composition should approximate the mean values for the grade. History of the heat should be known, including character of melt, pouring temperatures, and details of processing.

The recommended metallurgical tests are described in a later section; but certain of these tests should be conducted at the time of selection of the heat, to insure representative material. They are:

1. Check analysis
2. Surface inspection
3. Examination for decarburization
4. Penetration-fracture determinations
5. Microexamination
6. McQuaid-Ehn test
7. Magnaflux tests on finished springs

Two of the bars should be held for tension tests and metallurgical examination, and as stock for pilot pieces in control of heat-treatment; leaving twelve bars to be made up into springs.

Tapering and Coiling

Spring bars should be roll-tapered on both ends: Design *A* to

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a length of $117\frac{1}{4}$ in., with points $\frac{17}{64}$ in. thick; design *B* to a length of 88 in., with points $\frac{13}{64}$ in. thick. These taper dimensions are estimated to produce the standard tapered bearings of three quarters of a coil, after grinding 1/64 in. from each end of the spring.

For coiling, the springs should be heated rapidly and uniformly to a temperature of 1800 to 1900 F, and not allowed to soak. All practicable precautions should be taken against decarburization. Coiling should preferably be done on a preheated mandrel. Springs should be slow-cooled after coiling.

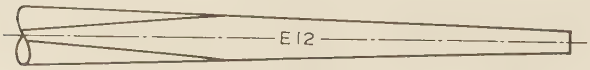

Fig. 1 Group Letter and Serial Number Stamped on Flat Side of Tapered-Spring End

It is suggested that each group of springs be assigned an identifying letter, and that the individual springs in each group be serially numbered from 1 up. The group letter and serial number should be cold-stamped on the flat side of each tapered end, Fig. 1.

Heat-Treatment

Springs should be heated in a controlled-atmosphere furnace to the recommended temperature for the grade, allowing about 60 min for design *A*, and 45 min for design *B*, total time in the furnace. They should be withdrawn from the furnace on a rising-temperature gradient, and immersed as rapidly as possible in a bath of high-grade quenching oil (Houghton Soluble, Atlantic Matchless, Rodmanol, or equivalent). If the bath is not mechanically agitated, springs should be kept in motion therein for at least 30 sec after immersion. The bath should be warm enough to be entirely fluid, but at no time should exceed 125 F.

Springs should then be drawn to the approved drawing temperature for the grade, preferably in a liquid bath or, as second choice, in an electrically heated furnace. With the liquid bath, the draw time (at temperature) should be 60 min for design *A*, or 45 min for design *B*; for any other medium, somewhat more time is recommended. After withdrawal from the draw medium, the springs should be immersed in cool oil, until they are no more than "hand-warm."

To check the results of the heat-treatment, one of the two spare bars should be cut into pieces about 5 or 6 in. long. These should be wired to the springs, and accompany them through the treatment. Brinell hardness tests on these pilot pieces should then give readings of 400 to 430, for most standard spring grades. Bending and fracture tests are also recommended on some of the pilot pieces, if facilities are available.

Preliminary Grinding

Both ends of each spring should then be ground in a chuck to give the springs an absolutely stable bearing, and to make them stand not more than $\frac{1}{8}$ in. per ft out of perpendicular. No more than the minimum amount of grinding required to accomplish these results is recommended. Further grinding, after the solid tests, will usually be necessary.

Preliminary Tests

Each spring should be compressed solid a sufficient number of times to put it into an approximately cyclic condition. For springs of high initial pitch, a large number of compressions is required to remove practically all "permanent set." It is suggested that the solid compressions be applied in sequences of five, and the free height measured after each sequence. When two successive measurements differ by less than $\frac{1}{32}$ in. the test can be considered complete. The number of tests required will be 5 to 10 for low-pitch springs and up to 25 or 30 for the higher pitches.

Final Grinding

Each spring should then be reground on both ends so that it seats firmly on a level surface, and stands not more than $\frac{3}{64}$ in. out of perpendicular.

Calibration Tests

Each spring should then be given a load-deflection test in an accurate testing machine. Following are the steps recommended in detail:

Apparatus. The apparatus used consists of the following:

1 Testing machine, preferably of the screw type. It should be of approximately 10 to 25 tons capacity for design *A*, and 5 to 15 tons for design *B*.

2 A depth-gage micrometer, L. S. Starrett catalog, No. 124, or equivalent, for taking the height readings.

3 A deep-jaw micrometer of proper size for measuring outside diameters.

4 A round-nose or needle-point micrometer: and

5 A vernier caliper, both for measuring bar diameters.

6 Two round or square bearing plates, about 3 in. larger than the outside diameter of the spring, not less than 1 in. thick, machined, hardened, and ground true flat and with parallel faces.

Procedure. Prick punch one bearing plate at its approximate center and, from this center, scribe several circles, the smallest of these being about the same diameter as the spring. Scribe a diametral line through the center; the two intersections of this line with the outer circle are the datum points for the height measurements. The outer circle should be of about the minimum diameter that will permit ready manipulation of the micrometer. The other circles will aid in visual centering of the spring on the bearing plate.

Mount the bearing plate, with the spring centered thereon, in the approximate center of the testing-machine bed, with the datum points oriented for maximum convenience in taking height measurements. Place the other bearing plate on top of the spring. Compress the spring solid at least twice, with a scale load about 25 to 50 per cent in excess of that necessary to effect visual contact between adjacent coils of the spring. Measure the free height of the spring, between bearing plates, at the two datum points, designating the two measurements as "right" and "left," respectively.

Apply successive loads in increments of 1000 lb for design *A*, or 500 lb for design *B*, measuring and recording the heights under each load, at the two datum points. The final load increment should be at about $\frac{1}{4}$ in. to $\frac{1}{2}$ in. above solid height. Compress the spring solid with loads as indicated, and measure the solid height. Release the spring with successive decrements of load, using the same loads as in compression, and measure the heights at each load. Remeasure the free height after the spring is fully released.

Care should be exercised not to overrun (or underrun) the desired load. If an overrun or underrun is accidently made, measure and record the actual load and corresponding heights. Do not attempt to run back to the intended load reading, as this will inevitably cause an inaccuracy in the height readings, due to the hysteresis effect.

Remove the spring from the machine, and measure the outside diameter over a semicoil, in six positions, with the deep-jaw micrometer. The spring should be rotated slightly between successive measurements, so that these are in different diametral planes. These measurements must be corrected to perpendicularity with the axis of the spring, as described later.

Measure the bar diameter in six places. Three of these should be perpendicular to the spring axis (one at each end and one about the center of the spring). These measurements can conveniently be made with the round-nose micrometer. For low-pitch springs, it is sometimes necessary to omit the measurement at the center, owing to the impossibility of inserting the micrometer between the coils. The other three measurements should be parallel to the spring axis; again, one near each end and one at the center. For these the vernier caliper is most convenient.

By way of illustration, a specimen log of one of the Committee tests is shown in Table 1. The spring is of design B. It is recommended that the averages of the two readings under each load be tabulated as the test progresses, since by this means any serious error in the test readings may instantly be detected. A well-made helical spring will show rather close to "straight-line" compression. Usually, with a high-pitch spring, the increments of deflection tend to increase slightly; with a low pitch they decrease somewhat.

TABLE 1 CALIBRATION TEST OF SPRING G-5

Loads	Compression Test: Right, in.	Compression Test: Left, in.	Compression Test: Average, in.	Release Test: Right, in.	Release Test: Left, in.	Release Test: Average, in.
Free after twice solid, lb	8.158	8.169	8.164	8.145	8.162	8.154
500	7.795	7.799	7.797	7.740	7.743	7.742
1000	7.441	7.443	7.442	7.351	7.354	7.353
1500	7.107	7.107	7.107	6.984	6.986	6.985
2000	6.764	6.767	6.766	6.632	6.636	6.634
2500	6.414	6.415	6.415	6.290	6.292	6.291
3000	6.152	6.153	6.153	5.947	5.951	5.949
3500	5.693	5.692	5.692	5.617	5.616	5.617
4000	5.334	5.336	5.335	5.312	5.314	5.313
Solid	5.213	5.213	5.213	...	...	...

Outside-diam measurements: 4.561, 4.573, 4.555, 4.555, 4.563, 4.550—average = 4.560

Size-of-bar measurements: 0.741, 0.737, 0.736, 0.748, 0.748, 0.743—average = 0.742

CALCULATION OF CONSTANTS

d	= size of bar	0.742 in.
D	= outside diameter of half coil	4.560 in.
p	= approximate pitch = $\frac{8.154 - 1.113}{5\ 5}$	1.280 in.
M	= mean helix diameter = $\sqrt{(3.818)^2 - (0.640)^2}$	3.764 in.
S_c	= fiber stress per 1000-lb load (conventional formula) = $\frac{1000 \times 3.764}{0.3927 \times (0.742)^3}$	23464 psi
c	= spring index = $\frac{3.746}{0.742}$	5.0728
k	= Rover-Wahl correction factor	1.3053
S_w	= fiber stress per 1000-lb load (with Rover-Wahl correction) = kS_c	30628 psi
H_s	= solid height	5.213 in.
P_s	= capacity (read from graph)	4165 lb
	maximum fiber stress solid (conventional formula) = $P_s \times S_c$	97730 psi
	maximum fiber stress solid (with Rover-Wahl correction) $P_s \times S_w$	127565 psi

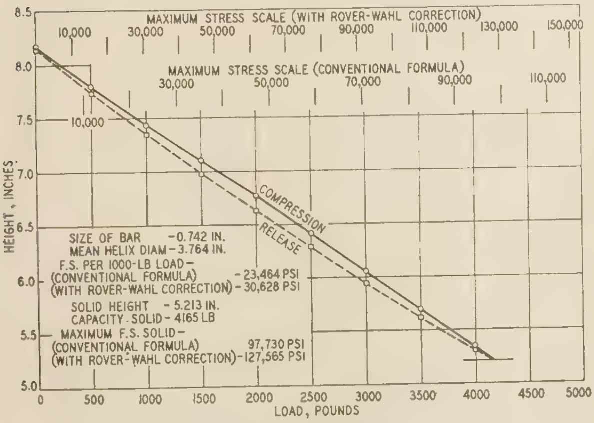

FIG. 2 LOAD-DEFLECTION GRAPH, SPRING G-5

The next step is to compute the stress scale of each spring, so that the load corresponding to the maximum desired stress in the fatigue test can be determined. The steps of the calculation are

d = size of bar (average of previous six measurements)

D = outside diameter of one-half coil (average of measurements)

p = approximate pitch = $\frac{(\text{free height} - 1^1/_2) \times d}{\text{total coils} - 2}$ or the approximate pitch can be scaled from the spring

M = mean helix diameter = $\sqrt{(D-d)^2 - (p/2)^2}$

S_c = fiber stress per 1000-lb load (conventional formula) $= \frac{1000 \times M}{0.3927 d^3}$

c = spring index = M/d

k = Rover-Wahl correction factor $= \frac{4c-1}{4c-4} + \frac{0.615}{c}$

S_w = fiber stress per 1000-lb load (with Rover-Wahl correction) = kS_c

It is recommended that a graphical presentation of the calibration test be made. The standard form used by the Committee is illustrated in Fig. 2. The test figures plotted are those given in Table 1.

Assignment of Stresses for Fatigue Test

In the Committee's experience, it has not been found necessary to test as many as twelve springs in order to establish a satisfactory stress-endurance curve. It has been the practice to withhold several springs from the fatigue test, and to use them for precision tests to establish and check design formulas and constants. Therefore, it is recommended that three springs (No. 1, the highest, No. 6 or 7, and No. 12) be set aside for the purposes referred to. If it is found that the nine remaining springs are insufficient for the purpose of the fatigue tests, they can be supplemented by one or more of the three extra springs.

Obviously no exact instructions can be given for assignment of test stress to each of the nine springs, as it is not known in advance how great a total range the stresses may have to cover. For plain carbon steels in which the maximum fiber stress solid in the highest spring is about 130,000 to 140,000 (with the Rover-Wahl correction), the first assignments may be 108,000 psi to No. 2, 100,000 to No. 3, and 92,000 to No. 5.

Further assignments will depend upon the endurance shown by these three springs. As an illustration, Table 2 gives the exact procedure used in assigning stresses for the Committee's group G springs.

TABLE 2 ASSIGNMENT OF STRESSES FOR GROUP G SPRINGS

Stresses assigned, psi		Cycles to failure	
G-2	108000		
G-5	92000		
G-7	84000		
		G-2	141900
		G-5	338700
		G-7	428000
G-9	80000		
G-10	76000		
		G-9	568600
		G-10	999300
G-11	78000		
G-13	74000		
		G-11	1334200
		G-13	2137500 (unbroken)
G-3	120000		
G-6	100000		
		G-3	95200
		G-6	169100

In the assignment of test stresses, it will be helpful to bear in mind that the stress at which the springs will run 1,000,000 cycles is usually 3000 psi to 5000 psi above the endurance limit.

It is highly important that the stresses, assigned to the various springs, bear a general relationship to the maximum fiber stress

solid. While the relationship need not and cannot be mathematically constant, the higher test stresses should not be assigned to the springs of lower solid stress.

The higher solid stresses are produced by removal of permanent set, i.e., by cold-working the springs; and this cold-working has a marked effect on the endurance. The ideal condition for the tests would be a group of springs with graded solid stresses, produced without cold-working; but this is obviously impossible. The logical alternative is to maintain a relation between the test stress and the solid stress, and so between the test stress and the cold-working effect.

The individual test stresses having been assigned, the test loads are calculated therefrom, by the formula

$$P = \frac{0.3927kd^3S_w}{M}$$

Each spring is then compressed under its test load in a testing machine, and the height accurately measured with the micrometer height gage. The springs are then ready for the fatigue test.

Conduct of Fatigue Test

The machine used for the fatigue test should be of sturdy and rigid construction, so that the cyclic deflection impressed on the springs will be accurate and uniform. Speed should not be so high, nor the stroke so great, as to cause impact stresses. Geared presses of the type suitable for this work usually run about 40 to 70 strokes per min, and the higher speed is not excessive for the purpose. A stroke of not more than twice the spring deflection is believed to be satisfactory. If possible, the machine should be equipped with a screw adjustment in the spring support, so that the rig can be accurately adjusted to give the desired deflection to the spring.

To set up a spring for the test, the machine is turned over until the ram is at the lowest point of its stroke, and the screw adjustment manipulated until the distance between ram and spring support corresponds to the "height under test load," described in a previous section. The ram is then raised, the spring mounted on the support, which should be shouldered to position the spring centrally under the ram, the machine again turned over to its lower center, and the compressed height of the spring checked.

The revolution counter is then set to zero or the initial reading recorded, and the test run started. The run on each spring should be as nearly continuous as is practicable. The machine on which recent Committee tests have been conducted is equipped with a photocell, which shuts down the machine when the spring breaks, so that practically no attention is required during the test run.

Almost any helical spring, in the course of a fatigue test, will suffer a slight change in rate, particularly during the first few thousand cycles of stress. It is therefore recommended that the spring be removed from the fatigue machine and the height under test load remeasured, say, after 5000 cycles, and again after 10,000 cycles, and finally after 25,000 cycles. At each of these stages, the machine should be reset to the new height reading; which, on the first recheck and usually on the second as well, will be a few thousandths under the previous figure.

The test is then continued until the spring fails, or has run 1,500,000 to 2,000,000 cycles. Usually no good purpose is served by any longer run than this; and with proper assignment of stresses, no more than one or two springs in the group need be run more than 1,000,000 cycles.

Analysis and Report of Tests

The primary object of the test is to establish a complete "stress-endurance curve" for the material represented. Such a curve, in addition to its scientific interest, is of considerable practical value to the spring industry. In many (perhaps most) spring applications, the design stress exceeds the endurance limit. This is often permissible because the failure of the spring does not constitute a catastrophe; therefore the consumer is interested in longest spring life per dollar of cost, rather than unlimited life regardless of cost. Therefore, the comparative life at stresses above the endurance limit becomes very important.

Hence, the first step in analyzing the test results is to plot them graphically. It is customary to use a semilogarithmic cross-section chart for this purpose, plotting stresses as ordinates (Cartesian scale), and cycles to failure as abscissas (logarithmic scale). If the tests have been carefully planned and carried out, the results should give an indication of a smooth curve, which approaches a horizontal (stress) asymptote. The latter can

TABLE 3 CALCULATION FOR PROBABLE S-N CURVE; GROUP E

Spring No.	$N/10^3$	$S/10^3$	NS	N^2	S^2	N^2S	NS^2
E-2	95.2	108	10281.60	906304	11664	97880832	1110412.80
E-3	107.1	100	10710.00	11470.41	10000	1147041.00	1071000.00
E-5	197.2	92	1814240	38887.84	8464	3577681.28	1669100.80
E-7	281.6	88	24780.80	79298.56	7744	6978273.28	2180710.40
E-6	232.6	84	19538.40	54102.76	7056	4544631.84	1641225.60
E-11	367.0	80	29360.00	134689.00	6400	10775120.00	2348800.00
E-10	851.0	78	66378.00	724201.00	6084	56487678.00	5177484.00
E-9	Unbroken	(76)					
Totals	2131.7	630	179191.20	1051712.61	57412	84489233.72	15198733.60

$A = \Sigma N \Sigma S - n\Sigma NS = 1342971 - 1254338.40 = +\ 88632.60$

$B = \Sigma N \Sigma NS - n\Sigma N^2S = 381981881 - 591424636 = -\ 209442755$

$C = \Sigma S \Sigma NS - n\Sigma nS^2 = 112890456 - 106391135 = +\ 6499321$

$D = (\Sigma N)^2 - n\Sigma N^2 = 4544145 - 7361988 = -\ 2817843$

$E = (\Sigma S)^2 - n\Sigma S^2 = 396900 - 401884 = -\ 4984$

$$a = \frac{AB - CD}{A^2 - DE} = \frac{-185634359 \times 10^5 + 183140662 \times 10^5}{78557378 \times 10^2 - 140441295 \times 10^2} \quad \frac{-2493897 \times 10^5}{-61883917 \times 10^2} = +\ 40.30$$

$$b = \frac{C - aE}{A} = \frac{+\ 6499321 + 200855}{88632.60} = \frac{+\ 6700176}{+\ 88632.60} = +75.60$$

$$K = \frac{\Sigma NS - b\Sigma N - a\Sigma S + nab}{n} = \frac{179191.20 - 161156.52 - 25389.00 + 21326.76}{7}$$

$$= 1996.06$$

Probable Equation ~ $(N - 40.30)(S - 75.60) = 1996.06$

Points on Curve	$N/10^3$	$S/10^3$	$N/10^3$	$S/10^3$	$N/10^3$	$S/10^3$
	40.30	∞	120	100.64	1000	77.68
	70	142.81	150	93.80	2000	76.62
	80	125.88	200	88.10	5000	76.00
	90	115.76	300	83.29	10000	75.80
	100	109.03	500	79.94	∞	75.60

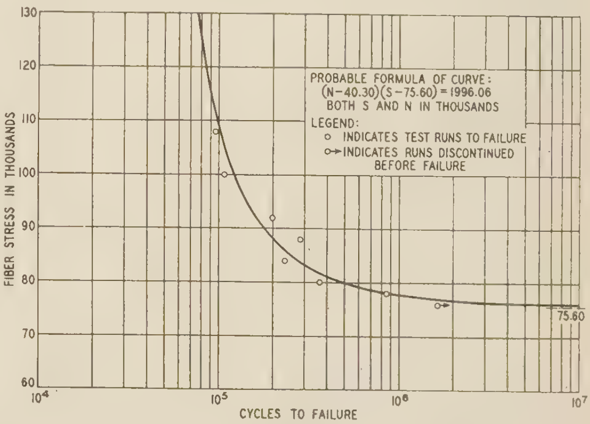

Fig. 3 Fatigue Tests at Wright Field of Hot-Wound Helical Compression Spring; Series E, Plain-Carbon Electric

logically be interpreted as the endurance limit of the material tested.

There is considerable evidence for the theory that the general stress-endurance curve for ferrous metals can be represented by an equation of the form

$$(S - b)(N - a) = k$$

where, S represents stress values and N values for number of cycles of stress before failure. Symbols b and a are, respectively, horizontal and vertical asymptotes to the curve.

A study made by the Committee has developed equations for the probable values of a and b, in terms of the summed values of the S and N observations and functions thereof. The equations are conveniently stated in terms of the following intermediate values:

$$\begin{aligned}
A &= \Sigma N \Sigma S - n\Sigma NS \\
B &= \Sigma N \Sigma NS - n\Sigma N^2 S \\
C &= \Sigma S \Sigma NS - n\Sigma NS^2 \\
D &= (\Sigma N)^2 - n\Sigma N^2 \\
E &= (\Sigma S)^2 - n\Sigma S^2
\end{aligned}$$

n being the number of pairs of observed values used in the summations. The final equations are

$$a = \frac{AB - CD}{A^2 - DE} \qquad b = \frac{C - aE}{A}$$

$$k = \frac{NS - a\Sigma S - b\Sigma N + nab}{n}$$

The derivation of these equations is given in full in the Appendix of this paper. As an illustration of the method, the analysis for one group of springs (group E) tested by the Committee is shown in Table 3. Fig. 3 shows the graphical plot for the same group, and the probable stress-endurance curve.

Special Tests on Springs Withheld From Fatigue Tests

The details of tests made on these springs vary from time to time as particular problems of spring design are brought before the Committee and selected for examination. Typical of such problems are:

Number of inactive coils in a helical compression spring.

Eccentricity of effective loading in a helical compression spring.

Obviously no definite recommendations can be made for the conduct of such special tests.

Supplementary Physical and Metallurgical Tests

It will usually be desirable to make a fairly complete metallurgical examination of the material, together with the usual physical tests. The following list should cover the ground quite thoroughly, and can sometimes be abridged more or less, depending upon the importance of the material under test.

Already mentioned as preferably to be collected in advance of the fatigue tests are the following:

(*a*) Heat history, covering melting and processing practice.

(*b*) Chemical composition, including determinations on carbon, manganese, phosphorus, sulphur, silicon, nickel, chromium, and any other alloys which are known or believed to be present.

(*c*) Inspection of bars for surface defects.

(*d*) File test for decarburization on finished springs.

(*e*) Microexamination on samples from control coupons, at 750× magnification, both transversely and longitudinally, for decarburization, nonmetallic inclusions, etc. Penetration-fracture tests; McQuaid-Ehn tests.

(*f*) Magnaflux tests on the finished springs.

(*g*) Hardness tests on control coupons.

The following additional tests are made on suitable specimens cut from control coupons which have accompanied the springs through the heat-treatment:

(*h*) Tension test for the usual physical characteristics, to be conducted in substantial accordance with A.S.T.M. Specifications.

(*i*) Torsion test for yield point, proportional limit and modulus of rigidity.

(*j*) Impact test on both notched and unnotched specimens.

(*k*) Rotating-beam endurance tests.

It must be admitted that much of the data resulting from the foregoing tests cannot as yet be correlated with current results of the fatigue tests on the springs. It would seem that some correlation must eventually be found. If not, it would mean that we must go farther afield, in search of some physical tests or other indexes which bear a definite relation to the spring endurance.

Back of the professed object of the endurance test lies the ultimate goal of defining the qualities in a spring steel which make for high endurance values in the springs. Bearing this in mind, the function of these miscellaneous tests, as a necessary part of the program, is apparent.

Appendix (1)

Development of equations for the most probable values of the constants a, b, and K, in the equation

$$(x-a)(y-b) = K$$

given a series of pairs of observed values of x and y, subject to random observational errors.

It will be assumed that the most probable values are such as to satisfy the equation

$$\Sigma(K'-K)^2 = \text{minimum}$$

where values of K' are obtained by substituting the successive pairs of observed values (x', y') in the equation

$$(x'-a)(y'-b) = K'$$

Performing the substitution, and dropping the prime marks for convenience, we get

$$\Sigma\left[(x-a)(y-b) - K\right]^2 = \text{minimum}$$

and by differential calculus

$$\frac{d}{da,\, db,\, dK}\,\Sigma\left[(x-a)(y-b)-K\right]^2 = 0$$

or

$$\frac{d}{da,\, db,\, dK}\,\Sigma(xy - bx - ay + ab - K)^2 = 0 \qquad (1)$$

Differentiating (1) with respect to a, b, and K, in turn

$$\Sigma(xy - bx - ay + ab - K)(y-b) = 0 \qquad (2)$$

$$\Sigma(xy - bx - ay + ab - K)(x-a) = 0 \qquad (3)$$

$$\Sigma(xy - bx - ay + ab - K) = 0 \qquad (4)$$

From (2) $\Sigma xy^2 - b\Sigma xy - a\Sigma y^2 + ab\Sigma y - K\Sigma y - b\Sigma xy + b^2\Sigma x + ab\Sigma y - nab^2 + nKb = 0$ (5)

From (3) $\Sigma x^2y - b\Sigma x^2 - a\Sigma xy + ab\Sigma x - K\Sigma x - a\Sigma xy + ab\Sigma x + a^2\Sigma y - na^2b + nKa = 0$ (6)

From (4) $\Sigma xy - b\Sigma x - a\Sigma y + nab - nK = 0$ (7)

n being the number of pairs of observed values in the summation

From (5) $$K = \frac{\Sigma xy^2 - 2b\Sigma xy - a\Sigma y^2 + 2ab\Sigma y + b^2\Sigma x - nab^2}{\Sigma y - nb} \qquad (8)$$

From (6) $$K = \frac{\Sigma x^2y - 2a\Sigma xy - b\Sigma x^2 + 2ab\Sigma x + a^2\Sigma y - na^2b}{\Sigma x - na} \qquad (9)$$

From (7) $$K = \frac{\Sigma xy - b\Sigma x - a\Sigma y + nab}{n} \qquad (10)$$

From (8) and (10) $n\Sigma xy^2 - 2nb\Sigma xy - na\Sigma y^2 + 2nab\Sigma y + nb^2\Sigma x - n^2ab^2$

$= \Sigma y\Sigma xy - b\Sigma x\Sigma y - a(\Sigma y)^2 + nab\Sigma y - nb\Sigma xy + nb^2\Sigma x + nab\Sigma y - n^2ab^2$ (11)

or $$b = \frac{\Sigma y\Sigma xy - n\Sigma xy^2 - a\left[(\Sigma y)^2 - n\Sigma y^2\right]}{\Sigma x\Sigma y - n\Sigma xy} \qquad (12)$$

From (9) and (10) $n\Sigma x^2y - 2na\Sigma xy - nb\Sigma x^2 + 2nab\Sigma x + na^2\Sigma y - n^2a^2b$

$= \Sigma x\Sigma xy - b(\Sigma x)^2 - a\Sigma x\Sigma y + nab\Sigma x - an\Sigma xy + na^2\Sigma y + nab\Sigma x - n^2a^2b$ (13)

or $$b = \frac{\Sigma x \Sigma xy - n\Sigma x^2 y - a(\Sigma x \Sigma y - n\Sigma xy)}{(\Sigma x)^2 - n\Sigma x^2} \qquad (14)$$

For convenience we write

$$A = \Sigma x \Sigma y - n\Sigma xy \qquad D = (\Sigma x)^2 - n\Sigma x^2$$
$$B = \Sigma x \Sigma xy - n\Sigma x^2 y \qquad E = (\Sigma y)^2 - n\Sigma y^2$$
$$C = \Sigma y \Sigma xy - n\Sigma xy^2$$

Then from (12) and (14) $$b = \frac{C - aE}{A} = \frac{B - aA}{D} \qquad (15)$$

Clearing (15) $$CD - aDE = AB - aA^2 \qquad (16)$$

and $$a = \frac{AB - CD}{A^2 - DE}$$

The value of b can then be determined from (15)

The value of K can then be determined from (10).

Discussion

G. V. Pickwell.[2] The author has presented clearly and simply an accurate and positive method of establishing the relative fatigue value of any steel suitable for use in this type of application. The following discussion is intended in no way to detract from the value of this method of procedure but, if possible, to enhance the value of these tests from a practical standpoint.

Preparation of Specimens. Care in the preparation of specimens for a laboratory test is always important, especially where any new material is to be compared, under the most favorable conditions, with other materials which have been tested under like conditions. However, the writer feels that it would be entirely too costly to handle production springs in the manner described by the author, especially, as pertaining to the method of grinding, compressing, repressing, and regrinding.

Assuming then, that in the production of springs on a commercial basis, considerably less accuracy and care would be permissible, it can safely be assumed that fatigue values obtainable would be somewhat lower than those resulting from the author's methods. Allowances then, should be made in the use of results obtained by these methods.

Effect of Creep in High-Pitch Springs. In the preparation of specimens, the procedure described by the author requires that springs be wound to various pitches, which results in various solid stresses. This procedure, as admitted by the author, has a "marked effect on the endurance" of the springs. Therefore, why not eliminate this variable either by testing all of the springs on a "not-pressed" basis in which the disturbing element of cold work would not be present, or winding them all to the same pitch before pressing, wherein the effect of cold work would be constant? Various test stresses could be produced by varying the stroke of the test machine or varying the number of active coils in the test specimens where a variable stroke is not available.

Effect of Hysteresis. Hysteresis in a spring, as evidenced by the load-deflection curve, has a marked effect upon the accuracy and results of a fatigue test. This is especially true of a spring made with an extremely high pitch and subsequently subjected to cold working by compressing solid. It should be given some consideration here. The author mentions this phenomenon and suggests a method of compensating for it during the test but makes no recommendation for correcting it. The writer has found that "bluing" after pressing, i.e., drawing at 550 to 600 F, for 30 min, has a very beneficial effect in reducing the hysteresis lag and change in load due to cold work. This should also help to eliminate the effect of these variables on the endurance of the specimens.

Plotting Results. For a long time, it has been a recognized fact that most endurance-test results follow rather closely a logarithmic curve. The plotted results of such a test usually approximate a straight line on a graph of logarithmic values in both directions. The writer plotted the results of the example given by the author on full log-log paper and found that the points lie more nearly on a straight line than they do on the curve as plotted on semilog paper. Any radical departure from this straight line very definitely indicates a stress of infinite endurance.

Incidentally, it indicates a lower endurance limit than that determined by the author's method. This is because the curve should coincide more nearly with the minimum values obtained in the test rather than represent an average of all the values.

Effect of Hardness. Another variable not recognized in this paper is the effect of final hardness of the steel on the fatigue life of the springs. Hardness will affect the endurance under load in at least two ways:

1 The solid load stress, as pressed, will increase with increase in hardness. It is generally recognized that endurance limits rise in some proportion to rising physical properties. From this, it would appear that the maximum endurance limits could be obtained by hardening the springs to their maximum safe hardness. This is true within certain limits. The endurance limit, as determined experimentally and under ideal conditions, might be the highest obtainable in a spring of, say, 50 C Rockwell, yet the springs might be dangerously brittle under conditions of shock loading or "impact loading," as mentioned by the author. The fatigue life above the endurance-limit stress might also be much less than if a softer material were used. It is quite evident then, that extreme hardness should be avoided except where high stresses are necessary and very smooth operation is possible.

2 Working in the opposite direction, the softer the steel is the more cold work it can stand, in pressing, without danger of breaking. This permits bringing into play a greater proportion of the cross-sectional area, hence a greater load-carrying capacity with lower surface fiber stresses, as compared to a much harder material. The advantage of this set of conditions is that steel of inferior quality, as related to minor surface defects, scratches, decarburization, etc., may be used more safely than an extremely hard steel.

Therefore, conditions of design and type of service will largely dictate the hardness requirements most suitable for each particular application.

If space is limited and working conditions are free from impact and shock, it might be advisable to use comparatively high hardness where necessary to impose higher stresses. Extra care in eliminating surface and other defects should then be exercised.

If working conditions are more severe and ample space is provided, a lower hardness may be used and the springs given considerable cold work in pressing. This will require more material for the job but cheaper steel and cheaper manufacturing processes may be suitable.

Carl Thumim.[3] The lucidity with which the author presents the testing procedure and the method of calculating results will undoubtedly improve the situation of obtaining comparable data from different sources.

The form of the S-N equation is such, that if $N = a$, S becomes infinite. This is obviously extrapolation carried to extremes and should be definitely guarded against. To be sure that there is no misunderstanding, the curve should properly be drawn with a horizontal section at the maximum-stress reading from 0 cycles up to the curved portion, as shown in Fig. 4 of this discussion.

We agree with the author that the S-N data provide an oppor-

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[3] Design Engineer, Power Circuit Breaker Division, Philadelphia Works, General Electric Company. Mem. A.S.M.E.

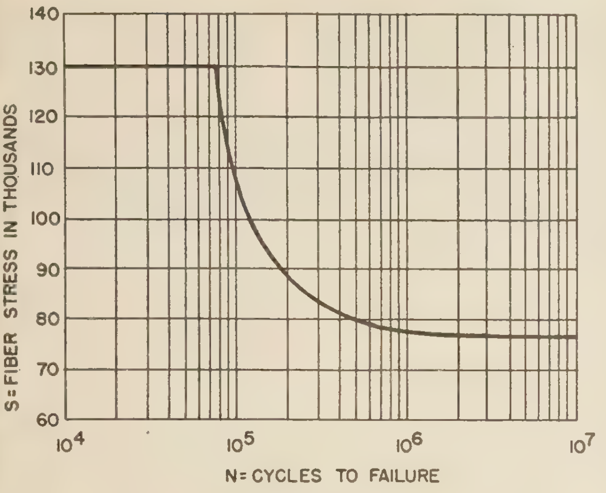

Fig. 4 Corrected Form of Curve Derived From S-N Equation

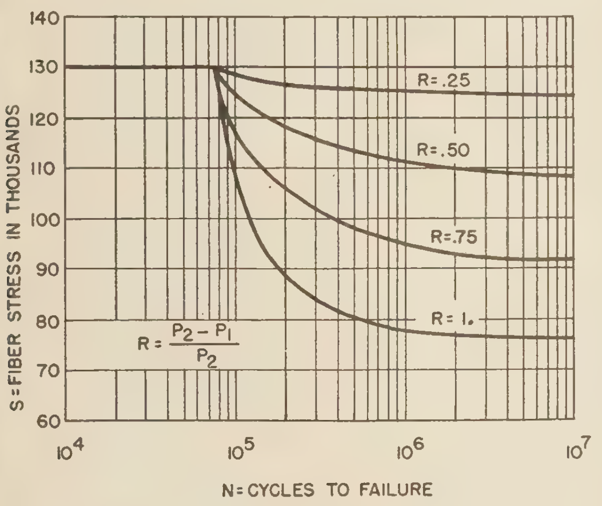

Fig. 5 Family of S-N Curves Derived From Different Values of R

tunity for balanced design. By its use it is possible to employ stresses higher than the endurance limit, which not only saves cost as stated, but reduces the space required. This may make a saving much more important than the difference in spring costs alone. Many applications to apparatus already designed or even in the hands of the user would be impossible if every factor reducing the space requirement were not employed.

While the S-N curve supplies a derating factor which reduces the available stress as the number of operations increases, it is possible in many cases to improve this factor and, therefore, the space-cost efficiency, by taking advantage of the stress-range factor. This often makes it possible to use the highest stress with the highest number of cycles. In a large proportion of applications the spring is not used from its free length to its minimum usable compressed length, but starts with an initial load which in many cases is a large proportion of its final load. If P_2 is taken as the final load and P_1 the initial load, then the stress-range ratio $R = (P_2 - P_1)/P_2$. When $P_1 = 0$, the S-N curve described in the paper obtains. As P_1 increases, R decreases and S decreases. Fig. 5 of this discussion shows a family of curves added to the original curve and plotted from data[4] obtained from a manufacturer of springs. To get the full benefit from S-N investigations, the code of procedure should be enlarged to call for testing for several values of initial spring compression.

Author's Closure

Mr. Pickwell's comments are very pertinent, and bring out clearly some conditions which are well worth attention. Considering them in detail:

Preparation of Specimens. The writer disagrees with Mr. Pickwell's remarks only to the extent that he does not believe the special care used in preparation and calibration of the specimens, and the conduct of the tests, would perceptibly increase the endurance values obtained, as compared with commercially made springs. What this special procedure accomplishes is the insurance of highly consistent and reproducible test results, of peculiar importance in plotting and interpreting an S-N diagram.

Effect of Creep. Mr. Pickwell's suggestions do not seem practical. The trouble is that the endurance limit, for the type of springs tested at Wright Field, is 70,000 to 75,000 psi or more; and at "solid" stresses ranging upward from 80,000 to 85,000 psi permanent set is noted, of magnitude increasing very rapidly as the solid stress increases. Therefore it is impossible to apply a graded series of fatigue-test stresses without encountering considerable permanent set in the more highly stressed springs.

Then two courses are open: We could (a) coil all the springs alike, to develop a common (high) solid stress, or (b) follow the practice described in the Code. Procedure (b), being the one more nearly in accord with good design practice, is the recommendation. Procedure (a) would afford interesting comparisons; the S-N diagram would be quite different. Several isolated comparisons, illustrating the point, were developed in the Wright Field tests. In every case, of two springs coiled to show different solid stress values but fatigue tested in a common stress range, the spring with the lower solid stress showed better endurance, in one case by a ratio of two to one.

Effect of Hysteresis. Mr. Pickwell here opens up a fascinating subject. His statement as to the effect of "bluing," i.e., a strain relief draw after testing, on the hysteresis loop, is quite correct. Some spring engineers are familiar with this phenomenon, but the only extensive discussion known to the writer is the article by L. E. Adams, the English authority, in the Carnegie Scholarship Memoirs for 1937. It includes a rather involved mathematical treatment, and quite a body of experimental results (on straight bars).

A few fatigue tests made under the writer's direction indicate that this strain-relief treatment does not materially improve the endurance. However, these results were not considered as conclusive. The method probably requires further development, and for this reason the writer made no reference to it in the Code.

The other comments by Mr. Pickwell relate to specimen variables rather than to test procedure, and therefore are not within the scope of the paper.

Mr. Thumim's comments on extreme values represented by the S-N equations bring up a very interesting point. In the preliminary study as to the suitability of the formula

$$(N - a)(S - b) = k$$

for representation of fatigue data on ferrous metals, the test data examined were logged values from rotating-beam tests of about ten steels, as reported by Professor Moore from his work at the University of Illinois Experiment Station. In every case

[4] This information is based on a large number of tests made by the Hunter Pressed Steel Company, Lansdale, Pa.

examined the value found for the constant a was *negative*, that is, the equations all predicted a finite stress for zero cycles. The individual stress values, in all but two cases, fell roughly about midway between the elastic limit and the ultimate tensile strength. In the two exceptions the values were about equal to the ultimate tensile.

Now the curious point is, that with all the springs tested at Wright Field, the S-N equation for every group, with a single exception, worked out to a *positive* value for a, i.e., the predicted stress was infinite for a positive number of cycles. The exception was a group of springs coiled cold from cold-drawn steel, and the test results on this group were peculiar in some other respects.

The writer would like to have someone offer a plausible explanation of this phenomenon. He himself will not attempt to do so, but will merely call attention to the fact that it is almost impossible to break a really well made compression spring by a static compression test. Under extreme test conditions the spring may take an enormous amount of permanent set, but it will almost never fail by fracture.

Heat Transfer to Hydrogen-Nitrogen Mixtures Inside Tubes

By A. P. COLBURN,[1] NEWARK, DEL., and C. A. COGHLAN,[2] BEACON, N. Y.

This paper gives results of experiments in which data were determined on heat transfer to air, to nitrogen, and to eight mixtures of hydrogen and nitrogen, ranging from 8.85 to 98 per cent hydrogen, flowing inside a steam-jacketed tube, 0.5 in. inside diam and 48.75 in. long. The data were well correlated when plotted as $\frac{h}{C_pG}\left(\frac{C_p\mu}{k}\right)^{2/3}$ versus $\frac{DG}{\mu}$, with the value of $\frac{C_p\mu}{k}$ ranging from 0.45 to 0.73. The data checked heat-transfer data of Nusselt and showed deviation from friction analogies.

Nomenclature

THE following nomenclature is used in the paper:

C_p = heat capacity of gas at constant pressure
c = rate of collection of condensate, lb per hr
D = inside diameter of tube
f = friction factor in Fanning equation
G = weight velocity of gas flow through tube
h = gas-film coefficient of heat transfer
j = heat-transfer factor = $\frac{h}{C_pG}\left(\frac{C_p\mu}{k}\right)^{2/3}$
k = thermal conductivity of gas
q_c = heat of condensation = $970(c - 0.0258)$, Btu per hr
q_G = heat gained by gas = $wC_p(t_2 - t_1)$, Btu per hr
t_1 = inlet temperature of gas, F
t_2 = exit temperature of gas, F
t_s = steam temperature, F
w = rate of air flow, lb per hr
μ = viscosity of gas
ϕ = function

Prandtl's number = $\frac{C_p\mu}{k}$

Reynolds' number = $\frac{DG}{\mu}$

Purpose of Investigation

Heat transfer to mixtures of hydrogen and nitrogen is becoming of increasing importance because large amounts of such mixtures are being processed in the manufacture of ammonia, and because of the use of hydrogen and hydrogen mixtures as the cooling media in rotating electrical machines. Furthermore, the properties of gas mixtures containing hydrogen vary greatly from the ideal-mixture rule; and there is a good basis for doubt whether heat-transfer formulas, derived from single gases, can be applied to such mixtures. At the same time, the variation in properties suggests the possibility of verifying or disproving the equations in common use for the calculation of heat transfer. It was the purpose of this investigation, therefore, to obtain data on heat transfer to hydrogen-nitrogen mixtures and to use the results to study various heat-transfer formulas.

This problem was attacked by Brunot (1)[3] who was particularly interested in hydrogen used to cool electrical machinery. He tested a commercial air cooler of the extended-surface type, using a mixture of 8.22 per cent nitogren in hydrogen, as well as air. From the results on air, values were predicted for the mixture, using proper values of properties for the mixture, and good agreement was shown with the experimental results. Brunot made an extensive review of data on the physical properties of hydrogen mixtures from the literature and his results are given in Figs. 1 and 2.

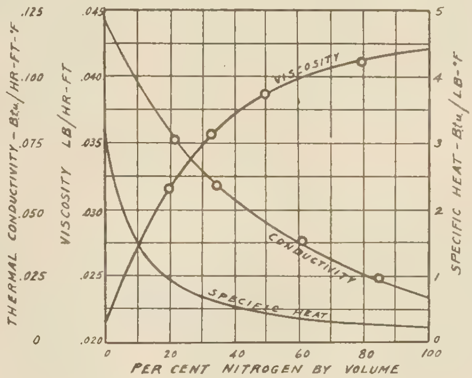

Fig. 1 Properties of Hydrogen-Nitrogen Mixtures at 70 F and 1 Atm Abs Pressure, From Brunot (1)

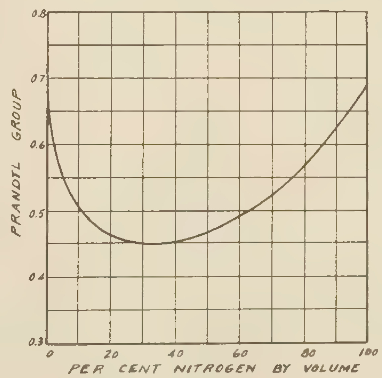

Fig. 2 Prandtl's Number as a Function of Composition for Hydrogen-Nitrogen Mixtures, From Brunot (1)

The property of chief interest is the Prandtl number, $C_p\mu/k$. The value of this number for pure hydrogen and for pure nitrogen is practically the same 0.73. For mixtures of these gases, the Prandtl number is less than for the pure components, having a minimum value of 0.45 at about 70 per cent hydrogen. Most heat-transfer formulas, which have been generalized to hold for

[1] Associate Professor of Chemical Engineering, University of Delaware.

[2] Chemical Engineer, The Texas Company.

Contributed by the Heat Transfer Professional Group, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

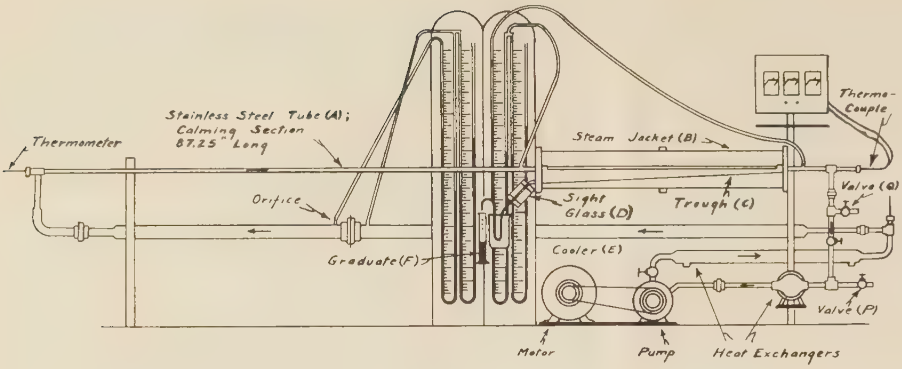

Fig. 3 Schematic Arrangement of Experimental Apparatus

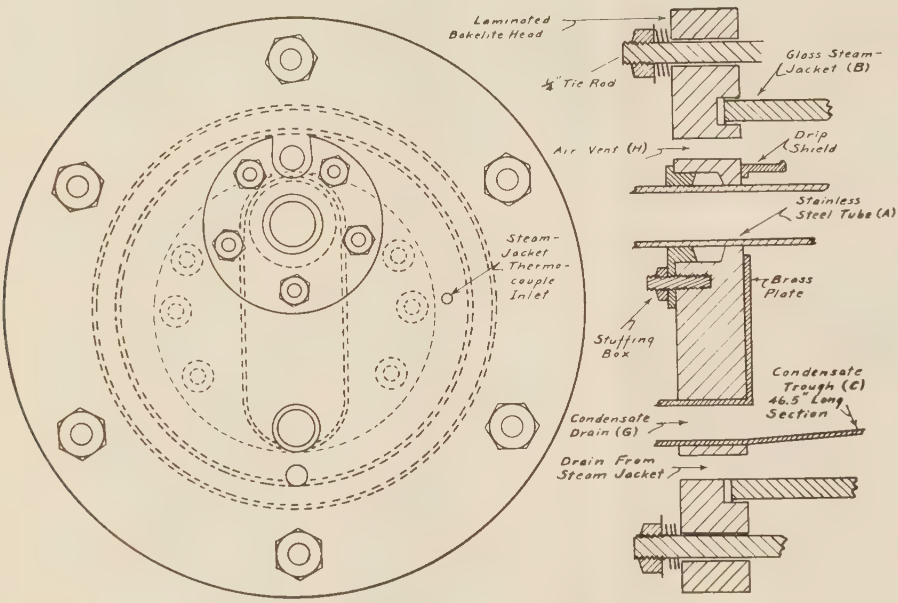

Fig. 4 Assembly of Experimental Apparatus at Gas-Inlet End

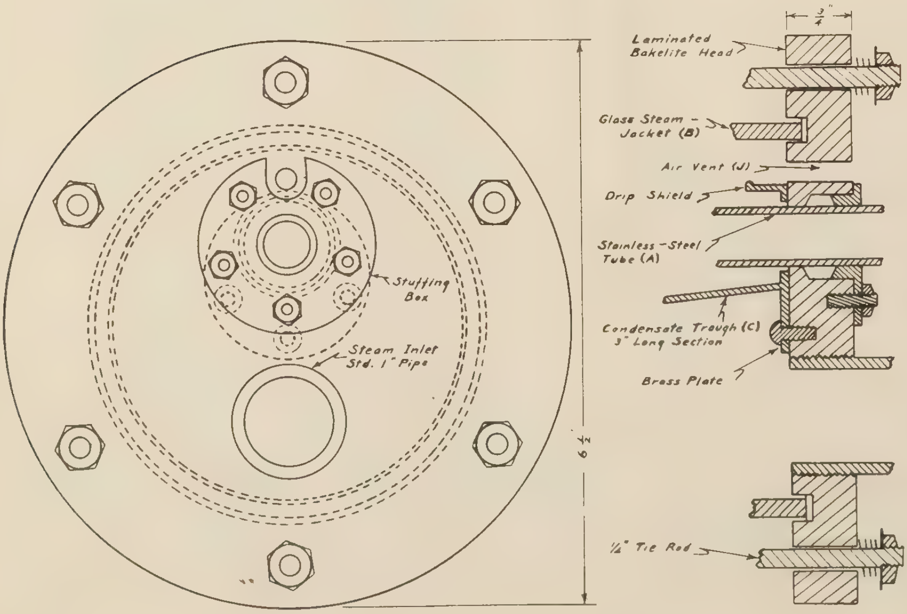

Fig. 5 Assembly of Experimental Apparatus at Gas-Outlet End

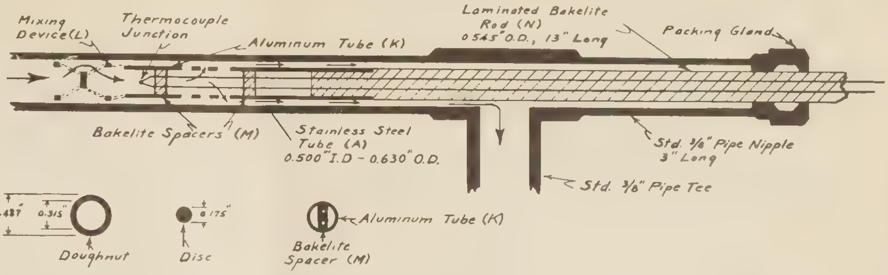

Fig. 6 Longitudinal-Section View of Outlet From Test Section, Showing Schematic Assembly of Thermocouple Used to Measure Outlet Temperature

both gases and liquids, include the Prandtl number. For gas mixtures, however, one might question whether the proper value to use is the Prandtl number of the mixture or some combination of the Prandtl numbers of the pure components.

If the Prandtl number of the gas mixture is the proper value to use in heat-transfer formulas, then data covering a range of values of this number from 0.73 to 0.45 might make possible a selection of the best type of formula. Chilton (2) has shown a convenient comparison of the exponential-type relation with the theoretical relations of Prandtl (3) and of von Kármán (4). These relations are different in the range of Prandtl numbers covered by these mixtures, and so the results should prove of great interest in testing these equations.

Details of Apparatus Used

The apparatus chosen for this investigation was of the gas-in-tubes type, inasmuch as it was felt that the most reliable results could be obtained in such an apparatus. A stainless-steel tube was used so that there would be no corrosion and change of surface conditions, and also because the low thermal conductivity of stainless steel would minimize heat conduction through the ends of the apparatus. At the same time the thermal conductivity is sufficiently great to make the thermal resistance of the tube negligible for the transfer of heat to the gas. The tube was steam-jacketed, the outer cylinder being glass so that the type of condensation on the tube could be observed. The condensate from the tube was collected in a trough located inside the steam space and led outside the apparatus. The condensate on the glass jacket, resulting from radiation losses, was drained separately and discarded. This arrangement was made in the attempt to obtain good heat balances. Atmospheric steam was supplied to the jacket, and some steam was continuously vented at both ends of the jacket to insure that no air would collect. The steam-inlet pipe to the jacket is 1 in. diam and is located below the trough. These conditions permit a very low steam velocity into the exchanger and at an elevation such that any moisture in the steam would not be carried into the trough in which the condensate from the tube collects.

TABLE 1 DIMENSIONS OF TEST SECTION

Stainless-steel tube	Inches
Inside diameter	0.50
Outside diameter	0.63
Total length	142.75
Heated length	48.75
Length of calming section	87.25
Unheated length at outlet end	6.75
Distance between pressure taps	52.50

The arrangement of the apparatus is shown in Fig. 3. A small blower was used to recirculate the gas, and coolers were inserted before and after the blower to bring the gas to room temperature at the entrance of the test apparatus. The rate of flow of the gas was measured by an orifice meter constructed with throat taps, according to the specifications of the Fluid Meters Report. The metering tube was 2 in. diam, and orifices used were of 0.375, and 0.255 in. diam. These were calibrated on air, using a calibrated gas-meter prover, and the coefficient was found to be 0.61. Dimensions of the test section are given in Table 1.

The glass steam jacket was supported by laminated-bakelite flanges, shown in Figs. 4 and 5, at the ends of the heated section, drawn together by tie rods outside the glass jacket. These flanges were made tight to the stainless-steel tube by packing glands. Owing to the low thermal conductivity of these flanges, very little heat could be conducted to the tube beyond the inside surfaces of the flanges.

The heat, given up by the steam in heating the gas, was determined by measuring the condensate collecting in the trough and draining through the sight glass at constant level. This amount was corrected for the small amount of liquid collecting on the flanges above the trough, and for the small heat loss from the sight glass; this correction was determined by blank runs when no gas was flowing, and was found to be 0.0258 lb per hr.

Temperature Determination

The temperatures were determined as follows: The inlet-gas temperature was measured by a calibrated thermometer at the entrance to the calming section. This temperature was adjusted by the coolers to be identical with the room temperature adjacent to the calming section, so that no increase or decrease in the temperature of the gas would take place before it reached the heated portion of the tube. The outlet temperature was measured with a special thermocouple, shown in Fig. 6, which was constructed to minimize radiation errors and to insure good mixing of the gas before the thermocouple. It was made so that it could be slid out of the way when pressure drops were measured. The aluminum tube K acted as a radiation shield. The mixing device L consisted of two "doughnuts" and one disk of copper held in place by being soldered to copper wires. This mixer was then painted with a thermosetting bakelite varnish and cured, in order to minimize heat conduction from the tube. Laminated-bakelite spacers M kept the thermocouple centered. The steam temperature was also measured with a thermocouple in the jacket, and checked against barometric pressure.

Commercial hydrogen and nitrogen were used, and the composition of the mixture was determined immediately following a series of runs by means of a Bureau of Mines apparatus. Although a special packing gland was built onto the blower, there was a slight leak at that point, and a small amount of air would leak into the apparatus. This was determined by analyzing the mixture for oxygen. This amount, less than 1 per cent, was calculated as nitrogen. The gas pressure in the system was essentially atmospheric. Actually, enough gas to cause a slight pressure was supplied to the system at the beginning of a run. This pressure soon dropped until atmospheric pressure prevailed at the stuffing box of the blower.

Results of Experiments

The principal results of this investigation are given in Table 2 and in Figs. 7 to 12. The reliability of the data is measured by the heat balances, the deviations from which are given by the

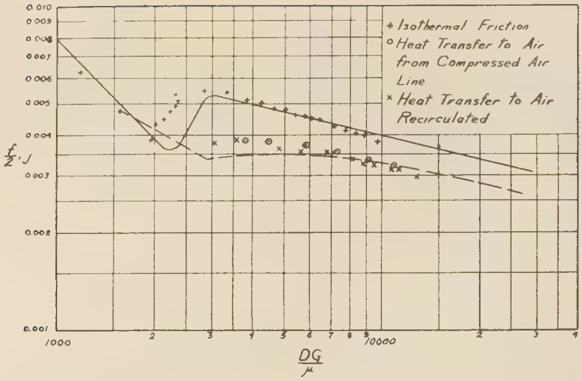

Fig. 7 Experimental Results Including Isothermal-Friction Data and Heat-Transfer Data on Air

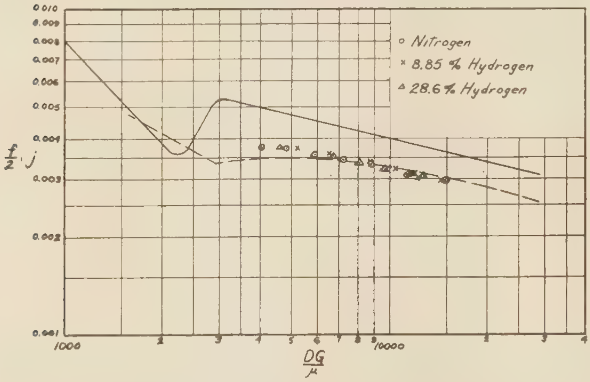

Fig. 8 Experimental Results on Heat Transfer to Pure Nitrogen and to Hydrogen-Nitrogen Mixtures Containing 8.68 and 28.6 Per Cent Hydrogen by Volume

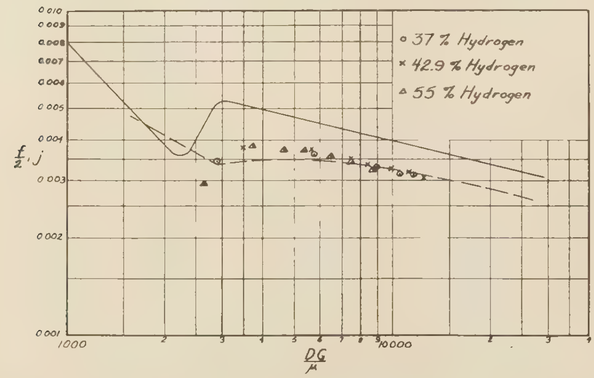

Fig. 9 Experimental Results on Heat Transfer to Hydrogen-Nitrogen Mixtures Containing 37, 42.9, and 55 Per Cent Hydrogen by Volume

column of data $(q_c - q_G)/q_G$. In most cases, this deviation is much less than 10 per cent. The quantity q_c is calculated from the rate of condensation c, corrected for the blank run, or (c — 0.0258) lb per hr, and the latent heat of steam at atmospheric pressure, 970 Btu per lb. The quantity q_G is calculated from the rate of flow of the gas, the temperature rise, and the specific heat of the mixture. The latter value was taken from Fig. 1 for the given mixture at 70 F, and corrected for the slight change with temperature to the average of inlet and outlet temperatures of the gas. The over-all heat-transfer coefficient was used as the gas-film value, inasmuch as the steam resistance was negligible

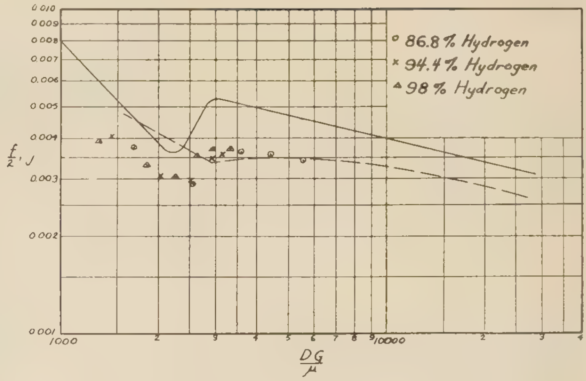

Fig. 10 Experimental Results on Heat Transfer to Hydrogen-Nitrogen Mixtures Containing 86.8, 94.4, and 98 Per Cent Hydrogen by Volume

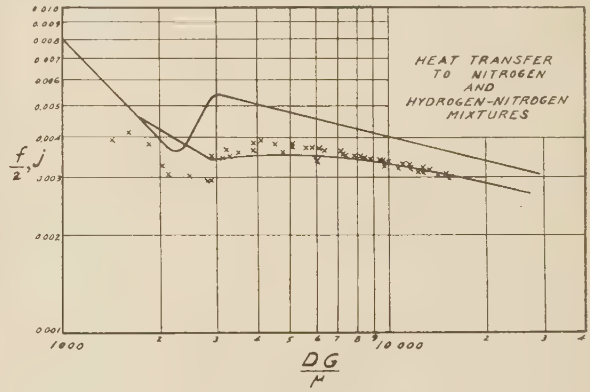

Fig. 11 Summary of Experimental Results on Heat Transfer to Nitrogen and Hydrogen-Nitrogen Mixtures

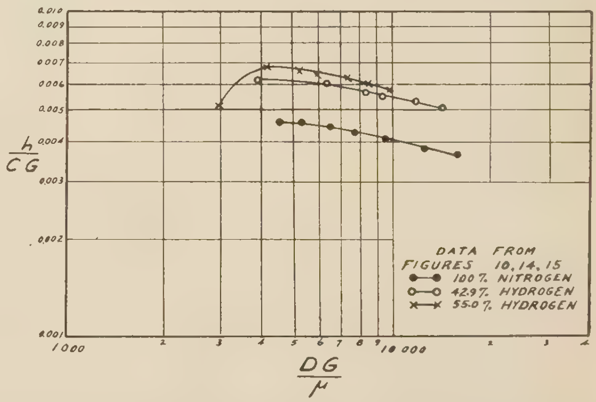

Fig. 12 Experimental Results on Heat Transfer to Nitrogen and Hydrogen-Nitrogen Mixtures Plotted Without Including Prandtl's Number

(condensation being dropwise) and the tube resistance was negligible. The logarithmic-mean temperature difference was employed in all runs for uniformity, although some of the runs are in the viscous region.

As a check on the apparatus, isothermal-friction data on air and heat-transfer data on air are shown in Fig. 7. On all data plots, the solid line is the friction line for smooth pipes as found by Drew, Koo, and McAdams (5), and the dashed line is a heat-transfer line, plotted according to Equation [1], representing the

TABLE 2 EXPERIMENTAL DATA AND CALCULATED RESULTS

Run No.	Gas	t_1 F	t_2 F	t_s F	c lb per hr	w lb per hr	$\frac{q_c - q_G}{q_G}$ x 100	j	$\frac{DG}{\mu}$
A 1	Air*	74.4	190.4	211.8	0.213	6.48	0.0	0.00388	3800
2		75.6	190.4	211.8	0.247	7.65	+1.0	0.00386	4490
3		75.0	189.0	211.8	0.311	9.90	+1.1	0.00373	5820
4		75.2	187.0	211.8	0.373	12.33	+0.2	0.00355	7300
5		75.2	184.8	211.8	0.443	15.22	−1.3	0.00338	9050
6		75.2	182.6	211.8	0.536	17.34	+1.8	0.00321	10950
B 1	Air**	75.2	189.4	211.6	0.190	5.23	+11.3	0.00378	3050
2		76.1	189.8	211.0	0.175	6.16	−13.8	0.00386	3600
3		74.9	188.0	211.6	0.261	8.29	+ 5.8	0.00365	4840
4		75.2	186.7	211.0	0.291	9.65	− 0.4	0.00358	5640
5		74.3	186.2	211.6	0.352	11.62	+ 1.2	0.00352	6800
6		73.4	184.0	211.1	0.389	13.85	− 4.8	0.00338	8100
7		74.8	182.7	211.0	0.433	15.10	+ 0.9	0.00327	8800
8		75.2	182.2	211.0	0.429	16.21	− 6.4	0.00323	9500
9		74.7	181.2	211.3	0.526	18.40	+ 2.8	0.00315	10750
10		73.4	180.4	211.0	0.533	19.10	0.0	0.00313	11200
11		75.6	179.2	211.3	0.595	21.90	+ 1.3	0.00300	12800
C 1	N_2	67.1	188.8	212.4	0.245	6.65	+ 5.0	0.00378	4050
2		67.1	188.0	212.2	0.255	7.80	− 3.9	0.00373	4800
3		68.4	185.8	212.4	0.318	9.55	+ 0.3	0.00360	5880
4		67.5	185.0	212.4	0.376	11.65	− 0.7	0.00346	7160
5		67.5	183.2	212.6	0.461	14.11	+ 3.0	0.00333	8700
6		69.3	180.0	212.4	0.563	18.44	+ 1.6	0.00310	11340
7		72.0	179.2	212.8	0.654	22.70	− 0.2	0.00298	14950
D 1	8.85%	74.5	192.0	211.1	0.285	7.81	− 1.2	0.00372	5130
2	H_2	74.5	190.6	211.1	0.353	9.91	− 2.3	0.00359	6510
3		74.5	188.4	211.1	0.465	13.17	+ 4.3	0.00339	8640
4		74.3	186.4	211.1	0.529	15.80	+ 0.8	0.00324	10360
5		74.1	185.2	211.1	0.595	18.21	− 0.2	0.00315	11950
6		73.8	183.6	211.1	0.637	20.15	+ 1.8	0.00304	12240
7		74.3	182.8	211.3	0.713	22.07	+ 1.9	0.00297	14480
E 1	28.6%	74.8	197.8	212.0	0.322	6.92	+ 3.8	0.00378	4640
2	H_2	75.0	195.4	212.0	0.451	9.93	+ 1.8	0.00352	6660
3		74.1	193.8	212.0	0.529	12.00	+ 0.4	0.00339	8050
4		75.2	192.0	212.0	0.628	14.40	+ 2.6	0.00321	9660
5		74.8	190.6	212.0	0.755	17.58	+ 2.5	0.00311	11790
6		73.2	189.8	212.0	0.803	18.88	+ 1.2	0.00305	12670
F 1	37%	72.0	195.2	212.0	0.216	4.23	− 7.1	0.00342	2900
2	H_2	71.8	197.2	212.0	0.430	8.38	− 2.4	0.00363	5760
3		72.3	195.6	212.0	0.552	10.90	− 0.4	0.00346	7490
4		72.5	193.8	212.0	0.638	12.97	− 1.1	0.00330	8900
5		72.7	192.4	212.0	0.735	15.15	− 0.5	0.00317	10400
6		70.5	191.6	212.0	0.844	16.90	+ 1.4	0.00313	11610
G 1	42.9%	73.4	198.8	211.1	0.304	5.05	+ 2.3	0.00380	3500
2	H_2	74.3	198.4	211.3	0.468	8.13	− 3.3	0.00372	5640
3		73.6	196.4	211.3	0.610	10.72	− 6.6	0.00350	7430
4		73.7	195.2	211.3	0.683	12.16	+ 3.5	0.00338	8430
5		73.7	194.0	211.3	0.794	14.37	+ 3.2	0.00327	9970
6		73.6	193.2	211.5	0.920	16.46	+ 5.6	0.00319	11420
7		73.4	191.6	211.3	1.006	17.94	+ 7.7	0.00308	12450
H 1	55%	72.8	191.6	212.0	0.263	3.64	+ 4.7	0.00293	2620
2	H_2	73.6	201.0	212.0	0.365	5.16	− 2.1	0.00387	3720
3		73.8	200.2	212.0	0.460	6.44	+ 1.4	0.00375	4640
4		71.1	199.0	211.3	0.518	7.40	+ 0.3	0.00372	5330
5		71.6	197.8	211.3	0.656	9.04	+ 5.0	0.00358	6520
6		72.0	196.6	211.5	0.722	10.38	+ 2.3	0.00341	7470
7		72.0	195.2	211.8	0.833	12.15	+ 2.6	0.00326	8750
I 1	86.8%	72.7	200.6	212.0	0.313	1.79	− 4.4	0.00373	1690
2	H_2	72.9	192.2	212.0	0.449	2.72	− 1.5	0.00290	2580
3		73.1	200.0	212.0	0.616	3.79	− 7.3	0.00365	3600
4		73.2	199.5	212.0	0.750	4.67	− 7.4	0.00358	4430
5		71.4	197.8	212.0	0.955	5.88	− 0.4	0.00341	5580
J 1	94.4%	71.4	198.8	211.5	0.331	1.30	− 7.0	0.00408	1445
2	H_2	72.5	188.4	211.5	0.455	1.84	− 2.4	0.00307	2030
3		72.1	187.0	211.5	0.558	2.29	− 1.4	0.00297	2525
4		71.0	193.0	211.5	0.670	2.67	− 3.7	0.00347	2940
5		71.4	194.2	211.5	0.885	2.84	− 8.6	0.00358	3130
K 1	98%	71.6	196.2	211.7	0.375	1.10	−10.3	0.00392	1317
2	H_2	71.3	187.2	211.7	0.455	1.56	−16.4	0.00330	1868
3		72.8	184.6	211.5	0.579	1.90	− 8.7	0.00309	2281
4		72.8	190.6	211.5	0.724	2.21	− 5.6	0.00356	2641
5		72.7	192.0	211.5	0.829	2.47	− 4.1	0.00370	2961
6		72.2	192.0	211.5	0.918	2.78	− 6.0	0.00371	3330

*Air from compressed air line.

**Air recirculated by blower.

data of Nusselt (6, 7) The friction data are shown in Fig. 7 to be in excellent agreement with the literature. In order to check a possible effect of the blower in recirculating the gases, heat-transfer data on air were obtained by using both air from the compressed-air line in the laboratory (where fluctuations had been well smoothed out by a surge tank and some 100 ft of pipe) and air recirculated by the blower. There is no appreciable difference in the results as will be seen from Fig. 7, and, furthermore, the data are seen to be in excellent agreement with those of Nusselt.

A plot of data on pure nitogen, shown in Fig. 8, is practically identical with that of the data on air shown in the previous figure. In calculating Reynolds' numbers, values of viscosity were obtained from Fig. 1 and then corrected to the mean gas temperature in the tube, the latter correction meaning an increase in the value of viscosity by about 10 per cent.

Data on mixtures of hydrogen and nitrogen are shown in Figs. 8 to 12. In calculating the ordinates for these plots, values of the Prandtl number for the mixtures were taken from Fig. 2. For these gases, the value of Prandtl's number is practically independent of pressure and temperature over moderate ranges, so that the values given in Fig. 2 for 70 F could be satisfactorily used for the temperatures encountered in these experiments. The close agreement of the results in Figs. 8 to 12, with the previously plotted data on air, and with the line representing the Nusselt air data, is proof that the proper value to use in calculating heat transfer for mixtures is the value of the Prandtl number of the mixture.

As final indication of the necessity of including this factor, Fig. 12 shows $h/(C_pG)$ plotted versus Reynolds' number. The strong divergence of the lines for the different mixtures proves the importance of the Prandtl number of the mixture in bringing the results into agreement.

The effect of the Prandtl number can be shown, from the results in Fig. 12, to be at least as important as the $^2/_3$ power, as often used in the exponential formula

$$\frac{h}{C_pG}\left(\frac{C_p\mu}{k}\right)^{2/3} = \phi\left(\frac{DG}{\mu}\right) \ldots\ldots[1]$$

In the range of values of Prandtl's number between 0.45 and 0.7, the Prandtl equation would predict about one half this effect, and the von Kármán equation something in between as shown by Chilton (2). While these results in themselves may not be sufficient to disprove these latter two equations, they are at least indications that the simple exponential type is more satisfactory for gas mixtures

than the more theoretical ones, and suggest that further improvements in the theory are possible.

These data also verify the Nusselt data on gases in showing a deviation from the analogy between fluid friction and heat transfer. If this analogy (7) held, the heat-transfer and friction lines on the plots would be coincident in the turbulent region. At low Reynolds' numbers, the divergence may possibly be explained by the presence of a greater "dip" region for heat transfer between viscous and turbulent flow than for friction. However, in the strong turbulent region, the heat-transfer factors are still a good 10 per cent below the friction factors. Above a Reynolds number of 10,000 the slope of the line representing the heat-transfer data is —0.2 but at lower Reynolds' numbers this slope becomes zero and then of opposite direction in the "dip" region.

There are a few results for the mixtures in the region of viscous flow, and while these seem to bear out the general shape of curves in the "dip" region and in the viscous region, there are not sufficient data to draw definite conclusions, other than that the Prandtl number appears as calculated.

While the results of this study, together with those of Brunot (1), appear to prove that the Prandtl number applies for gases according to Equation [1], they show that further study of the analogies between fluid friction and heat transfer is highly desirable.

BIBLIOGRAPHY

1 "Properties of Hydrogen Mixtures," by A. W. Brunot, Trans. A.S.M.E., October, 1940, pp. 613–616.

2 "Engineering in the Service of Chemistry," by Thomas H. Chilton, *Industrial and Engineering Chemistry*, vol. 32, January, 1940, pp. 23–31.

3 "Eine Beziehung zwischen Wärmeaustausch und Strömungswiderstand der Flüssigkeiten," by L. Prandtl, *Physikalische Zeitschrift*, vol. 11, 1910, pp. 1072–1078.

4 "The Analogy Between Fluid Friction and Heat Transfer," by Th. von Kármán, *Engineering*, vol. 148, 1939, pp. 210–213.

5 "The Friction Factor for Clean Round Pipes," by T. B. Drew, E. C. Koo, and W. H. McAdams, Trans. American Institute of Chemical Engineers, vol. 28, 1932, pp. 56–72.

6 "Der Wärmeübergang in Rohrleitungen," by W. Nusselt, *V.D.I. Mitteilungen über Forschungsarbeiten*, Heft 89, 1910, pp. 1–38; also *Zeit. V.D.I.*, vol. 53, 1909, pp. 1750–1755, 1808–1812.

7 "A Method of Correlating Forced Convection Heat Transfer Data and a Comparison With Fluid Friction," by A. P. Colburn, Trans. American Institute of Chemical Engineers, vol. 29, 1933, pp. 174–209.

Discussion

R. H. Norris.[4] It is of interest to compare the authors' test results for the region of viscous flow with theoretical results, even though, as the authors admit, their test data in this region are too scanty to be conclusive.

Fig. 13 of this discussion shows points representing the authors' test results compared with curves evaluated from a recently

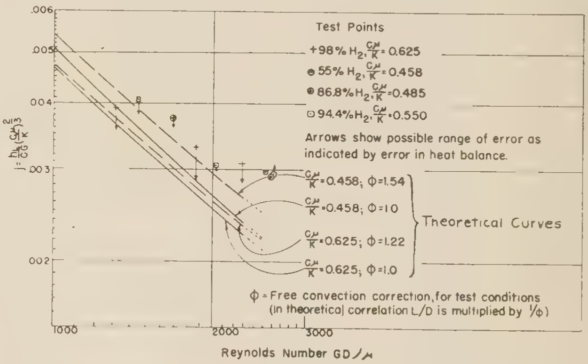

Fig. 13 Comparison of Test Points of Colburn and Coghlan With Theoretical Curves of Norris and Streid for Hydrogen-Nitrogen Mixtures in the Laminar-Flow Region, for Flow in a Pipe With $L/D = 97.5$

published[5] correlation of theoretical results, using the logarithmic-mean temperature difference basis. When the empirical correction proposed by Colburn (7) for free convection is included, and the possible range of error of the test results indicated by the heat-balance discrepancy is allowed for, the agreement between test and theory is reasonably good (of the order of 10 per cent), below Reynolds' number of 2200. For higher Reynolds' numbers, transition to turbulent flow has presumably begun. The fact that the test values somewhat exceed the theoretical values may indicate that the correction for free convection here applied to the latter is not quite sufficient, or that the flow is not completely laminar.

[4] General Engineering Laboratory, General Electric Company, Schenectady, N. Y. Jun. A.S.M.E.

[5] "Laminar-Flow Heat-Transfer Coefficients for Ducts," by R. H. Norris and D. D. Streid, Trans. A.S.M.E., vol. 62, August, 1940, pp. 525–533.

Electric-Slip Couplings for Use With Diesel Engines

By A. D. Andriola,[1] Groton, Conn.

The fact that, in the last 12 months, at least twenty Diesel-driven vessels have been equipped with electromagnetic-slip couplings in this country indicates its value for ship-propulsion purposes. This paper explains the functions of the device as (1) to reduce torque-variation intensity at the reduction gears, and (2) to permit two or more engines to be rapidly coupled and uncoupled to and from the gearing to a common propeller shaft. Elements of the system are described and theoretical principles of the mechanism are analyzed. A brief comparison is given of the electric-slip coupling with the hydraulic system. The paper concludes with mention of additional applications in conjunction with internal-combustion engines.

The progress made in Diesel-engine design, in terms of reduced specific weight and size, has been achieved mainly by substantial increases in rotative speed, working pressure, and number of cylinders per unit. This trend has brought to the fore many important problems. Not the least of these is that of dealing satisfactorily with torsional vibration.

In marine installations, especially, two factors combine to make this a problem of major importance: (1) The operating range extends over a large portion of the span from zero to maximum speed. (2) Efficient propeller speeds are such as to require a speed-reducing device when high-speed engines are used. Mechanical gearing is preferred because of the attendant economy, simplicity, and efficiency, as compared to other types. Experience, however, shows susceptibility to wear and failure, unless proper precautions are taken to limit vibration transmission from the engine to the gearing. In special cases, these difficulties have been entirely obviated by the adoption of the Diesel-electric system of propulsion. This alternative involves a substantially higher first cost and a lower over-all efficiency and does not recommend itself to wide commercial usage. In this connection, therefore, the recent use of electric couplings in geared-marine-Diesel installations is of considerable interest.

Functions of Electric Coupling

Briefly, the electric coupling is a device for transmitting torque electromagnetically across an air gap, there being no mechanical connection between the coupling halves. Units of widely different characteristics have been developed for various uses, but the type which is being applied to marine service utilizes induction-motor principles and is termed the electric-slip coupling.

The idea of transmitting torque through an air gap is not new. As early as 1921, a coupling of this type intended for marine use was built and tested by Sperry,[2] but apparently was never put into service. The first commercial application recorded is that made by the Swedish firm, Allmanna Svenska Elektriska Aktiebolaget, or A.S.E.A., in a pilot-boat installation in 1935. Use of the coupling on a large-scale basis has since proceeded steadily. In this country alone, at least twenty vessels, totaling 48 units, were equipped last year. Credit for the particular embodiment of the marine type of coupling is also shared by the Westinghouse and Elliott Companies, whose engineers were working along identical lines at the same time as A.S.E.A.

The primary functions of the device are (1) to reduce torque-variation intensity at the gears, and (2) to permit two or more engines to be rapidly coupled and uncoupled to and from the gearing to a common propeller shaft. Several typical arrangements are shown in Fig. 1. Incidental to these uses, other important advantages are simultaneously obtained in the way of power flexibility and vessel maneuverability, closely approximating those of the Diesel-electric drive. These have been discussed in part by Metz and Ericson[3] and are of sufficient importance to warrant restatement here; these are as follows:

1 Increased reliability of the plant is achieved by the use of a multiple-engined arrangement.

2 Any one unit may be shut down for repairs without stopping the vessel.

3 Economic cruising at partial speeds is possible by operating only a portion of the available units.

4 The engines may be conveniently warmed up at the dock prior to vessel departure.

5 Maneuvering in close waters or during docking can be facilitated by operating some of the engines in the ahead direction and the remainder astern. The coupling between gearing and either set of units can be rapidly made or broken by means of a simple switch. An appreciable saving in starting air is thereby also effected.

6 When reversing, as in an emergency, there is no need to wait until the inertia of the entire system is dissipated. Instead, the engines may be uncoupled, reversed, and torque applied to the propeller shaft while the latter is still turning in the ahead direction. The coupling is therefore capable of braking effect.

An earlier analysis[3] of coupling-torque transmission under vibratory conditions indicates that the action was not completely understood. Nevertheless, the operating experience accumulated since establishes the coupling as very well suited to the requirements of Diesel geared drives. It has also indicated other applications in which the coupling characteristics may prove advantageous either as a vibration controller or as a clutch, or both.

Elements of Electromagnetic Coupling

Fig. 2 shows the main components of the coupling in elementary form. The unit consists of two concentric rotors, a set of slip rings and brushes, an external source of direct-current supply, and a control panel. One rotor is a multipole-magnet ring with individual poles energized from the slip rings mounted on the supporting shaft. The second rotor comprises a short-

[1] Engineer in charge of Engine Calculating Department, Electric Boat Company. Jun. A.S.M.E.

[2] "Compounding the Combustion Engine," by E. A. Sperry, Trans. A.S.M.E., vol. 43, 1921, pp. 677–716.

Presented at the National Meeting of the Oil and Gas Power Division, Asbury Park, N. J., June 19–22, 1940, of The American Society of Mechanical Engineers.

[3] "Electromagnetic Slip Couplings for Use With Geared Diesel Engines for Ship Propulsion," by G. L. E. Metz and N. Ericson, Trans. Institute of Marine Engineers, vol. 49, 1937-1938, pp. 237–248.

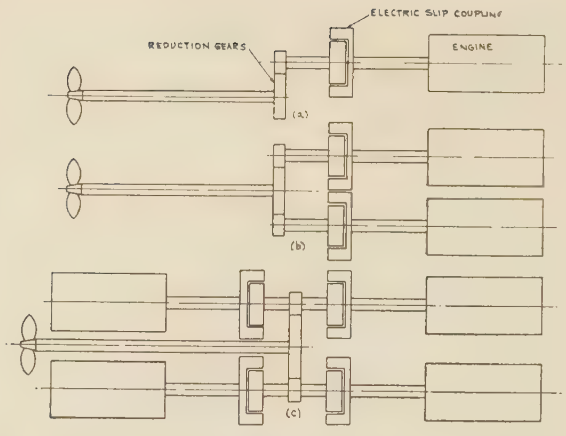

FIG. 1 TYPICAL INSTALLATIONS OF ELECTRIC-SLIP COUPLINGS TO GEARED MARINE-DIESEL PROPULSION SYSTEM

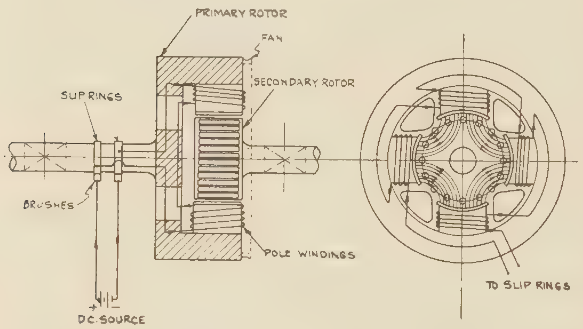

FIG. 2 COMPONENT PARTS OF TYPICAL ELECTROMAGNETIC-SLIP COUPLINGS

circuited winding of the squirrel-cage type. As illustrated in Fig. 2, the pole ring is shown on the outer rotor, but the position of the windings may be reversed without affecting the action of the coupling. Both arrangements have been used in practice. From a structural standpoint, the short-circuited winding is inherently the more rugged of the two and should preferably be used as the driving rotor. Of the two arrangements, the latter results in a higher natural frequency of the engine system; this is generally desirable from a vibration point of view. Irrespective of position, the pole ring and the short-circuited member are termed the "primary" and "secondary" windings, respectively, to denote the magnetic path.

Torque transmission across rotors occurs electromagnetically. If the primary is energized, relative rotation of the two members causes the secondary conductors to cut across a magnetic field of alternating polarity. The currents thereby induced set up an interrotor torque, tending to rotate the driven portion in the direction of the driving rotor. It is clear that the two members can never rotate at the same speed for, without relative rotation, or slip, torque cannot be induced. Under normal conditions, however, slip is an extremely small percentage of the driving speed. At full load, for example, its magnitude is about 1 per cent and torque-transmission efficiency is consequently high. A set of typical slip-torque-coupling characteristics for full and partial excitation values is shown in Fig. 3. Interrotor torque varies from a linear function of slip at very low slip speeds to a maximum at from 7 to 10 per cent slip. Beyond this point, the coupling characteristic is definitely unstable and application of torque, in excess of the maximum shown, results in stalling. For proper operation, the coupling must be designed so that the stalling torque is substantially above that to be transmitted.

With Diesel engines, the normal torque variation at rated load may approach ±100 per cent, depending upon the number of cylinders. A.S.E.A. practice is to design for 170 per cent of full mean torque; this value is somewhat higher than the torque variation which obtains in Diesel units of six or more cylinders.

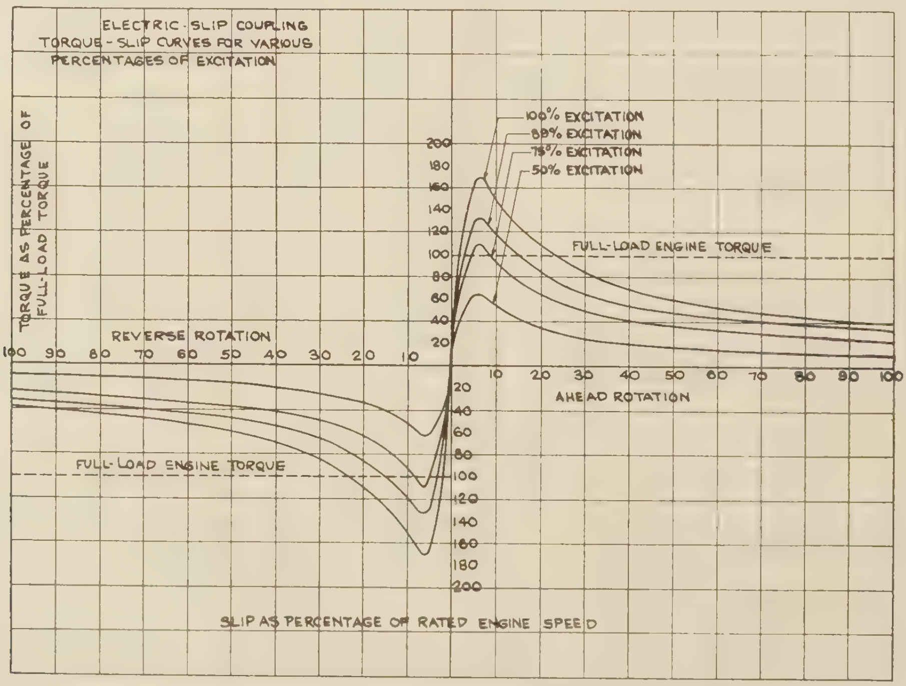

FIG. 3 ELECTRIC-SLIP COUPLING TORQUE-SLIP CURVES FOR VARIOUS PERCENTAGES OF EXCITATION

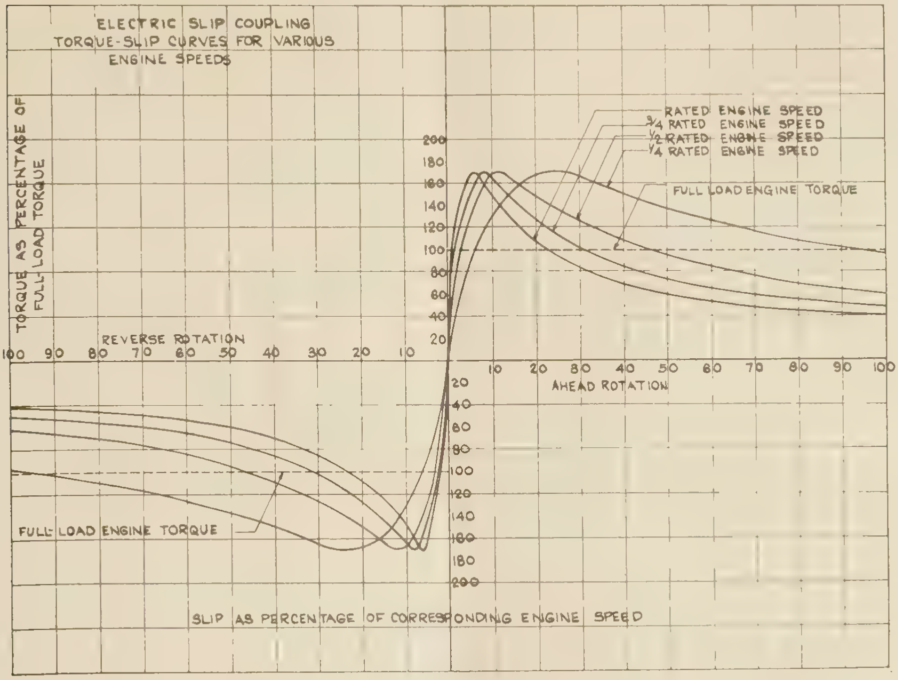

Fig. 4 Electric-Slip Coupling Torque-Slip Curves for Various Engine Speeds

Under very rapid torque variation, electrical effects momentarily increase the stalling point to about 300 per cent of rated capacity, further reducing the chance of instability under average driving conditions. The stalling feature of the coupling, however, is desirable since the maximum shafting stress can thus be held to a predetermined limit. This is especially true of systems subject to seizure on the driven portion of the system. In marine installations, a suddenly fouled propeller would constitute such a seizure. The stress imposed on the engine members under such conditions might conceivably cause failure in a mechanically integral system.

The induced-coupling torque depends only upon the actual slip between rotors, i.e., it is independent of engine speed. Consequently, at starting or low-speed conditions, the total torque which can be developed at the engine is available for load pick-up. Torque-slip curves for partial speeds are shown in Fig. 4; the increase in torque at 100 per cent slip with decrease in engine speed is clearly indicated.

Construction Similar to Squirrel-Cage Motor

The similarity between the coupling and the short-circuited squirrel-cage induction motor, as regards the general construction, electromagnetic action, and slip-torque characteristics, is readily apparent. In fact the two units may be said to be practically identical except as to the means used to obtain a rotating flux. In the three-phase alternating-current induction motor, for example, three-phase alternating current is supplied to the stator windings and produces a rotating flux with a speed dependent upon the supply frequency and the number of poles. In the electric-slip coupling, this is accomplished by mechanical rotation of a direct-current excited-pole ring and the flux speed is equal to the mechanical speed of the primary rotor. Otherwise, the basic electrical considerations are the same for both units.

Noticeable deviations from usual induction-motor practice are of a constructional nature and are dictated by mechanical considerations which the induction motor is not required to satisfy. These pertain to the method of rotor support, which has been previously described as mechanically separated. Actually, the two rotors are overhung on their respective shafts, a pilot bearing being purposely omitted to accommodate a certain degree of misalignment between the driving and driven portions of the system. This construction requires use of an air gap appreciably larger than for the induction motor. A.S.E.A. couplings employ air gaps of 0.2 to 0.4 in., depending upon the rated capacity of the coupling. Compared with induction-motor practice of air gap $= 0.15\ \sqrt[4]{KW}$ the gap for a 1000-hp coupling is about 4 times as large as that for a motor of identical capacity. The reluctance of the magnetic path is largely that of the air gap, and thus for equal flux densities the coupling requires a proportionately larger excitation energy. Nevertheless, excitation loss in the coupling does not exceed 1 to 2 per cent of the rated capacity. Over-all transmission efficiency, accounting for both slip and excitation losses, is approximately 97 per cent. The overhung-rotor construction, however, does introduce a mechanical problem of some importance. When the two rotors are displaced from the concentric position, the resultant eccentricity is accompanied by a proportional unbalanced radial pull in the direction of the smaller air gap. If unrestricted, this force would ultimately result in mechanical contact of the two rotors accompanied by magnetic locking. To avoid this effect, it is only necessary to design the supporting shafts so that the load-deflection characteristic is steeper than the unbalanced pull-eccentricity relation of the rotors.

Cooling requirements are met in a manner similar to that employed for motors. For this purpose a radial-bladed fan is built into the outer rotor at its extreme edge as shown in Fig. 2. Openings in the end flange of this same member provide a natural flow path for air over both windings. The effectiveness of the arrangement is a function of coupling speed. No difficulty, however, is encountered even when reasonably long sustained periods of operation at low speeds is a requirement, since the

excitation current is normally reduced with correspondingly less heat to be dissipated.

Method of Coupling Control

Coupling control under operating conditions is quite simple. The coupling is made or broken instantaneously by operation of a switch located at the engine control board. Thus, both engine and coupling are operated by the engine attendant. Normally, the excitation circuit is kept closed and the engines handled exactly as in a direct-connected installation. During docking or channel maneuvering, the engines may be left running and the propeller shaft controlled by means of the excitation switch. Where several units drive a common shaft, the load must be divided equally among the engines, i.e., the slip at each coupling must be the same. This latter quantity is measured by a simple stroboscopic arrangement and the necessary adjustment then made by variation of the fuel supply. The stroboscope consists of two concentric bands with a number of equally spaced holes drilled in their peripheries. One band is carried on each of the two coupling rotors. The relative rotational movement can be clearly seen when a light source is placed within the inner band; measurement is then accurately made by a stop watch.

Two years ago, the author had the opportunity to observe the operation of two A.S.E.A. couplings during a trip of several days' duration aboard the Swedish motor vessel *Astri*. No occasion arose for an emergency-speed reversal, but all other normal maneuvers at docking and in free route were carried out. Changes or reversals in speed were rapidly made without signs of shock or lag between engine and propeller speeds. Some of these operations were timed and may be of interest; the interval measured was that from the initial telegraph bell to the moment the propeller shaft reached speed, and includes the time taken by the attendant to reset the telegraph prior to adjusting the engine throttle. Incidentally, the excitation circuit was kept closed at all times. The values follow:

From	To	Time, sec
Stop	1/2 Speed ahead	10
1/2 Speed ahead	Stop	10
Stop	1/2 Speed astern	14

These times represent average performance.

When a wide operating speed range, of the order of 3 to 1, is used, the excitation current is usually reduced for low-speed operation. This is done by means of an additional resistance placed in series with the excitation circuit. By such control, the efficiency of the coupling can be held approximately to a constant value over the entire operating range.

Vibratory Characteristics of Coupling

The slip-torque characteristics previously discussed are those which obtain under conditions of uniform driving torque. Actually, the torque delivered by the Diesel engine is periodic in nature and may contain large variations from the mean value. The net motion at the driving rotor is a combination of harmonic motions of differing frequencies superposed upon a uniform rotation. Tests, conducted on early installations, showed a marked suppression of oscillating motion across the coupling and the action was interpreted as equivalent to that of a viscous-fluid device. Recent experimental and analytical data, however, show that the electrical effects which obtain under vibratory conditions give rise to both elastic and damping components of torque within the coupling. This effect was first brought to the author's attention by G. J. Dashefsky of the New York Navy Yard, who noticed the characteristic during tests conducted on a coupling furnished by the Westinghouse Company. Model tests recently made available by the A.S.E.A. to the Electric Boat Company and tests on a full-size coupling made at the latter plant confirm Dashefsky's observations. A mathematical analysis has also been developed by two members of the A.S.E.A. staff, Dr. Dreyfus and H. Arnemo, and is given in the Appendix. Values computed on this basis agreed closely with those obtained experimentally on the model coupling; these tests incidentally covered a vibratory range from zero to 20 cycles per sec frequency at the driving rotor. Since the preparation of this paper, a similar mathematical analysis was presented to the American Institute of Electrical Engineers (A.I.E.E.) by Lory, Kilgore, and Baudry.[4] This approach differs from that of A.S.E.A., and, since a check of results is afforded, no duplication is involved in presentation of the latter.

Frequency Range

The frequency range of practical interest is considerably higher than that covered in the model-coupling tests. Calculations for an 850-hp 460-rpm coupling have been made by the author up to a value of 100 cycles per sec. This corresponds to a natural frequency of 6000 vibrations per min and covers in general all critical speeds of practical interest in marine drives. The computed values are given in Fig. 5. At low frequencies, the damping component is larger and the two become equal, in this particular design, at about 5 cycles per sec. Beyond this point, the decrease in elasticity is large compared to that of the damping. Both components may be considered as constant in value above 30 cycles per sec. In a previous reference,[4] the elastic characteristic is compared to a weak mechanical spring, and the tendency is to treat the driving and driven systems as separate entities. While in many cases the treatment is perfectly applicable, it must nevertheless be noted that the term "weak" is a comparative one and therefore somewhat misleading. In marine installations, where a long length of shafting from the engine to the propeller is involved, it is possible for the line-shaft flexibility to exceed that of the coupling. While the condition is not a usual one, its possibility shows that the device should only be applied on the basis of a careful analysis of the entire vibratory system.

Generally, the elasticity of both the driving and driven portions will be quite low compared to that of the coupling. The effect of the coupling elasticity under these conditions is to introduce an additional mode of vibration of very low frequency, compared to those of either portion treated separately. The higher modes of vibration correspond closely to those obtained for the separate portions of the system. In the fundamental mode, all masses to one side of the coupling move practically in phase. The accompanying stress in the shafting is consequently very low. Actually, the system is equivalent to one of two masses, i.e., the driving and driven portions connected by the coupling elasticity. The dynamic effect may be evaluated simply, if the damping between rotors is neglected. Let m = circular frequency of forced vibration; J_1, J_2 = driving and driven inertias, respectively; K = coupling elasticity; p = circular frequency of natural vibration, i.e., $p = \sqrt{[K(J_1 + J_2)/J_1J_2]}$; $M \sin mt$ = vibratory torque delivered to the driven member. Then it can be shown that

$$M_1 = M \frac{J_2}{(J_1 + J_2)} \frac{1}{[1 - (m/p)^2]} \sin mt \ldots\ldots\ldots [1]$$

In a twin-engined single-screw geared drive the phase relation between torque components delivered by each engine will modify Equation [1] as follows: Introducing the additional notation r = vibratory order under consideration, ψ = phase angle between similar cranks of the two engines, and J_1 = moment of

[4] "Electric Couplings," by M. R. Lory, L. A. Kilgore, and R. A. Baudry, Trans. A.I.E.E., vol. 59, 1940, pp. 423–428.

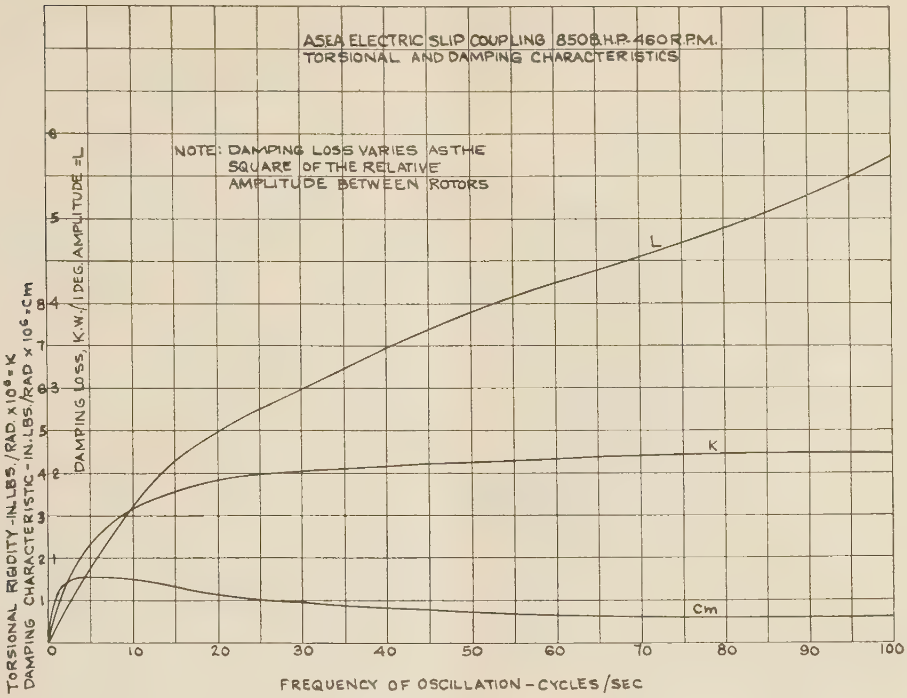

Fig. 5 Torsional and Damping Characteristics of an A.S.E.A. Electric-Slip Coupling; 850 Bhp, 460 Rpm

inertia of each engine, we obtain (1) with both engines in phase, i.e., $\sin (\psi r/2) = 0$

$$M_1 = M \frac{J_2}{(2J_1 + J_2)} \frac{1}{[1 - (m/p)^2]} \sin mt \qquad [2]$$

(2) engines 180 deg out of phase, i.e., $\cos (\psi r/2) = 0$ and each engine vibrates against the other about the gear as a node

$$M_1 = M \frac{1}{[1 - (m/p)^2]} \sin mt \qquad [3]$$

The effectiveness of flexibility between engine and gearing is a function of the ratio of forced to natural frequencies of vibration (m/p). If (m/p) is less than 1, the flexible element actually increases torque variation at the gears and defeats the purpose for which it is intended. Reduction obtains only when (m/p) is greater than $\sqrt{2}$. The criterion for satisfactory gear operation is that $(T - M_1)$ be positive, where T is the mean torque. Otherwise, the gears separate periodically with resultant noise and wear. In a twin-engine electric-coupling drive, a given phase relationship obviously cannot be maintained. Equation [3] then applies, since it gives the minimum torque reduction possible.

Investigation of the Resonant Range

In the previous analysis, damping was neglected to show clearly the similarity of requirements for various driving arrangements. The damping component will modify the quantitative results, and must be accounted for in investigation of the resonant range. Theoretically, only the major orders need be considered for the fundamental vibration mode; all engine members oscillate in phase. Actually, the minor orders, $^1/_2$ to 2, inclusive (four-cycle engines of six or more cylinders), may produce undesirable effects. These occur at considerably higher speeds than the major orders and may fall within the operating range. An analysis for a two-mass system with damping between masses is given here; the extension to the multiple-engine arrangement may be carried out in similar fashion. The following nomenclature is used.

Nomenclature

θ = angular displacement, radians
$\dot{\theta}$ = angular velocity $= d\theta/dt$, radians per sec
$\ddot{\theta}$ = angular acceleration $= d^2\theta/dt^2$, radians per sec^2
c = damping constant, in-lb per radian per sec
ϕ = phase angle of motion, radians
M = harmonic torque acting on J, in-lb
$j = \sqrt{-1}$
1, 2 = subscripts

The differential equations of motion are

$$\left.\begin{array}{l} J_1\ddot{\theta}_1 + K(\theta_1 - \theta_2) + c(\dot{\theta}_1 - \dot{\theta}_2) = M \\ J_2\ddot{\theta}_2 - K(\theta_1 - \theta_2) - c(\dot{\theta}_1 - \dot{\theta}_2) = 0 \end{array}\right\} \qquad [4]$$

Writing the harmonic torque as a complex quantity, the solution is given by

$$\theta = \lambda(\cos mt + j \sin mt)$$

Substituting this in Equation [4] and reducing, the following values are obtained:

For magnitude of vibratory displacement

$$\left.\begin{array}{l} \lambda_1 = \dfrac{M}{m^2(J_1 + J_2)} \sqrt{\dfrac{(K - J_2m^2)^2 + c^2m^2}{K^2[(m^2/p^2) - 1]^2 + c^2m^2}} \\ \lambda_2 = \dfrac{M}{m^2(J_1 + J_2)} \sqrt{\dfrac{K^2 + c^2m^2}{K^2[(m^2/p^2) - 1]^2 + c^2m^2}} \end{array}\right\} \qquad [5]$$

For phase angles of motion

$$\left.\begin{aligned}\phi_1 &= \tan^{-1}\frac{-cm^3J_2^2}{\{K(K-J_2m^2)[(m^2/p^2)-1]-c^2m^2\}(J_1+J_2)}\\ \phi_2 &= \tan^{-1}\frac{cm^3\,J_1J_2}{\{K^2[(m^2/p^2)-1]-c^2m^2\}(J_1+J_2)}\end{aligned}\right\}\ldots[6]$$

The torque at the gears is given by

$$M_1 = K(\theta_1-\theta_2) = \frac{MJ_2K}{(J_1+J_2)}\sqrt{\frac{1}{K^2[(m^2/p^2)-1]^2+c^2m^2}}\ldots[7]$$

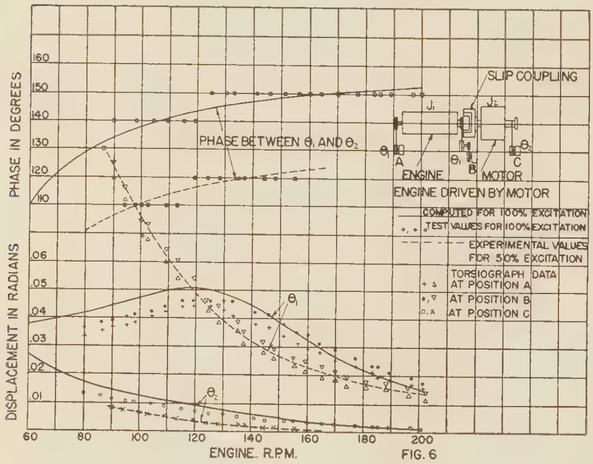

Fig. 6 Comparison of Estimated and Recorded Values for Resonant Ranges Controlled by Coupling Elasticity

At resonance $m/p = 1$, and Equations [5], [6], and [7] reduce to

$$\left.\begin{aligned}\lambda_1 &= \frac{M}{cm^3(J_1+J_2)}\sqrt{(K-J_2m^2)^2+c^2m^2}\\ \lambda_2 &= \frac{M}{cm^3(J_1+J_2)}\sqrt{(K^2+c^2m^2)}\end{aligned}\right\}\ldots\ldots[8]$$

$$\left.\begin{aligned}\phi_1 &= \tan^{-1}-KJ_2/-cmJ_1\\ \phi_2 &= \tan^{-1}K/-cm\end{aligned}\right\}\ldots\ldots\ldots\ldots[9]$$

$$M_1 = \frac{MJ_2K}{cm(J_1+J_2)}\ldots\ldots\ldots\ldots\ldots\ldots[10]$$

If the predominant elasticity is that of the coupling, this analysis will yield rather accurate results. A comparison of estimated and recorded values for a system with this disposition of elasticity is given in Fig. 6. These data cover only the resonant range controlled by coupling elasticity. At higher speeds no perceptible motion was recorded at the motor side of the coupling. Sample torsiograms from 95 to 201 rpm are shown in Fig. 7. Since these tests were performed with the motor driving the engine, the harmonic torque at the engine could be accurately estimated. The close agreement between analytical and test values thus directly confirms the electrical theory of the Appendix, by means of which the coupling characteristics were determined.

For the higher modes of vibration the procedure is well known and need not be discussed here.

Comparison of Electromagnetic With Hydraulic Couplings

The largest application of electric-slip couplings has thus far

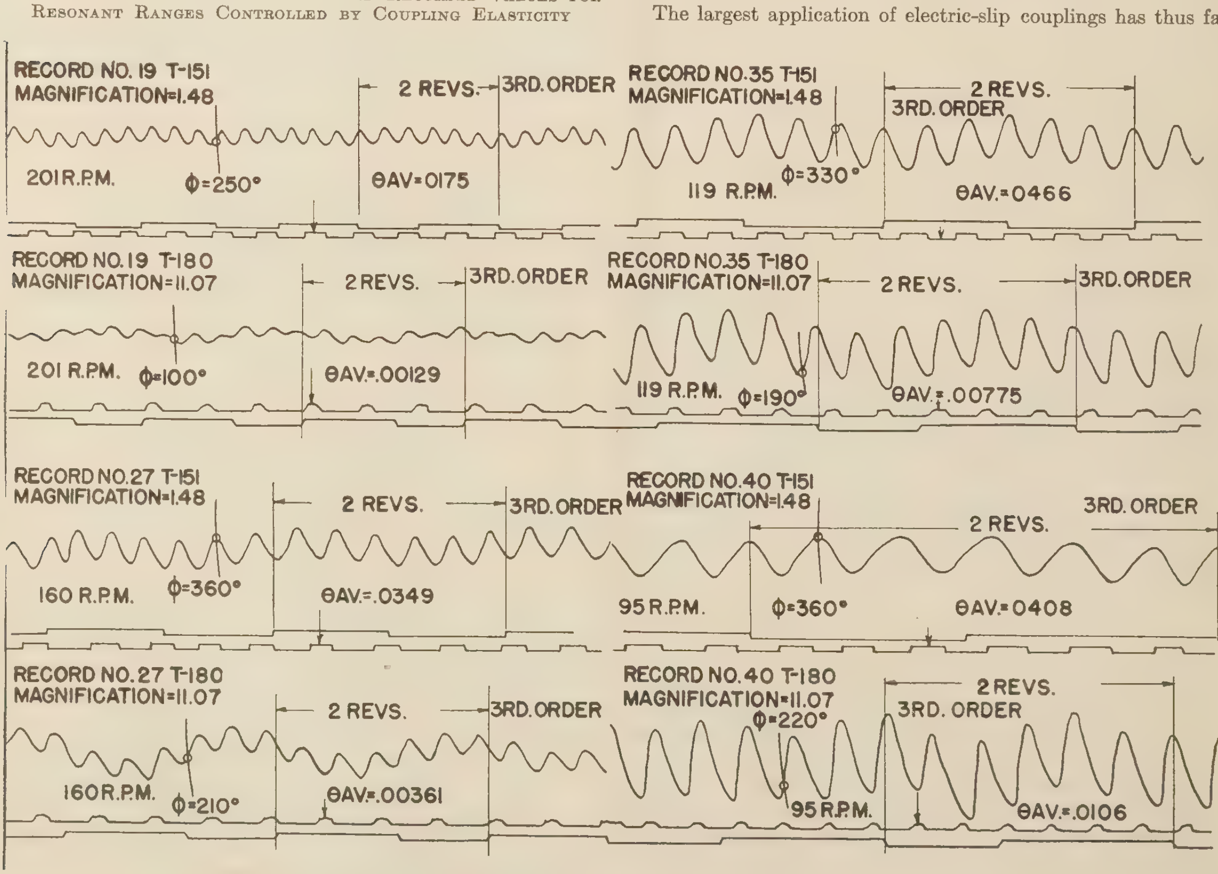

Fig. 7 Sample Torsiogram From 95 to 201 Rpm

been in geared marine drives of the multiple-engine single-screw type, where rapid engine disengagement is desirable. The hydraulic coupling has been similarly employed and a comparison is of interest. Each coupling requires auxiliary equipment and, in this respect, both have disadvantages. In efficiency each averages about 97 per cent. For hydraulic couplings, the loss is approximately a fixed percentage of speed, but the same is true of the electric-slip coupling with economy-resistance control. For speeds below 400 rpm, the electric-slip coupling is definitely superior in weight and size. As more experience with the latter is obtained this limit may be raised considerably. The torque capacity of the electric coupling varies as the diameter squared times length of rotor, or D^2L, and is independent of speed; in the hydraulic coupling the variation is directly as $(\text{diameter})^2 \times$ speed. Within limits, the dimensions of the former may be varied, provided D^2L remains constant. The same facility is lacking in the latter. The mechanically separated rotors of the electric coupling eliminate the possibility of wear, thereby reducing maintenance. Hydraulic couplings of modern design, however, are relatively wear-free and no distinct advantage may be claimed in this respect. As regards rapidity of coupling action, however, the electric coupling has no equal. Where direct current is available, however, adoption of the electric-slip coupling in preference to the hydraulic unit is indicated. Otherwise, with weight and size equal, choice appears to rest on cost.

Additional Applications of Electric-Slip Couplings

Exclusive of geared applications, the coupling characteristics suggest its use for several other services in which the internal-combustion engine is employed. It should be appreciated, a priori, that for a given torque-weight ratio, units with widely different torque-slip properties can be constructed. The quantitative values, in Figs. 3 and 4, apply strictly to the short-circuited squirrel-cage type. With a wound secondary connected through slip rings to a variable resistance, for example, variable-slip-speed characteristics can be obtained. In general, for every known type of motor or generator, a corresponding electric coupling is possible. Additional applications then fall into the following categories:

(*a*) Systems in which seizure of the driven members is likely to occur, such as suction-dredge installation or vessels operating in shallow or ice-filled waters.

(*b*) Direct-drive marine installations required to operate at propeller speeds considerably below one third maximum engine speed. With a wound-rotor construction, slip speeds of the order of 50 per cent are possible for short periods. The losses are approximately proportional to slip but nevertheless small due to the low powers developed in this speed range.

(*c*) Drives requiring smooth-turning characteristics at the load. If in addition rapid clutch action is desired, maximum utility of the device is attained.

From a vibration standpoint alone the electric-slip coupling, properly used, will effect a highly satisfactory drive. Comparable results, however, are also possible with less expensive mechanical devices. Thus, unless some additional function is to be performed, the greater cost involved is not usually justified.

Acknowledgment

The author wishes to express his appreciation to Mr. E. Nibbs, chief engineer, Electric Boat Company, for permission to publish the analysis given in the following Appendix. Also to E. S. Dennison and G. F. Dashefsky for helpful criticism and suggestions.

Appendix

The following nomenclature is used in the Appendix:

Nomenclature

α = $\alpha_0 \sin mt$ = relative angular displacement between coupling rotors, radians
ϕ = flux, maxwells
$\bar{B}$ = direct-current field, gauss
L = length of iron, cm
R = radius of air gap, cm
S, S', S'_0 = current densities, amp per sq cm
ρ = resistance, ohms
f = frequency of oscillation of coupling rotors, cycles per sec
μ = a-c permeability of iron, gauss per oersted
τ_s = slot pitch, cm
h_t = tooth height, cm
a_t = effective tooth area, sq cm
r_t = tooth resistance, ohms
x_t = tooth reactance, ohms
i_{st} = conductor current, amp
i_t = tooth current, amp
z = impedance, ohms
r_s = conductor resistance, ohms
x_s = conductor-leakage reactance, ohms
ζ = specific impedance, ohm per cm periphery per cm length
I_b = sum of currents under pole arc, amp
δ = magnetic air gap, cm
b = effective pole arc, cm
l_k = inductance, h
r_{kb} = short-circuit ring resistance under pole arc, ohms
r_{kl} = short-circuit ring resistance in space between poles, ohms
e = induced voltage in secondary conductor, v
S_a = current density per unit periphery, amp per sq cm per cm
b_k = length of short-circuit ring under pole arc, cm
B = pulsating air-gap field, B includes $\bar{B}$, gauss
n = number of poles
T = induced-coupling torque, kg-m
j = $\sqrt{-1}$

The torque delivered by the engine to the driving rotor of the coupling is a periodic function. Analytically it may, therefore, be considered as composed of a constant torque and a number of harmonic-torque components of differing frequency. Transmission of torque through the coupling then produces a constant slip between the rotors upon which is superposed a relative vibratory motion. In the following analysis the relative speed difference, i.e., slip, is neglected. The relative angular displacement between rotors can then be written as

$$\alpha = \alpha_0 \sin mt \qquad [11]$$

Provided no counteracting ampere turns are induced in either rotor, there arises in the secondary part of the coupling, in each of the gaps between poles, a cross field, shown in Fig. 8 with a flux of

$$2\phi = 2\bar{B}LR\alpha \qquad [12]$$

Actually, the motion gives rise to eddy currents in the iron of both rotors, as well as currents in the cage winding, and the resultant pulsating field has a distribution as shown in Fig. 9. The a-c permeability μ of the iron is a function of ϕ/L, $R\alpha$, and

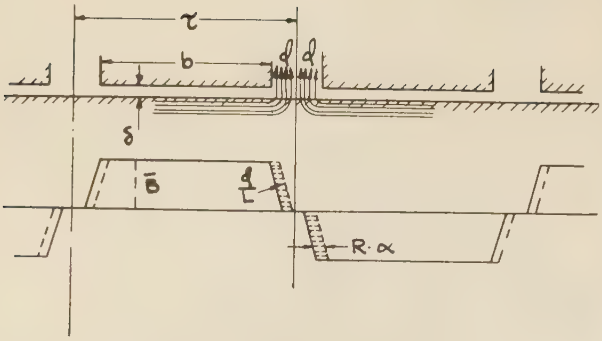

Fig. 8 Cross Field in Secondary Part of Coupling

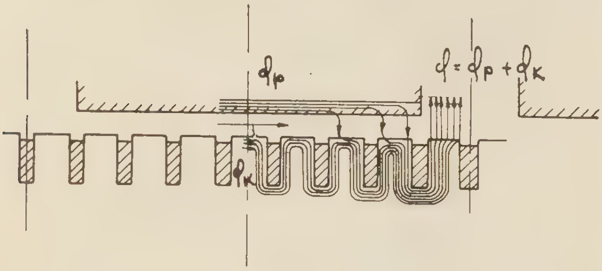

Fig. 9 Distribution of Pulsating Field

m, and is estimated according to previous investigations which show that for f cycles per sec and 50 cycles per sec

$$(\phi/L)_{50} = \sqrt{(f/50)}\ (\phi/L)_f \qquad [13]$$

The a-c permeability in general will be rather high, especially since, at higher frequencies (see Fig. 9) ϕ is divided into ϕ_p along the pole-shoe surface and ϕ_k along the short-circuit winding. In view of this, the ampere turns consumed for the iron will be neglected.

The fact that the teeth lead currents parallel to the cage winding must, however, be taken into account since this will lessen its effective resistance and also influence its leakage reactance. The cage winding is not insulated. To a current density S in the conductor there corresponds a current density S'_0 at the bottom and sides of the slot, as well as at the air-gap surface of the tooth with a value

$$S'_0 = S(\rho_{cu}/\rho_{Fe}) \qquad [14]$$

The decrease in S' in the iron is approximately given by

$$S' = S'_0 e^{-(1-j)\eta x} \qquad [15]$$

where

$$\eta = 2\pi\sqrt{(f\mu/\rho_{Fe})10^{-5}} \qquad [15a]$$

For example, with $\mu = 1000$, $f = 2 \sim 8$ cycles per sec and, $\rho = 0.2$, then

$$\eta = 2\pi\sqrt{0.01} \sim 2\pi\sqrt{0.4} = 2 \sim 4$$

If the teeth are not particularly narrow the current in the conductor $i_{st} = Sbh$ is followed by a current in the tooth equal to

$$i_t = S\frac{\rho_{Cu}}{\rho_{Fe}}\frac{1+j}{2\eta}(\tau_s + 2h_t)$$

the effective tooth area is thus

$$a_t = (\tau_s + 2h_t)/\eta$$

and its resistance and reactance

$$r_t = x_t = \frac{\eta L}{\tau_s + 2h_t}\rho_{Fe}\,10^{-4}$$

Within the active length, tooth and conductor are connected in parallel. Outside this length the conductor has an impedance of $\Delta r_s - j\Delta x_s$ and the effective impedance of one slot pitch is thus

$$z = \frac{(r_s - jx_s)(r_t - jx_t)}{(r_s + r_t) - j(x_s + x_t)} + \Delta r_s - j\Delta x_s \qquad [16]$$

or

$$z = (r_s + \Delta r_t) - j(x_s + \Delta x_s) - \frac{(r_s - jx_s)^2}{(r_s + r_t) - j(x_s + x_t)} \qquad [16a]$$

The cage winding can therefore be considered as a cylinder with a specific impedance

$$\zeta = z\tau_s/L \qquad [17]$$

Differential Equations and Their Solution

The sum of the currents I_b under the pole arc produces a flux of $2\phi_l$ in the interpolar space, i.e.

$$\phi_l = \frac{I_b}{2}\frac{0.4\pi}{\delta}L(y\delta) = \frac{I_b}{2}l_k\,10^{-8} \qquad [18]$$

where the factor y and the inductance l_k are determined by field plotting, as shown in Fig. 10.

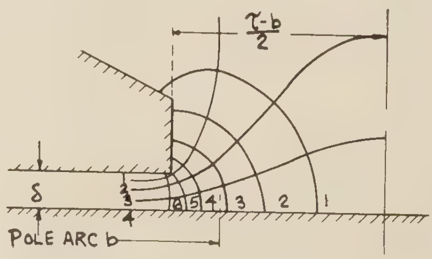

Fig. 10 Method of Field Plotting

The voltage induced in each of the cage-winding conductors under the pole arc is

$$e = -\frac{d\phi}{dt}10^{-8} = -\bar{B}LR\frac{d\alpha}{dt}10^{-8} = -E\alpha_0\cos mt \qquad [19]$$

also

$$0.4\pi S_a = -\delta\frac{dB}{dx} \qquad [20]$$

and

$$\zeta L\frac{d}{dx}(S_a) - \frac{2r_{kb}}{b_k}\int_0^x S_a\,dx = jmBL\,10^{-8} \qquad [21]$$

from which is obtained

$$\frac{d^2}{dx^2}(S_a) = \left[\frac{2r_{kb}}{b_k\zeta L} - jm\frac{0.4\pi}{\delta}\frac{10^{-8}}{\zeta}\right]S_a \qquad [22]$$

or

$$\frac{d^2}{dx^2}(S_a) - \beta^2(S_a) = 0 \qquad [23]$$

where

$$\beta^2 = \frac{2r_{kb}}{b_k\zeta L} - jm\frac{0.4\pi}{\delta}\frac{10^{-8}}{\zeta}$$

The solution of Equation [23] is

$$S_a = (S_a)_{b/2}\frac{\cosh \beta x}{\cosh \beta(b/2)} \qquad [24]$$

From this then

$$I_b/2 = \int_0^{b/2} S_a dx = \frac{b}{2}(S_a)_{b/2}\frac{tgh\beta^{b/2}}{\beta(b/2)} \qquad [25]$$

Note that for

$$\beta(b/2) > 3, \ tgh\ \beta(b/2) \cong 1$$

From the foregoing and Equation [19], we may write

$$jE\alpha_0 = (S_a)_{b/2}\zeta L + \frac{I_b}{2}(r_{kl} - jml_k)$$

$$= \frac{I_b}{2}\left[\zeta\frac{L}{(b/2)}\frac{\beta(b/2)}{tgh[\beta(b/2)]} + r_{kl} - jml_k\right]$$

$$= \frac{I_b}{2}\left[z\frac{\tau_s}{b/2}\frac{\beta(b/2)}{tgh\ \beta(b/2)} + r_{kl} - jml_k\right] \qquad [26]$$

or

$$I_b/2 = jE\alpha_0(j\theta + \gamma) = E\alpha_0(j\gamma - \theta) \qquad [26a]$$

The total coupling torque, i.e., for n poles, is equal to

$$T = n\bar{B}LI_bR(10^{-8}/9.81)$$

Upon substitution of the expression for I_b given in Equation [26a] we obtain

$$T = 2n\bar{B}LRE\alpha_0(j\gamma - \theta)(10^{-8}/9.81) \qquad [27]$$

The $j\gamma$ and θ components are proportional to $-d\alpha/dt$ and $-\alpha$, respectively.

Thus we have for the two torque components:

Damping component

$$c\frac{d\alpha}{dt} = c\alpha_0\, m\cos mt = 2n\bar{B}LRE\alpha_0\, j\gamma$$

elastic component

$$K\alpha = K\alpha_0 \sin mt = 2n\bar{B}LRE\alpha_0\,\theta \qquad [28]$$

Discussion

M. R. Lory.[5] Mr. Andriola is to be congratulated on his interesting and informative paper. The application of electric couplings on a large scale has come so recently in this country that little has been written about them. This paper is a valuable addition to the literature.

It is fortunate that the author has included some information from his associates at A.S.E.A. That company's actual experience in building couplings antedates our own by several years because geared-Diesel drive has been popular in Europe for some time, while its use on a large scale is quite recent here. However, we are now making rapid strides, as indicated by the fact that the writer's company alone now has more total horsepower in couplings built or under construction than have been built or are on order abroad, based on the latest published information.[6] We are now building the largest electric couplings in the world, rated 4375 hp at 180 rpm, for use with Sun-Doxford engines on four Maritime Commission CP-3 cargo and passenger vessels. Of the 62 motorships already built or on order on June 1, 1941, for the Maritime Commission, 38 have geared drives of which 30 are equipped with electric couplings and 8 with hydraulic couplings, while 24 are direct drive.

The principal difference between the electric couplings for these Maritime Commission ships and foreign-built couplings is in the amount of torque available at high slip for maneuvering. Figs. 3 and 4 of the paper show a coupling with 40 per cent torque at 100 per cent slip. The Maritime Commission engineers recognized that, if more torque than this were provided, the couplings could be used extensively as an aid in maneuvering. Consequently, their specifications required a minimum of 75 per cent torque up to 140 per cent slip.

The *Mormacpenn*, first of four C-3 cargo ships built by the Sun Shipbuilding and Drydock Company, and the first geared-Diesel ship to be completed under the Commission program, has electric couplings. Each of these ships has four Busch-Sulzer engines rated 2230 hp at 240 rpm, driving through Westinghouse electric couplings and Falk gears. The couplings on these ships have proved very satisfactory in service. The ships are exceptionally easy to maneuver. The engine and coupling controls are centralized in a control desk. The right-hand lever on the desk controls the operation of all four couplings. The usual procedure is to warm up two engines ahead and two astern at the "stand-by" signal. Then the operator can carry out any maneuver except "full ahead" or "full astern" by means of the coupling control and engine-speed levers. No starting air is consumed and the ease of operation is comparable to Diesel-electric drives. The writer has observed response to six bells in 1 min when docking.

The couplings on these vessels have about 100 per cent torque at high values of slip. This enables a crash-stop reversal to be made by disconnecting the engines from the propeller and reversing them at no load. The couplings are then energized and reverse the propeller while the engines run on fuel. This method of reversal is very fast and uses little starting air. The *Mormacpenn* was forced to make a crash stop in New York harbor to avoid a collision in a fog. The propeller was turning at higher than full speed astern in less than 1 min. While this time was shortened by the fact that the ship was not up to full speed ahead when the reversal was started, the quick reversal was credited with avoiding a crash. It is doubtful if a coupling with torque, as shown in the curves of the paper, would be able to reverse the propeller from full speed.

The writer was greatly interested in the mathematical analysis of the characteristics of the coupling which affect torsional vibration. When his company first studied electric couplings, some engineers recognized the torsional characteristics and worked up curves similar to the author's Fig. 5 for the Navy. Discussions of torsional characteristics led up to the extensive tests made by Mr. Dashefsky which were mentioned in the paper. It is impossible to compare the formulas in the paper exactly with those published[4] by the writer and others because differences in construction modify the analysis. The A.S.E.A. couplings are built of solid iron in the secondary core and the analysis must take care of eddy currents induced in the iron parts. We use laminated-iron cores and these eddy currents have negligible effect and were

[5] Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

[6] *A.S.E.A. Journal*, March, 1940.

neglected in the analysis. The curves obtained show that the two methods give similar results.

It is hoped that the amount of mathematical work done in determining the characteristics of electric couplings has not created the impression that torsional vibration presents a serious problem when they are used. Rather, the studies have been undertaken so that the designer may calculate the characteristics accurately, and predict the performance. When we know how a coupling acts, we can apply it properly. The large number in trouble-free active service constitutes the best proof that such couplings do protect the gears from torsional vibrations as well as aiding in maneuvering and performing all their other functions as discussed in the paper.

The question has come up regarding the heating of the couplings during maneuvering; for example the chief engineer on one Scandinavian vessel has said that he could not maneuver with the couplings because they overheated. It is not surprising that this happened with couplings which were not designed for this service.

As mentioned previously, the Maritime Commission engineers are largely responsible for the use of the couplings for maneuvering. For their ships, they specified sufficient torque to enable the couplings to perform this service, and also made sure that the couplings were adequate from a heating standpoint.

During the maneuvering, the couplings act as clutches. There is no mechanical contact between the two members, and hence there is no wear to cause maintenance. However, as in any clutch when the slip is high, energy must be dissipated. This energy appears as heat in the bars of the squirrel-cage winding. The time of operation at high slip is short, but the rate of heat generation is high, and little of the heat can flow out of the bars. A large portion of it has to be stored in the bars, and the only way to keep the temperature down is to provide a large mass of material to store the heat. For this reason, the bars must be made as large as possible.

This squirrel-cage winding is very rugged, since it consists of bars driven into slots in the core, and brazed with a high-temperature alloy at the ends to short-circuiting rings. There is no insulation to roast out, and the winding can stand high temperatures without injury. The most severe operating condition is during a reversal from full ahead when the couplings are disengaged while the engines are reversed and brought to about half speed astern. When the couplings are energized, they must bring the propeller to rest and then pull it up to the engine speed astern. They are designed to do this with moderate temperature rise, since margin must be provided to take care of any unusual operating conditions which might increase the heating.

During the trials of the eight Maritime Commission ships using electric couplings which have been completed, the couplings were often subjected to service several times worse than a normal reversal without injury. On the trials and the several trips in active service, the couplings have been subjected to every normal type of maneuver plus many abnormal ones without any damage of any kind from overheating.

Author's Closure

Mr. Lory's discussion constitutes an important addition to the material presented, especially as regards operating experience with couplings of this type.

Since presentation of the paper, we have had the opportunity to study closely the operation of two couplings in a twin-screw, direct-drive installation. Maintenance has been conspicuous by its absence, although these units have been in service 8 months in addition to a continuous 17-day shop test at 80 per cent power and 24 hours at full rating. While these couplings are not specifically designed to reverse the propeller from full speed, early ship-board tests soon established their adequacy for this maneuver; the units have been so operated since. It should be recognized, however, that in the interval between coupling disengagement and re-engagement, a reduction in propeller-shaft speed and necessary reversing torque will have taken place. The amount of reduction is a function of ship inertia and resistance.

Mr. Lory properly emphasizes the true importance of the vibration studies made for these couplings, and the author earnestly hopes that misinterpretation has not occurred as a result of the space devoted to this aspect of design.

Flexible Couplings for Internal-Combustion Engines

By J. Ormondroyd,[1] Ann Arbor, Mich.

Four typical dynamical cases of torsionally flexible "linear" couplings are examined: (1) Instantaneous applications of the maximum engine torque; (2) instantaneous stoppage of the engine or the driven member; (3) dangerous torsional resonance; and (4) tooth chatter in geared drives.

The basic purposes of any coupling are to tie together component parts of a rotating assembly and to transmit the operating torque safely between the parts. The wide diversity of coupling designs indicates that they are often expected to be more than mere concatenating links and transmitters of torque. Even when the component parts of the rotating assemblage are supposed to maintain fixed relative positions, the problem of alignment has forced the design of couplings with various degrees of freedom compatible with carrying out their basic functions. A large class of couplings embodying elastically flexible elements exists. These couplings are not only expected to concatenate component parts, transmit torque, and provide a certain amount of leeway in alignment, but they are also expected to provide a protection to the rotating system which would not exist if the flexibility were omitted.

The protection needed by the system is not always clear to the design engineer. In general, flexible couplings are useful in "dynamic" situations in which angular velocities are changing or in which the driving or delivered torques are variable. A second generalization, which cannot be emphasized too strongly, is that a flexible coupling is embodied in a complete rotating system, and its effects depend as much on the system characteristics as on its own properties. Such data as hub size, installation dimensions, and allowable horsepower per hundred revolutions per minute based on nominal load torque are necessary, but they are not sufficient to determine a successful application.

The effect of a flexible coupling is measured by the difference in operation with the added flexibility and the operation without it. The difference to be expected can often be predicted by dynamic analysis of the whole rotating system with and without the coupling.

Four typical dynamical situations in which torsionally flexible couplings are often considered will be examined. In all cases the coupling will be considered as "linear," that is, the angular deflection or twist between the driving and driven sides of the coupling will be proportional to the torque transmitted in a static test. The four cases are:

1 Instantaneous application of the maximum engine torque.
2 Instantaneous stoppage of the engine or the driven member.
3 Dangerous torsional resonance.
4 Tooth chatter in geared drives.

Instantaneously Applied Torque

This situation is an idealized limiting case of suddenly applied torque. It is worse than any actual case can be. If a coupling can be made to meet this situation safely, it will be more than adequate to meet any rapidly rising torque. To simplify the problem consider the engine as a single body of moment of inertia I_1 and the driven member as another body of moment of inertia I_2. The coupling and other connecting shafting has an over-all spring constant K. The suddenly applied engine torque is T. Friction torque and load torque can be ignored since the maximum distress in the connecting shafting will occur soon after the engine torque is applied and long before any load or friction torque can be developed. Let θ_1 be the motion of I_1 and ϕ be the angle of twist in the coupling members. Under these assumptions, Newton's second and third laws give the following equations of torque

$$I_1\ddot{\theta}_1 = K\phi + T \qquad [1]$$

$$I_2(\ddot{\theta}_1 + \ddot{\phi}) = -K\phi \qquad [2]$$

Eliminating θ_1, we get

$$\ddot{\phi} + K\frac{I_1 + I_2}{I_1 I_2}\phi = -\frac{T}{I_1} \qquad [3]$$

From this

$$\phi = A \sin pt + B \cos pt - \frac{T}{I_1 p^2} \qquad [4]$$

where

$$p^2 = K\frac{I_1 + I_2}{I_1 I_2}$$

and A and B are constants depending on initial conditions. When $t = 0$, $\phi = 0$, and $\frac{d\phi}{dt} = 0$, therefore

$$\phi = \frac{T}{I_1 p^2}(\cos pt - 1) \qquad [5]$$

The torque in the coupling connection is

$$K\phi = \frac{I_2}{I_1 + I_2} T (\cos pt - 1) \qquad [6]$$

The maximum absolute value of the torque in the coupling members occurs when $\cos pt = -1$. It is

$$M_{\max} = \frac{2T}{1 + \dfrac{I_1}{I_2}} \qquad [7]$$

The torque twisting the connecting members is independent of the spring constant of these members, depends on the ratio I_1/I_2, and can never be greater than $2T$.

Evidently, this is one dynamic problem in which a spring coupling offers no advantages. If the ordinary shafting which connects the two rotating members is made strong enough to stand twice the maximum possible torque that the engine can put out, everything has been done that can be done for this particular case.

[1] Professor of Engineering Mechanics, University of Michigan. Mem. A.S.M.E.

Contributed by the Oil and Gas Power Division and presented at the Annual Meeting, New York, N. Y., Dec. 2–6, 1940, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Instantaneous Stoppage of One Rotating Member

Using the same system as before, there are two cases, depending on which rotating member is stopped. In either case the entire kinetic energy of the free body must be stored in the elastic coupling members, if the energy lost in frictional dissipation is ignored. In these cases

$$(1/2)K\,\phi^2_{\max} = (1/2)I_1\omega^2 \text{ (if } I_2 \text{ is suddenly stopped)} \qquad [8]$$

or

$$(1/2)K\,\phi^2_{\max} = (1/2)I_2\omega^2 \text{ (if } I_1 \text{ is suddenly stopped)} \qquad [9]$$

In both equations ω is the angular velocity of the entire rotating system just before the sudden stoppage. In either case

$$\phi_{\max} = \frac{\omega}{\sqrt{\dfrac{K}{I_n}}} \qquad [10]$$

where n is either 1 or 2.

The maximum twisting torque in the coupling member is

$$M_{\max} = K\,\phi_{\max} = \sqrt{KI_n}\cdot\omega \qquad [11]$$

If M_0 is the maximum torque with the usual rigid coupling and M_1 is the maximum torque with the flexible coupling in place, and K_0 is the original spring constant of the coupling shafting and K_1 is the spring constant of the flexible coupling, then

$$\frac{M_1}{M_0} = \sqrt{\frac{K_1}{K_0}} \qquad [12]$$

Any reduction in twisting torque desired can be made by choosing K_1 small enough. Naturally the coupling parts must be strong enough to withstand successfully the twisting torque M.

This case is a particular instance of impact. If two absolutely rigid bodies collide, the impact force is infinite in magnitude. If a spring is interposed between the bodies, the forces between them become finite and controllable.

Torsional Resonance

This phenomenon has been treated extensively in technical literature and will not be discussed in detail here. The natural frequencies and "normal elastic curves" which can be calculated by using the Holzer method familiar to many engineers are fruitful guides in judging the usefulness of flexible couplings in particular cases. These data, usually determined for the case of no frictional damping, must often be supplemented by estimates of the effects of damping on the amplitudes and stresses which occur at resonance.

In applying a flexible coupling to a rotating system its effects on more than the lowest natural mode of vibration of the system must often be considered. If the operating speed of the engine-driven system is constant, a flexible coupling will often suffice to make operation at that speed free from dangerous amplitudes of motion. If the operating speed is variable, flexible couplings without damping devices are often "snares and delusions." There are some systems in which a properly designed flexible coupling will put the lowest natural frequency of the system below the idling speed, and the next higher mode of motion may have its resonance above the running range; but many actual systems, such as modern line airplane engines, are not so easily handled. The coupling may make one mode safe; but have no protective value at all in another mode of motion.

This problem presents an example of the influence of the entire rotating system on the operating characteristics of the flexible coupling.

Gear-Tooth Chatter

Consider an internal-combustion engine driving a centrifugal pump through a step-up gear. An eight-cylinder line engine with a maximum speed of 900 rpm will be considered. Since we are interested in general ideas, the system can be represented in the simplified form of three flywheels connected by two torsional springs. This system permits a general survey of the possibilities of gear-tooth chatter at resonance in the one-noded mode of motion. It would not serve well to investigate the possibilities in the two-noded mode of motion.

Let I_1 = the moment of inertia of the engine rotating parts
I_2 = the moment of inertia of the flywheel
I_3 = the combined moment of inertia of the gearing and centrifugal pump (all reduced to the engine-shaft speed)
K_1 = the spring constant of the engine shaft
K_2 = the spring constant of the connecting elements between the flywheel and the slow-speed gear. This will include any flexible coupling which may be used.

The calculations made assume that the gear teeth remain in contact on one side even during resonance vibrations. In fact this is a necessary state of affairs if the gear is to operate quietly. It can be realized in practice if the maximum inertia torque of the pump and pinion never exceed the load torque transmitted through the gearing to the pump.

It is proposed to make a preliminary survey of the vibration and gear-chatter possibilities of this system by merely calculating the frequency and "normal elastic curve" without damping, using the usual Holzer method. The analysis proposed does not give the actual amplitudes of motion which will be encountered in operation at resonance. It merely gives an idea of the best results which can be attained by varying the parts of the system. The best that can be done by varying the flexibilities and moments of inertia may not be good enough, in which case special devices for ameliorating the vibration conditions may have to be introduced into the system. However, the operating conditions in many systems have been made safe by simply varying the flexible and inertia elements.

Two major assumptions are made in the following calculations. They are:

1 The damping in the system is unknown; but it is assumed to remain at or near the same value no matter what changes are made in the system.

2 For the major critical speeds in the lowest mode of motion the energy input per cycle is assumed to be proportional to the average of the relative amplitudes of motion of the engine I_1 and the flywheel I_2.

Based on these two assumptions the actual amplitudes encountered will be proportional to the average of the relative amplitudes of the engine and flywheel in any given combination considered.

For the simplified system considered the single-noded frequency $f = p/2\pi$ can be calculated from

$$p^2 = (1/2)\left(\frac{K_1}{I_1}+\frac{K_1}{I_2}+\frac{K_2}{I_2}+\frac{K_2}{I_3}\right) - \sqrt{(1/4)\left(\frac{K_1}{I_1}+\frac{K_1}{I_2}+\frac{K_2}{I_2}+\frac{K_2}{I_3}\right)^2 - \frac{K_1K_2}{I_1I_2}\cdot\frac{I_1+I_2+I_3}{I_3}} \qquad [13]$$

and the relative amplitudes of motion for I_1, I_2, and I_3 are respectively

$$\frac{A_1}{A_1} = 1.00 \qquad [14]$$

$$\frac{A_2}{A_1} = 1 - \frac{p^2}{K_1/I_1} \quad \text{[15]}$$

$$\frac{A_3}{A_1} = \frac{1 - \dfrac{p^2}{K_1/I_1}}{1 - \dfrac{p^2}{K_2/I_3}} \quad \text{[16]}$$

To make the problem concrete assume that the system is originally laid out so that $I_1 = 60$ lb in. sec²; $I_2 = 200$ lb in. sec²; $I_3 = 160$ lb in. sec², $K_1 = 5 \times 10^6$ lb in. per radian; and $K_2 = 30 \times 10^6$ lb in. per radian. Substituting these values in the equations it is found that the one-noded, fourth-order critical speed will occur near 725 rpm and that the node (or point of maximum shear stress) lies in the crankshaft near the flywheel. Since the engine might operate at this speed this information is quite disturbing.

By introducing a flexible coupling between the flywheel and the gear, it may be possible to place the critical speed at a low running speed and also the point of maximum twist may be shifted from the crankshaft into the coupling where damage will be less expensive and more easily repaired. The effect of varying K_2 by introducing additional flexibility is seen in Fig. 1.

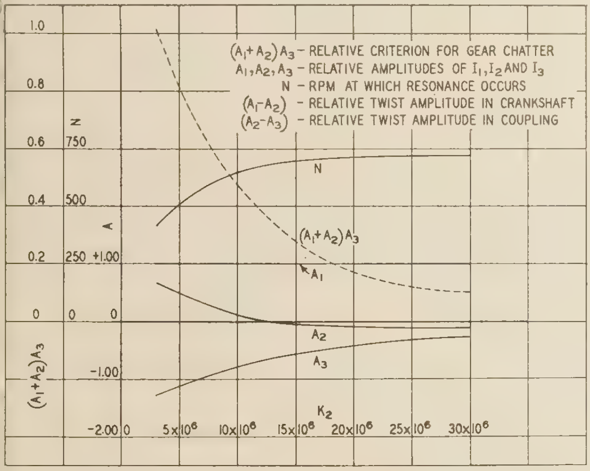

Fig. 1 Effects of Varying K_2, I_2 = 200 Lb In. Sec²

By reducing K_2 to one tenth of its original value it is possible to depress the critical speed down to 400 rpm and to displace the largest twist from the crankshaft into the coupling. The actual twist in the crankshaft is also reduced to about one half of its original value.

This last conclusion follows from the fact that the actual amplitudes and twists are proportional to $\frac{A_1 + A_2}{2}$. The relative twist in the crankshaft is $A_1 - A_2$. The actual twist will then be proportional to $\frac{A_1 + A_2}{2} \cdot (A_1 - A_2)$, or to $\frac{A_1^2 - A_2^2}{2}$. For two different values K_2 and K_2' the relationship between actual twists will be

$$\frac{1 - \dfrac{(A_2')^2}{(A_1')^2}}{1 - \dfrac{A_2^2}{A_1^2}}$$

For $K_2 = 30 \times 10^6$ lb in. per radian and $K_2' = 3 \times 10^6$ lb in. per radian, this ratio is approximately one half.

The data from Equations [13] to [16] also permit an estimate of the possibility for tooth chatter. If the actual moment of inertia of the centrifugal pump and its pinion is I_3', its equivalent value reduced to gear speed is n^2I_3', where n is the gear speed-up ratio. The maximum inertia torque of the pump and pinion, referred to the gear, is $n^2I_3'p^2A_3$. Since the actual value of A_3 is proportional to $\frac{A_1 + A_2}{2}$, the actual value of the inertia torque is proportional to $\frac{n^2I_3'p^2A_3(A_1 + A_2)}{2}$. The load torque transmitted through the gear is the pump-load torque. For a centrifugal pump this torque is $T = CN^2$, where N is the engine rpm

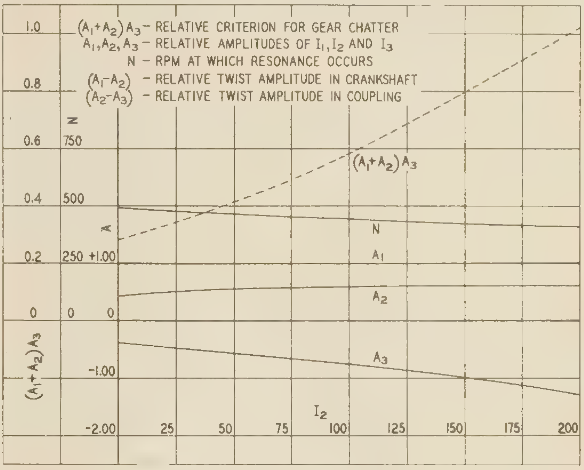

Fig. 2 Effects of Varying I_2, $K_2 = 3 \times 10^6$ Lb In. per Radian

and C is a constant depending on the dimensions of the pump. For quiet operation at resonance $CN^2 > \frac{n^2I_3'p^2A_3(A_1 + A_2)}{2}$ or $\frac{n^2I_3'p^2A_3(A_1 + A_2)}{2CN^2} < 1$. Since at resonance with the fourth-order harmonic torque $N^2 = \frac{900}{16}p^2$, then

$$\frac{8}{900} \cdot \frac{n^2I_3'}{C} A_3(A_1 + A_2) < 1$$

Because n^2, I_3', and C are independent of changes in K_2, it can be seen that the tendency for the gears to chatter is proportional to $A_3(A_1 + A_2)$.

From Fig. 1 it can be deduced that this quantity $A_3(A_1 + A_2)$ has increased 10 to 1 as K_2 is reduced from 30×10^6 lb in. per radian to 3×10^6 lb in. per radian. While this rough analysis does not indicate whether the gear teeth will actually chatter, it does emphasize the fact that tooth chatter is far more likely to occur with the flexible coupling than without it. While the introduction of the coupling has reduced the stress in the crankshaft and lowered the critical speed to a range in which the engine is less likely to run, it has increased the probability of gear noise at this lowered critical speed enormously. Evidently, the introduction of additional flexibility without some other change in the system may, in this case, merely change the problems encountered in the design.

What other characteristic of the system can be changed so that the favorable changes introduced by the coupling can be maintained while the unfavorable developments can be minimized? Naturally, the engine, gears, and pump are not easily

modified. The flywheel is a member which can be modified without undue practical difficulty. Fig. 2 indicates the change of conditions brought about by varying I_2 from 200 lb in. sec² to 20 lb in. sec², keeping K_2 constant at 3×10^6 lb in. radian. For $I_2 = 20$ lb in. sec² the value of N is not raised much nor is the actual twist in the engine shaft changed much; but the value of $(A_1 + A_2)A_3$ is reduced to one third of its value for $I_2 = 200$ lb in. sec². While this value is still three times greater than it was in the original design, it is a large change in the right direction.

A further change which could be investigated would be the introduction of an additional flywheel between the flexible coupling and the slow-speed gear. This would reduce N, the actual twist in the crankshaft, and $(A_1 + A_2)A_3$, all steps in the right direction. This additional flywheel, if used at all, should not be introduced on the high-speed side of the gearing since it would then increase I_3' and thereby increase the tendency to produce gear chatter.

After these three things have been done, all steps which are easy to take have been taken. If, after all this, the system actually has amplitudes of vibration so large that dangerous stresses exist in the crankshaft or coupling and gear chatter develops, then the designer is really confronted with a difficult problem. Devices especially designed to reduce vibration would have to be introduced into the system, and this represents a major problem after the system is put into operation.

This last case was discussed in some detail to indicate that successful coupling application is only possible through complete dynamic analysis of the entire rotating system. It should be remembered that only the effects on the lowest mode of vibration have been investigated. In many cases the second and third modes of motion might have to be analyzed in the same manner in order to insure safe or noiseless operation.

Nonlinear Couplings

It is a popular misconception that flexible couplings which have torque-deflection curves that are not straight lines are cure-alls for torsional-vibration troubles. It is often imagined that torsional resonance cannot occur if such a coupling is introduced into the rotating systems. This belief may be based on the statements made by recognized authorities that no infinite amplitudes of motion are possible in a system which contains a nonlinear coupling, even if frictional damping were completely absent. While this is true, and it is also a fact that very complicated relationships exist between torque, frequency, and amplitude of motion, it should be understood that conditions resembling resonance with linear couplings also exist with nonlinear couplings. Amplitudes of motion large enough to cause trouble can exist at certain frequencies even if nonlinear couplings are used. The reader is referred to a paper entitled "Steady Oscillations of Systems With Nonlinear and Unsymmetrical Elasticity," by Manfred Rauscher, Trans. A.S.M.E., vol. 60, 1938, p. A-169. This paper indicates methods by which such couplings can be analyzed and also refers to numerous other papers on this subject that could be perused to get a complete picture of the situation existing when nonlinear couplings are used.

Coupling Strength and Safety

In this paper only the effects of the elastic properties of the coupling have been considered. The ability of the flexible elements to withstand the twisting torques encountered in operation has been completely omitted. A great variety of couplings could be used to get the same flexibility. Each one considered would have to be analyzed to ascertain its adequacy to meet the operating conditions at resonance. If it is strong enough to take the torques at resonance safely, it is more than safe at all other operating speeds. The most general remark that can be made in this connection is that safety in a flexible coupling is to be attained by using the largest possible volume of elastic material which gives the desired spring constant in the space available for the coupling. Also, the most efficient use of elastic materials in couplings is gotten by stressing the materials in pure tension, pure compression, or pure shear. This is usually only practical in couplings in which rubber is the elastic medium. Where metals form the elastic elements reasonable deflections are gotten only by using the material in twist or bending. Under these modes of stressing a fair percentage of the metal is not carrying large stresses. Under these conditions either very high fatigue limits must be used or volumes of metal hard to pack into reasonable space limitations must be considered.

Discussion

E. L. Davis.[2] Referring to case 1, of the paper, the instantaneous application of maximum engine torque, the formula derivation shown in Equations [1] through [7] is considered as a theoretical problem correctly derived but impractically used. This problem is intended to represent an engine driving some machine. It is believed incorrect to assume a single disk as representing the reciprocating-and-rotating-mass system of the entire engine, when a solid coupling connects the engine and driven machine. On the other hand, it is acceptable to consider the case, as originally intended, when there is an abundance of flexibility produced through the medium of a flexible coupling. A typical practical problem of this nature was calculated by the author in a previous article.[3] In the case of a solid coupling, the node was between the last cylinder and the engine flywheel, whereas, in the case of a flexible coupling the node was in the coupling hub mounted on the engine shaft.

A close approximate derivation can be made for solid couplings only by using Equations [1] to [5], inclusive. By placing the node in the mass I_1, we have $p^2 = \dfrac{K}{I_2}$ instead of $K\dfrac{I_1 + I_2}{I_1 I_2}$. Then $\phi = \dfrac{T}{I_1 \dfrac{K}{I_2}}(\cos pt - 1)$ and Equation [6] becomes $K\phi = \dfrac{T}{I_1/I_2}$ $(\cos pt - 1)$ and Equation [7] becomes $M_{\max} = \dfrac{2T}{I_1/I_2}$ instead of $\dfrac{2T}{1 + \dfrac{I_1}{I_2}}$.

The formula $M_{\max} = \dfrac{2T}{I_1/I_2}$ can be used for solid couplings, and the formula $M_{\max} = \dfrac{2T}{1 + \dfrac{I_1}{I_2}}$ can be used for flexible couplings.

In comparing the problems given in the author's previous article,[3] the value $\dfrac{I_1}{I_2} = 0.846$ for both solid- and flexible-coupling problems. When using Equation [7] as revised and as shown, we have values of $M_{\max}$ as $1.082T$ and $2.46T$, respectively. This means that the torque in the shaft for a solid coupling is 2.27 times that for a flexible coupling. It also shows that the maximum stress in the case of the solid coupling is in the crankshaft, while the maximum stress in the case of a flexible coupling is in the coupling.

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[2] Analyst, The Falk Corporation, Milwaukee, Wis.

[3] "Problem of Torsional Vibration Increases With Engine Power," by J. Ormondroyd, *Machine Design*, vol. 3, June, 1931, pp. 37–40.

Other values of $\frac{I_1}{I_2}$ and their respective shaft torques are shown in the following table:

$\frac{I_1}{I_2}$	1	0.846	$\frac{1}{2}$	0
M_{max} for flexible coupling	T	$1.082T$	$1\frac{1}{3}T$	$2T$
M_{max} for solid coupling	$2T$	$2.46T$	$4T$	∞T

Referring to the author's case 4, tooth chatter in gear drives, it is pointed out in the concrete example given that, in the case of a solid coupling, the major critical fourth-order speed occurs at 725 rpm and, in the case of the flexible coupling, the minor second-order critical is at 400 rpm. Inasmuch as the operating range of this drive is from 700 to 900 rpm and the lowest possible running range would be 500 rpm, it is evident that the use of a flexible coupling here is advantageous. Fig. 1 of the paper could be slightly revised by drawing two horizontal lines at 700 and 900 rpm to show the operating-speed range for gear-chatter comparison.

In regard to the equations on the third page of the paper, it is found that π^2 has been omitted. These equations should read as follows

$$N^2 = \frac{900}{\pi^2 16} p^2$$

and

$$\frac{\pi^2 8}{900} \cdot \frac{n^2 I'_3}{C} A_3(A_1 + A_2) < 1$$

W. P. Schmitter.[4] *Reduction of Dynamic Loads.* The author has reduced four typical dynamical situations to relatively simple expressions. Many practical cases cannot be so readily analyzed. Take, for instance, the rather complex example of an engine-driven system containing a gear train with known tooth-spacing errors. The magnitude of the dynamic loading of the gears will depend, among other things, upon the rigidity of the system. Resilient couplings on both sides of the gears will permit a greater degree of acceleration and deceleration of the gear masses in response to the errors, thus, not only localizing their effects, but making for materially lower tooth stresses than with nonresilient couplings. The solution for any given case may be obtained by following the methods developed in a bulletin[5] published by the Society.

Impact Loading From Driven Machine. Practical examples of case 2 are seen in systems in which sudden load decelerations of the driven machine take place, thus requiring considerable energy to be absorbed. There are numerous cases of the use of resilient couplings to alleviate bad situations in severe rolling-mill and similar drives. Where all the factors are known, the relative stresses can be computed.

Torsional Resonance. Severe resonance may frequently be avoided by application of nonlinear resilient couplings, because the tuning changes with the increased amplitude. We agree that complete analysis of the rotating system is necessary in order to avoid unfavorable situations. No flexible coupling can be expected to operate satisfactorily in a bad critical.

We consider damping a most important property of the "Steelflex" (Bibby) coupling of the writer's company. The hysteresis loop obtained in static testing is due to its characteristic design. This is further increased dynamically by the action upon the grease within the sealed enclosure.

[4] Chief Engineer, The Falk Corporation, Milwaukee, Wis.

[5] "Dynamic Loads on Gear Teeth," A.S.M.E. Report of Special Research Committee on Strength of Gear Teeth, 1931.

Gear-Tooth Chatter. Resilient couplings of this type may be relied upon to avoid certain types of tooth chatter. In the rotary-drum crushing field, herringbone pinions had formerly to be shrouded to avoid their axial displacement during the break in tooth contact from effects of the cascading material. The shrouds are eliminated whenever the couplings mentioned are applied because the potential energy stored is sufficient to make the pinion follow the gear when rapid speed fluctuations occur.

Coupling Strength and Safety. It is exceedingly difficult to draw any arbitrary conclusions with respect to the most efficacious use of materials in flexible couplings since so much depends on design, as the following analysis will demonstrate. Maximum strength in an articulated double-acting coupling is obtained when the shear strength of the individual interlocking elements is equal. If a nonlinear resilient coupling of the type previously mentioned is designed so that the grid is in shear at the limit-load peak, its strength at that point is equal to that of the nonresilient coupling. At lower torques, the grid transmits the load through a continuous variable-span beam. The entire grid material including loops is constantly under stress, thus the resilience of the coupling is unusually high, despite comparable size and safety factors.

A. M. Wahl.[6] For some time past, the writer has been interested in the design of couplings for induction-motor drives,[7] particularly those subject to frequent starts and stops as is the case, for example, in the roll-table drive used in continuous strip mills. Because of the electrical characteristics of the induction motor, such systems on starting are subject to a suddenly applied pulsating torque at the line frequency which dies out after a time. This type of transient-torque application gives rise to two effects, i.e. (1) if the natural frequency of the drive approaches the line frequency, a resonant condition will be present, and (2) because of the sudden application of torque an impact effect occurs, which is augmented by the nonlinear characteristic of the usual coupling. In certain practical applications, considerable trouble has arisen from these causes. In cases where motors are started and stopped frequently, this problem is of particular importance, since a sufficient number of cycles of stress may take place eventually to cause failure of the mechanical parts of the system.

The writer wonders whether or not a similar condition may not be present during the starting of an internal-combustion engine, coupled to its load by means of a flexible coupling. In such a case, a suddenly applied pulsating torque, set up as a consequence of the explosions of individual cylinders, would be present. For the nonlinear coupling usually applied in such cases, such a torque might give rise to stresses of considerable magnitude as a consequence of impact effects. In addition, because of the pulsating torque, due to the explosions of individual cylinders, it would appear that there is a possibility of increased torque due to resonance for certain values of the natural frequency of the system. It is realized that such conditions probably occur but rarely in practice, however, in cases where such systems are started and stopped frequently an analysis of such torques might be worth-while.

The writer agrees with the author that conditions resembling resonance may occur even with nonlinear couplings. Such conditions have, in fact, been observed in tests on induction-motor drives, the torque being measured by means of a magnetic torsi-

[6] Mechanics Department, Westinghouse Research Laboratories, East Pittsburgh, Pa. Mem. A.S.M.E.

[7] "Transient Torques in Induction-Motor Drives," by A. M. Wahl, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 63, 1941, pp. A-17–A-22.

ometer. In certain cases due to such effects torques up to about 8 times the nominal torque have been recorded.

Author's Closure

Mr. Davis is under a grave misapprehension when he considers that a solid coupling necessarily puts the node (in the single-noded mode of motion) in I_1. The node will never be there with either solid or flexible coupling unless $I_1 = \infty$. When $I_1 = \infty$ Equations [1] and [2] no longer describe the torques in the system. In Equation [1] of the paper the torque T acts on I_1. If $I_1 = \infty$ no finite value of T will move it. In this case ϕ would be zero at all times and the twist torque in the connecting shaft would always be zero. This checks Equation [7] for the case in which I_1/I_2 approaches infinity.

If torque T is suddenly applied to I_2 with the flexible shaft (including either the solid or flexible coupling) built into a rigid, infinite body I_1 the maximum torque in the shaft and coupling will never exceed $2T$. The conclusions drawn by Mr. Davis on the assumption of a node in I_1 are therefore erroneous.

Fig. 1 and the discussion from which it arises were intended only to indicate trends toward the possibility of tooth chatter. No lower limit to the operating range of the engine was mentioned. As a matter of fact the numerical data used in the discussion were taken from an actual installation in which tooth chatter did develop at low (but actual) operating speeds. The actual installation had the tooth chatter removed at all operating speeds by reducing I_2 (the engine flywheel) to the smallest practical value.

The equations on the third page of the paper should contain π^2 as pointed out by Mr. Davis. The omission of π^2 makes no change in the discussion of trends since none of the constants in the last equation on page three was used in plotting the curves. The factor A_3 ($A_1 + A_2$) is all that appears in the discussion or the curves.

The author has no comment to make on most of Mr. Schmitter's discussion but would like to reiterate most strongly that nonlinear couplings are not always a cure for resonance. Speeds at which excessive amplitudes of motion developed in systems containing nonlinear couplings have been observed by the author and many other engineers. These speeds can be predicted if the torsional characteristics of the coupling and the system are known.

The author has never found damping which naturally exists in any coupling to be very efficacious in reducing amplitudes at resonance. Coupling damping which is effective must usually be achieved by deliberate design. Large damping is seldom encountered in mechanical designs merely as an accidental by-product.

The author has seldom run into torque problems arising from the starting or stopping of internal-combustion engines. Such engines run through an infinity of torsional critical speeds in starting. Practically all of these critical speeds resonate with such small harmonics that they are not even detected. Even a potentially dangerous critical speed can be passed through with safety if the engine has enough torque to accelerate rapidly. The author has seen two installations in which the engine characteristics were such that the engines lingered in bad criticals on the way up in speed. The amplitudes of motion built up to considerable value in both cases before the engine governors could feed enough oil to push on past the energy-absorbing vibration.

It is interesting to note that Mr. Wahl has observed conditions similar to resonance in a system containing a nonlinear coupling.

Combustion Explosions in Pressure Vessels Protected With Rupture Disks

By MERL D. CREECH,[1] OKLAHOMA CITY, OKLA.

All too frequently combustion explosions occur in industrial pressure vessels, particularly compressed-air receivers, resulting in loss of life and property damage. The research described in this paper is the first step in determining means to prevent this loss of life and property damage when combustion explosions do occur in pressure vessels by protecting the vessels from excessive overpressure with suitable frangible rupture disks.

Nomenclature

The following nomenclature is used in connection with tables appearing in the text of this paper:

P_s = rupture-disk bursting pressure, psi
P_1 = initial compression pressure, psi abs. This is combined partial pressure of propane vapor and compressed air, comprising explosive mixture
P_2 = maximum explosion pressure in vessel, psi abs
w_a/w_p = ratio of weight of air to weight of propane vapor in explosive mixture
P_P = partial pressure of propane vapor in explosive mixture, in. mercury
$(dp/dt)_{avg}$ = average rate of pressure rise, psi per sec
$(dp/dt)_{max}$ = maximum rate of pressure rise, psi per sec

Methods of Protecting Pressure Vessels

In many industrial applications a pressure vessel is used for the storage of an explosive mixture of compressed air and some combustible vapor. Each of these installations represents a hazard to life and property. A common example is air receivers. Although compressed air is not in itself explosive, the introduction of oil into the receiver either from a defective compressor or by faulty operation does create a condition responsible for explosions, resulting in great loss of life and much property damage every year.

Even though these vessels are always provided with relief valves to safeguard them from overpressure, they are not protected from the very rapid pressure rise during a combustion explosion of their contents. Since the relief valve does not protect them and it is obviously difficult to prevent an occasional accidental explosion, it seemed worth-while to investigate the ability of a rupture disk to relieve harmlessly the explosion pressure.

The experimental work which had been done previously on combustion explosions was investigated. It was found that considerable study had been given to combustion explosions in small bombs, while little work had been done using larger closed vessels. Nothing had been done using a relatively large vessel protected by a rupture disk.

Propane and compressed air constituted the explosive mixture chosen. This particular combination was selected not because it would be exactly similar to any explosive mixture likely to be encountered in practice but because propane is easy to obtain, its properties are well known, and it is readily mixed with air to form an explosive mixture. In this way the behavior of a rupture disk when used to relieve a combustion explosion could be studied. If the preliminary results proved to be favorable, a study of more specific types of explosive mixtures could be undertaken later. As will be evident, the results were promising and it is intended to study in more detail specific examples of explosive mixtures such as are actually encountered in industrial applications.

Apparatus for Studying Explosive Mixtures

Fig. 1 is a scale drawing of the experimental apparatus used. It consists of an explosion vessel mounted on a concrete foundation 8 ft in diam and 3 ft thick to take the recoil when the rupture disk bursts. The foundation is 5 ft below the ground level. Surrounding the vessel is a steel shell 8 ft in diam and 10 ft high. The 5-ft section of the shell projecting above the ground is surrounded by a concentric steel shell 14 ft in diam; the space between the two is filled with earth. This is a safety measure should the vessel burst during an explosion.

The vessel itself is a specially constructed arc-welded and radiographed pressure vessel of $1^5/_8$-in-thick high-tensile-strength steel, 24 in. inside diam × 10 ft high. It was designed for a safe working pressure of 2000 psi. The bottom of the vessel has an elliptical head welded to the shell. The top consists of a special flange arranged to have various-size rupture disks bolted to it.

The vessel has a stuffing box installed to allow for the operation of a fan in the vessel to promote turbulence and create an intimate mixture of the explosive vapor and the compressed air. In addition, there is a connection for admitting compressed air and propane, a connection for the pressure gage and manometer, threaded openings for three spark plugs, and threaded openings for the three pressure-recording indicators.

The ignition of the explosive mixture in the vessel is by means of a specially constructed spark plug using a 30-a fuse strip, connected across a 110-v a-c circuit. When the fuse melts, the arc formed ignites the explosive mixture in the vessel.

The explosion pressure was recorded by three high-speed engine indicators. The drums of the indicators were driven by a synchronous motor. The drum speed was measured and found to be 23.5 in. per sec.

To synchronize the drums and locate on the diagrams the exact instant the disk ruptured, each drum was equipped with a stylus on a bell crank. The bell crank on each drum was connected to a $^3/_{16}$-in-diam steel rod running vertically parallel to the axis of the explosion vessel. This rod was connected by means of another bell crank to a fine steel wire tightly stretched over and approximately 6 in. above the rupture disk. When the disk burst, the wire caused the rod to move upward approximately $^1/_2$ in., actuating the three bell cranks on the indicator drums. Thus the three indicator diagrams were marked simultaneously at the instant the rupture disk burst.

Test Procedure

The fuel was liquefied propane, such as is sold commercially for domestic gas appliances. The drum of fuel was connected in

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

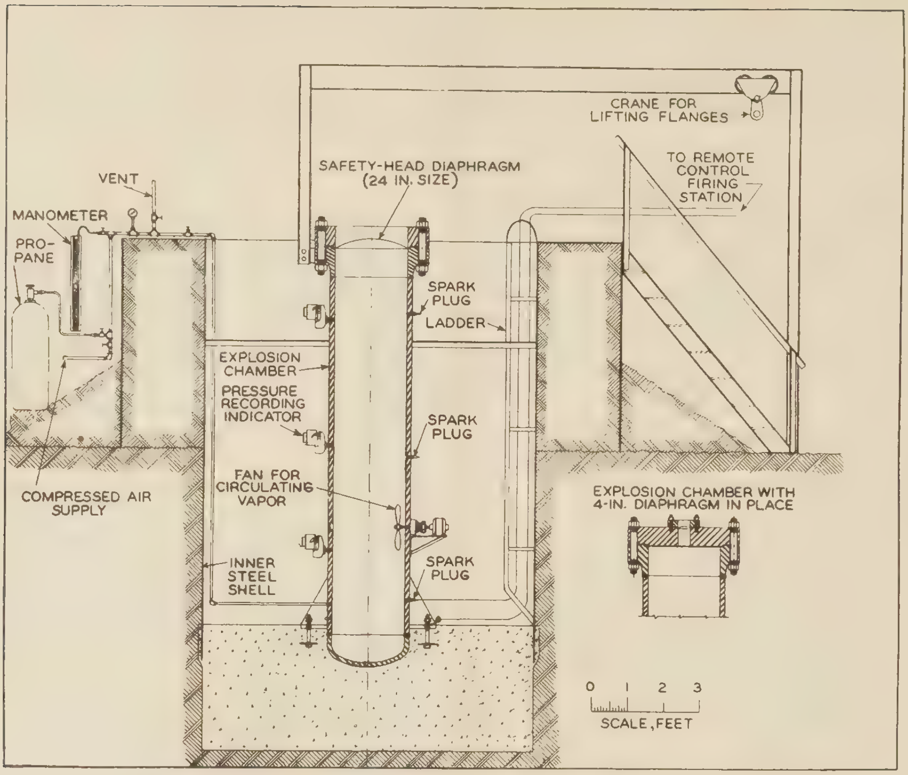

Fig. 1 Arrangement of Experimental Explosion Vessel

a manner to permit the propane to vaporize and flow into the explosion vessel. The amount of fuel used in any explosion was determined by measuring the partial pressure of the vaporized propane in the explosion vessel with a mercury manometer. After the required amount of propane had been admitted, the vessel was charged with compressed air to the preselected initial compression pressure.

The fuel used was assumed to be pure C_3H_8. This was probably not strictly true but small amounts of lighter and heavier fractions would not alter its characteristics appreciably. It was also assumed that the vapor pressure and specific volume could be calculated by applying the ideal-gas equation of state, $Pv = wRT$, where R is equal to 35.1 lb. Furthermore, in all calculations to determine the fuel-air ratio the barometric pressure was assumed to be 15 psi.

Each explosion test was conducted as follows:

After the desired rupture disk had been bolted to the top of the vessel, the paper was fastened to the indicator drums and the synchronizing mechanism adjusted. Next, the spark plug was screwed into the vessel. Then fuel was admitted until the proper partial pressure was registered by the manometer. The valve between the manometer and the vessel was closed and compressed air admitted until the proper pressure, as shown by the pressure gage, was attained. The valve between the vessel and the pressure gage was closed. Next, after walking to the remote-control station approximately 100 yd away, the fan in the vessel was started and allowed to run for 1 min. Then the fan was stopped, the indicators started, and the ignition switch closed. After the explosion, the indicators were stopped and the ignition switch opened.

The paper was removed from the indicator drums and the vessel purged of its burned gases by blowing compressed air in at the bottom of the vessel. In cases where the 1500-psi rupture disk was used and did not rupture, it was removed and the burned gases purged as before.

In some of the tests, three indicators were used. In others only one indicator was used. In all of the indicator diagrams reproduced in this paper, where three indicators were used, the diagrams are arranged one above the other; the upper diagram in the reproduction being the diagram from the indicator located at the top of the vessel, the middle diagram being the diagram from the indicator at the middle of the vessel, and the lower diagram being from the indicator located at the bottom of the vessel. Wherever possible, the three diagrams are reproduced so that a vertical line will intersect all three explosion lines at the same instant of time. Time is measured from the instant the disk ruptured. Where three indicators were used and the disk did not rupture, there was no way of locating the zero-time line on the diagrams.

In all of the diagrams there are two parallel horizontal lines. The lower line is the atmospheric-pressure line and the upper line is the initial compression-pressure line.

A series of tests was conducted to determine the effect of varying the air-fuel ratio. Each of these tests was conducted using a 4-in-diam 1500-psi-bursting-pressure rupture disk. Since the 1500-psi bursting pressure was much higher than any explosion pressure encountered, this converted the explosion vessel into a closed bomb without any relief. A pressure of 45 psi gage was used as the combined pressure of the propane and compressed air in each of these explosions. The amount of propane in the explosive mixture was varied from approximately the lower explosive limit nearly to the upper explosive limit.

Results of Tests

The data taken from the indicator diagrams for these tests are given in Table 1. In Fig. 2 is plotted the relation between the

TABLE 1 DATA TAKEN FROM INDICATOR DIAGRAMS DURING TESTS TO DETERMINE EFFECT OF VARYING AIR-FUEL RATIO

Test no.	P_1	P_2	P_2/P_1	$(dp/dt)_{avg}$	$(dp/dt)_{max}$	P_P	w_a/w_p
13	60	290	4.84	315	638	4	20.5
19	60	390	6.50	620	862	5	15.5
14	60	435	7.24	715	1785	5	15.5
40	60	495	8.25	1290	1850	6	12.9
21	60	385	6.42	596	1830	6	12.9
15	60	425	7.01	804	1355	6	12.9
26	60	495	8.25	1455	1860	6	12.9
29	60	495	8.25	1660	1860	6	12.9
22	60	395	6.58	484	1110	7	10.9
16	60	405	6.67	954	1110	7	10.9
17	60	415	6.92	978	*	8	9.4
18	60	320	5.33	793	*	9	8.3
20	60	320	5.33	654	*	9	8.3

* Maximum rate of pressure rise for these diagrams would be misleading.

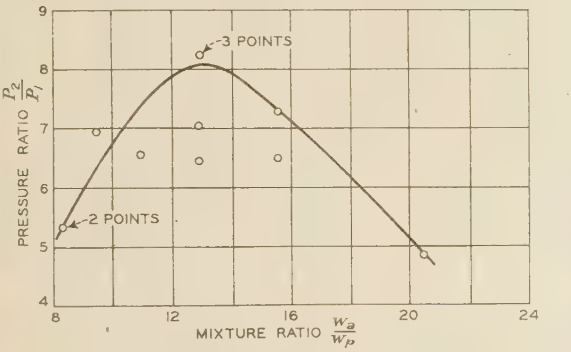

FIG. 2 CURVE SHOWING EFFECT OF AIR-FUEL RATIO ON EXPLOSION PRESSURE

P_1 = initial compression pressure
P_2 = maximum explosion pressure
w_a = weight of air in mixture
w_p = weight of propane vapor in mixture

air-fuel ratio and the ratio of the maximum explosion pressure to the initial compression pressure.

Examination of the indicator diagrams reveals that the ratio of maximum explosion pressure to the initial compression pressure was greatest when a mixture of 12.8 parts of air to 1 part propane by weight was used. This is shown clearly in Fig. 2. The rate of pressure rise was also found to be a maximum for this mixture. For the leaner mixtures, the indicator diagrams were similar to Fig. 3 (test No. 15). For the richer mixtures, Fig. 4 (test No. 18) and Fig. 5 (test No. 29) may be considered typical diagrams.

The vibrations or pressure waves recorded on the diagrams in Figs. 4 and 5 are to be found on all diagrams for the richer mixtures and are never found on the diagrams for the leaner mixtures. The frequency of these pressure waves is approximately 117 per sec. They appear to be the sum of several vibrations of differing frequencies. The small ripples are of approximately the same period as the natural frequency of the indicator recording mechanism. Another characteristic is that the amplitude of these vibrations is a maximum at the point of maximum explosion pressure. Yet another characteristic is that the vibrations are very similar for the diagrams taken at the top and bottom of the vessel. However, for the diagram taken at the middle of the vessel, their amplitude is very much less and their frequency is not so well defined. This is well illustrated in Fig. 5.

Since the air-fuel ratio of 12.8 to 1 gave the highest explosion pressure and the maximum rate of pressure rise, it was used in all subsequent experiments.

Having selected the air-fuel ratio to be used, a series of tests based on this air-fuel ratio was made with varying initial compression pressures. These explosions, too, were made without any relief, using the vessel as a closed bomb. The data from the

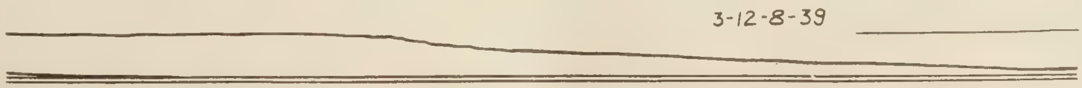

FIG. 3 TEST NO. 15: TYPICAL INDICATOR DIAGRAM FOR CLOSED VESSEL WITHOUT RELIEF; RUPTURE DISK DID NOT BURST
(For this test the spark plug was at the bottom of the explosion vessel and the indicator at the top. The partial pressure of the propane vapor was 6 in. of mercury and total initial pressure was 60 psi abs.)

FIG. 4 TEST NO. 18: TYPICAL INDICATOR DIAGRAM SHOWING PRESSURE WAVES UNDER CERTAIN CONDITIONS WHEN RUPTURE DISK DID NOT BURST
(For this test the spark plug was at the bottom of the vessel and the indicator at the top. The partial pressure of the propane vapor was 9 in. of mercury and the total initial pressure was 60 psi abs.)

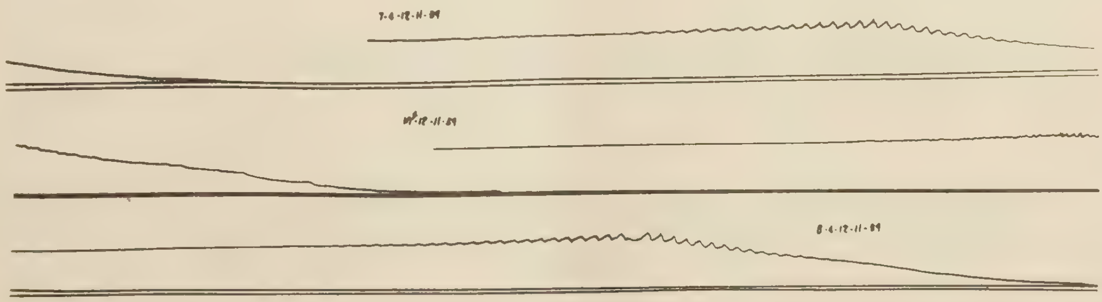

FIG. 5 TEST NO. 29: THREE SIMULTANEOUS INDICATOR DIAGRAMS FOR CLOSED VESSEL WITHOUT RELIEF
For this test the spark plug was at the bottom. The partial pressure of the propane was 6 in. of mercury; the total initial pressure was 60 psi abs.)

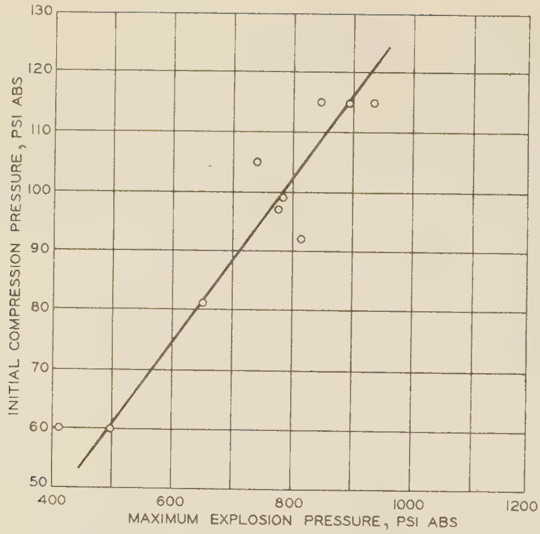

Fig. 6 Curve Showing Relation Between Initial Compression Pressure and Maximum Explosion Pressure

TABLE 2 DATA TAKEN FROM INDICATOR DIAGRAMS DURING TESTS CONDUCTED WITH SELECTED AIR-FUEL RATIO

Test no.	P_1	P_2	P_2/P_1	p_P	w_a/w_p	$(dp/dt)_{avg}$
31	81	655	8.09	8.8	11.7	1925
36	92	815	8.86	10.3	11.3	2730
41	99	785	7.94	9.9	12.8	2000
33	97	772	7.98	11.0	11.2	1450
23	105	735	7.00	12.0	11.1	1190
35	115	845	7.35	13.3	11.0	1720
37	115	935	8.15	13.3	11.0	3370
42	115	895	7.76	11.9	12.3	2100

TABLE 3 DATA TAKEN FROM INDICATOR DIAGRAMS DURING TESTS USING 4-IN-DIAM RUPTURE DISK

Test no.	P_s	P_1	P_2	P_2/P_1	p_P	w_a/w_p
30	66	60	255	4.25	6.0	12.8
28	66	60	285	4.75	6.0	12.8
27	66	60	255	4.25	6.0	12.8
43	122	99	455	4.45	9.9	12.8
44	188	115	475	4.13	11.5	12.8

TABLE 4 RESULTS OF SIX TESTS USING 8-IN-DIAM RUPTURE DISK

Test no.	P_s	P_1	P_2	P_2/P_1	p_P	w_a/w_p
51	134	65	245	3.77	6.5	12.8
49	105	85	305	3.59	8.5	12.8
48	75	65	215	3.31	6.5	12.8
47	122	97	395	4.07	9.7	12.8
46	105	85	305	3.59	8.5	12.8
45	75	65	225	3.46	6.5	12.8

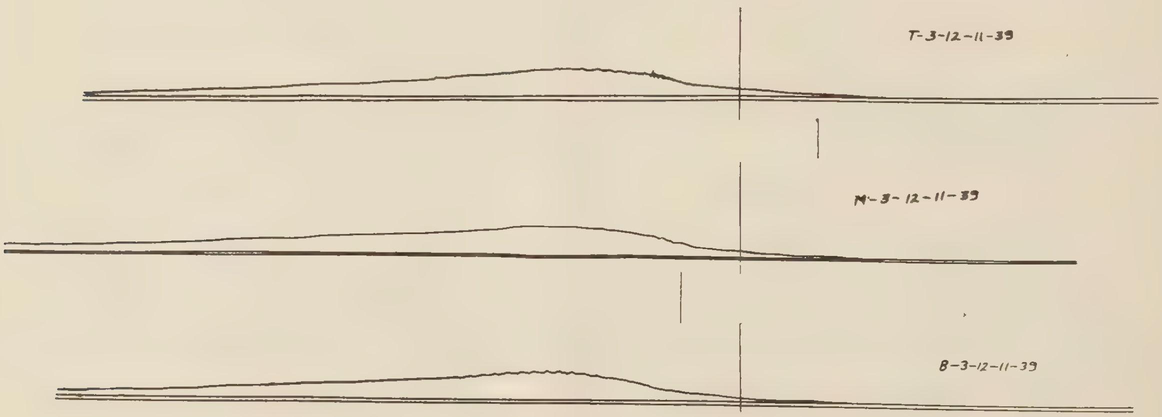

Fig. 7 Test No. 28: Typical Indicator Diagrams Where 4-In-Diam Rupture Disk Is Used

(The spark plug was at the bottom of the vessel. The partial pressure of the propane was 6 in. of mercury and the total initial pressure was 60 psi abs.)

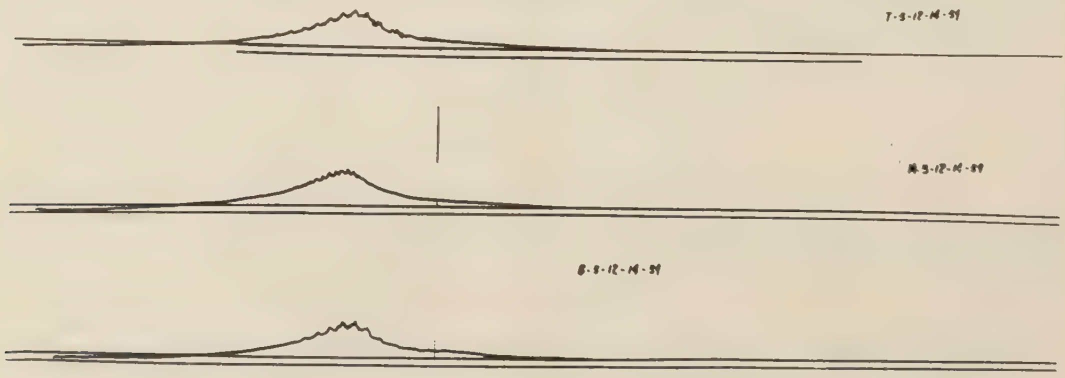

Fig. 8 Test No. 47: Indicator Diagrams for Test Where 8-In-Diam Rupture Disk Was Used

(The spark plug was at the bottom of the vessel. The partial pressure of the propane was 9.7 in. of mercury and the total initial pressure was 97 psi abs.)

TABLE 5 RESULTS OF FOUR TESTS USING 12-IN-DIAM RUPTURE DISK

Test no.	P_s	P_1	P_2	P_2/P_1	P_p	w_a/w_p
55	150	115	335	2.91	11.5	12.8
54	85	71	175	2.46	7.1	12.8
53	105	85	215	2.53	8.5	12.8
52	85	71	175	2.46	7.1	12.8

TABLE 6 RESULTS OF TESTS TO DETERMINE THE EFFECTS OF VOLUME OF EXPLOSION VESSEL

Test no.	P_s	P_1	P_2	P_2/P_1	P_p	w_a/w_p
60	134	103	265	2.58	10.3	12.8
59	75	65	165	2.54	6.5	12.8
58	105	85	215	2.53	8.5	12.8
57	122	97	255	2.48	9.7	12.8

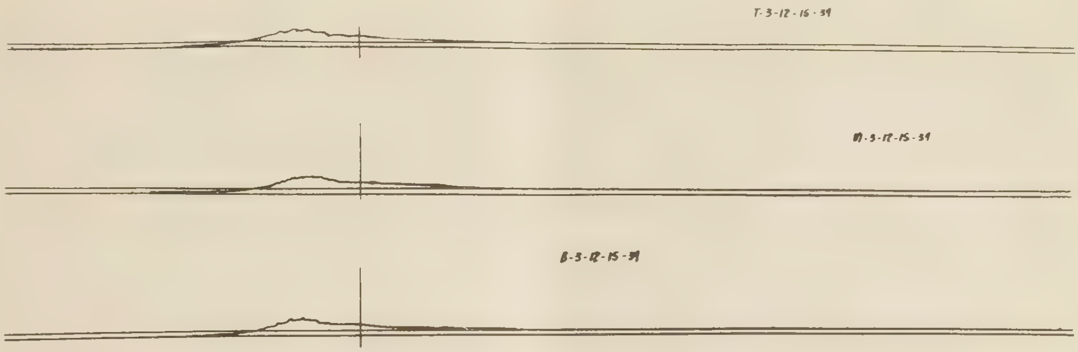

FIG. 9 TEST NO. 54: TYPICAL INDICATOR DIAGRAMS WHERE 12-IN-DIAM RUPTURE DISK WAS USED TO RELIEVE EXPLOSION PRESSURE
(For this test the spark plug was at the bottom of the vessel. The partial pressure of the propane was 7.1 in. of mercury and the initial total pressure was 71 psi abs.)

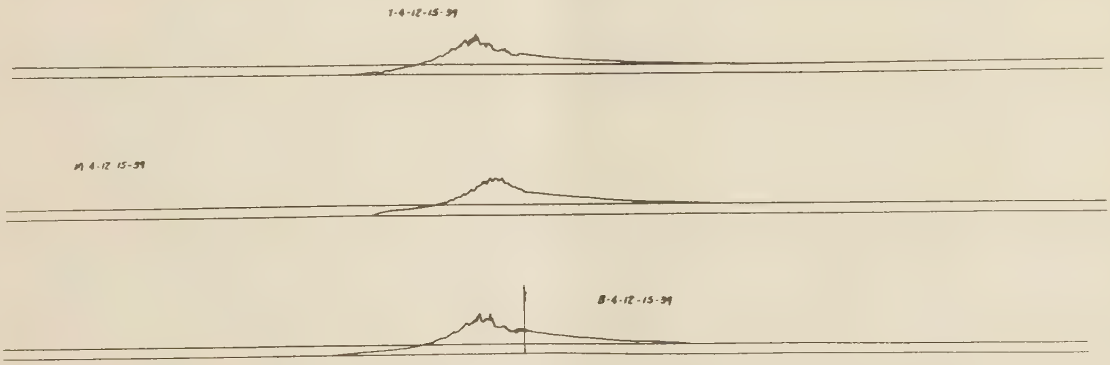

FIG. 10 TEST NO. 55: TYPICAL INDICATOR DIAGRAMS WHERE 12-IN-DIAM RUPTURE DISK WAS USED
(For this test the spark plug was at the bottom of the vessel. The partial pressure of the propane was 11.5 in. of mercury, while the initial tota pressure was 115 psi abs.)

FIG. 11 TYPICAL INDICATOR DIAGRAM SHOWING LARGE PRESSURE WAVES WHEN LARGE DISK RUPTURES
(The spark plug was at the bottom of the vessel and the diagram is from the indicator at the bottom of the vessel. The partial pressure of the propane vapor was 5 in. of mercury. The total initial compression pressure was 75 psi abs. The rupture disk was 16 in. in diam.)

indicator diagrams for these explosions are given in Table 2. The relation between the initial compression pressure and the maximum explosion pressure is plotted in Fig. 6.

An examination of the data plotted in Fig. 6 indicates that the relation between the maximum explosion pressure and the initial compression pressure in a closed vessel is linear for the range of pressure covered. The maximum explosion pressure was approximately eight times the initial compression pressure.

Next, the flange for the 4-in-diam rupture disk was bolted to the explosion vessel. Five tests were conducted using an air-fuel ratio of 12.8 to 1 and various initial compression pressures. The data taken from the indicator cards for these tests are tabulated in Table 3. A typical group of three simultaneous indicator cards is reproduced in Fig. 7 (test No. 28).

It was found that, when the vessel was protected with a 4-in-diam rupture disk, the maximum explosion pressure had dropped to 4.25 times the initial compression pressure.

A series of six tests was made, using an 8-in-diam rupture disk on the explosion vessel. The data taken from this group of tests are given in Table 4. Fig. 8 (test No. 47) is a reproduction of a typical group of three simultaneous indicator cards. It was found that, when an 8-in-diam rupture disk was used to relieve the explosion pressure, the maximum pressure attained by the exploding mixture in the vessel was further reduced to 3.6 times the initial compression pressure.

Four tests were made using a 12-in-diam rupture disk. The data from these tests are given in Table 5. The three simultaneous indicator cards are reproduced in Fig. 9 (test No. 54) and in Fig. 10 (test No. 55). These are typical of all the tests for this size rupture disk. Here it was found that for this size rupture disk the maximum pressure was only 2.5 times the initial compression pressure.

To determine the effect of the volume of the explosion vessel, a group of four tests was undertaken using 8-in-diam rupture

disks with the vessel one half full of water. The data from these tests are tabulated in Table 6. The ratio of the maximum pressure to initial compression was found to be 2.5 instead of 3.6 for the entire vessel.

Additional tests were conducted using 16-in- and 24-in-diam rupture disks to relieve the explosion pressure. Fig. 11 is a typical indicator card made using a 16-in. rupture disk. It will be noted that, immediately after the disk ruptured, several pressure waves of extremely large amplitude were recorded. Similar waves were also present when the 24-in-diam rupture disk was used.

To demonstrate that these pressure waves are some function of the exploding gases and not the bursting of the rupture disk, two 16-in-diam disks similar in every respect were ruptured by rapidly raising the air pressure in the vessel with compressed air. When the disk ruptured, the pressure rapidly declined to atmospheric without any of these vibrations being present.

Since these large-amplitude vibrations were present only when 16-in- and 24-in-diam rupture disks were used and not when 12-in-diam and smaller were used under identical conditions, it seems logical to assume the relation of the diameter of the vessel to the diameter of the rupture disk determines whether or not these waves will appear. Until further studies can be made and these pressure waves explained more fully, it will be impossible to interpret the value of these large-diameter rupture disks for relieving combustion explosions.

Fig. 12 is a plot of all of the foregoing data. This shows clearly the effectiveness of the various sizes of rupture disks in relieving the explosion pressure. As was to be expected from Fig. 6, the maximum explosion pressure is a linear function of the initial compression pressure in every case.

Conclusion

As was stated at the outset, the object of this investigation was the determination of the effectiveness of a rupture disk for relieving the rapid pressure rise in a pressure vessel, harmlessly, during a combustion explosion of its contents. The experimental data presented here are merely a first step toward the solution of this problem. This matter is quite complicated and much must yet be learned about combustion explosions in general before the problem can be called solved. Many of the variables which might affect the results were either disregarded or only crudely or partially controlled. However, the results thus far obtained do indicate that by using a higher factor of safety in designing the vessel together with a rupture disk of suitable size,

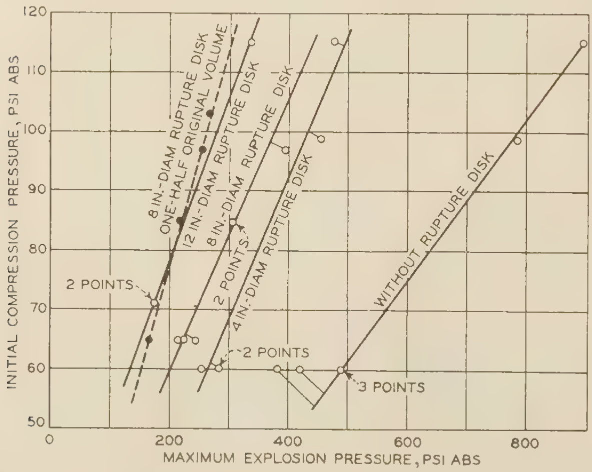

Fig. 12 Curves Showing Relation Between Initial Compression Pressure and Maximum Explosion Pressure for Various Tests

every vessel containing an explosive combustible mixture can be protected. For many of the less violently explosive mixtures, a rupture disk alone will give absolute protection from a destructive explosion.

Discussion

C. E. Huff.[2] Will rupture disks protect air-receiver tanks and air lines in which combustion explosions occur? To find an answer to that question, the writer's company began the explosion study described in part by the author.

Observe that, when violent combustion explosions of "ideal" explosive mixtures of propane and air were created in the sturdy test vessel, there was no difficulty in limiting the maximum pressure in the vessel by the use of a rupture disk:

A 4-in. rupture disk permitted a maximum explosion pressure of 4.25 times the working pressure.

An 8-in. rupture disk kept the maximum explosion pressure in the test vessel down to 3.6 times the working pressure.

A 12-in. rupture disk limited the maximum explosion pressure to 2.5 times the working pressure.

When 16-in. and 24-in. rupture disks were used, the explosion pressure was yet further reduced, but accurate readings were not obtained.

When similar mixtures of propane and air were exploded with the test vessel sealed with a heavy disk to give the effect of a pressure bomb, the maximum explosion pressure was approximately 8 times the working pressure. You will see then how great was the limiting effect of rupture disks of various sizes. Tests were conducted using acetylene and air as the explosive, and the results were quite similar to those quoted.

These experiments certainly show that rupture disks of reasonable size will provide much needed protection to pressure vessels in which combustion explosions of hydrocarbon gas and air mixtures occur.

It may be said: "Although rupture disks limited explosion pressures in these tests, combustion explosions of 'broken-down' lubricating oil and air in actual compressed-air systems may not be as easily controlled." Let us see:

Many are familiar with accounts of the terrific compressed-air-system explosion in a western New York chemical plant late in 1939. Two air compressors were wrecked, several receiver tanks were burst, and pipe fittings were shattered at all turns in the air lines. An oft-repeated question was, "Would rupture disks have protected that air system?" Perhaps now we have an answer!

Recently, an engineer from the same chemical plant related that, soon after the explosion had occurred, rupture disks were installed at several points in the rebuilt air system. According to this engineer, only a short time ago an explosion occurred in the air system and the bursting of one rupture disk prevented damage to the system. The damage to the unprotected system had exceeded $250,000; the explosion which burst the rupture disk cost only the price of a replacement rupture member.

This is only one of many instances where it is known that rupture disks have protected compressed-air systems. On the other hand, we have never heard of the bursting of an air tank or line protected by a rupture disk.

It has been indicated that this explosion study is to be continued, in fact, even now the tests are going forward. However, a more fertile field for observing the protective capacity of rupture disks is in active service where, in the case of thousands of air tanks and other pressure vessels, equipped with rupture disks, the record shows that not a single pressure vessel protected by a proper rupture disk has ever been damaged by overpressure.

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Mathematics of Surge Vessels and Automatic Averaging Control

By C. E. MASON,[1] FOXBORO, MASS., and G. A. PHILBRICK,[2] SHARON, MASS.

In this paper the authors report on a practical application of the quantitative methods which they have described previously (2, 6)[3] in connection with process and control analysis. First, the properties of surge vessels are considered from a functional point of view. The influence on these properties of externally applied controls is next discussed. Proceeding from the simpler to the more involved, control systems of various types are introduced and applied to a vessel. The performance of each of these applied systems is separately examined and illustrated under significant assumed conditions. Considerable attention is given to a definite method of control which involves, as the master instrument, one having a proportional-plus-floating characteristic, and which, it is felt, may justifiably be referred to as "automatic averaging control."

Introduction

An account of the use of "automatic averaging control" as an operating technique in modern continuous processing was given recently in a paper (1) by J. B. McMahon. The present paper is devoted to a quantitative presentation of the mathematics underlying this interesting branch of automatic control.

Dynamically, a surge vessel can be compared both to a shock absorber and to a flywheel. Fluid systems possessing such properties are supposed to absorb or release fluid at such times and in such a manner that violent changes in one or more of a group of related flows need not be accompanied by violent changes in another.

In the case of a surge vessel to which fluid is continuously supplied and from which fluid is continuously withdrawn, all flows pertaining to the vessel may be grouped into two sets—a summed "inflow" and a summed "outflow." When these two flows are exactly equal, the quantity of fluid stored in the vessel remains constant. In general, one of the flows will fluctuate and it will be desired to minimize the effect of such fluctuation on the other flow. For convenience it may be assumed that the inflow is the independently fluctuating quantity and that the outflow varies in some fashion as a result. The reverse circumstance is equally significant, but the two problems are basically analogous and the treatment of one will suffice.

In the case of a tank holding liquid, which for the sake of concreteness will be considered as typical of all possibilities,[4] the level at which the liquid stands is an indication of the quantity stored up in the tank. Thus the three variables, i.e., inflow, level, and outflow, may be taken as completely descriptive of the dynamic state of the system. The behavior of any two of these variables definitely determines the behavior of the third. In uncontrolled surge vessels, the dynamic relationships between inflow and level, level and outflow, and inflow and outflow may have all variety of forms. When a definite relationship of the proper type is enforced between the level and the outflow by the application of automatic control, it will be shown that the efficiency of the surge vessel as a "shock absorber" can be increased to a remarkable extent. In such an application, it should not be considered that the level is being "controlled" in the conventional sense that a predetermined value of level is to be held to within close tolerances, nor indeed that the outflow is to be so controlled. In reality, the true objective of this type of automatic control is to maintain continuously an advantageous relationship between these two variables.

Beginning with an uncontrolled vessel, having only "self-regulation," the application of control is presented in stages leading up to the full automatic-averaging-control installation. Each stage is accompanied by an illustration showing results obtainable in practical cases. Included in each figure is a diagrammatic sketch of the particular physical system considered. In every case the system shown comprises a vessel with a flow line leading to the vessel and a flow line leading from the vessel. Indicating instruments are shown symbolically and are applied to the inflow, level, and outflow. The instrument applied to the inflow serves merely, in each instance, to give a continuous indication of that variable, whereas in some of the cases the level or the outflow or both are controlled as well as measured; this is shown by replacement of the indicator by a controller.

The nature of the relationships among inflow, level, outflow, and time, under cyclic disturbances, makes it appear necessary to resort to the somewhat intricate involvements of classical differential equations in order to develop explicit quantitative expressions for these relationships. However, an investigation into the possibilities offered by the symbolic forms of Heaviside's operational calculus discloses an uncanny applicability to these purposes. Thus, even though the details of the operational methods themselves are beyond the scope of the present paper, such methods have been employed in the analytical development. For the benefit of those interested in the formal mathematics, a condensed description of the operational procedure is given (in italics) in the text under its respective section. The final expressions which give the over-all relationships under cyclic conditions are included in the main body of the text, which is so arranged that complete continuity is not lost by the reader who omits the mathematical development.

If the validity of the final expressions can be established either by inspection or by actual usage, it is by no means necessary that the actual user even be concerned with their origin or the manner of their development, except for the personal satisfaction he might derive from a familiarity with the details of the mathematical machinery. Oliver Heaviside himself expressed this attitude in his famous query:

"Shall I refuse my dinner because I do not fully understand the process of digestion?"

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[1] Director of Engineering, Mason-Neilan Regulator Co., Boston, Mass., formerly Director of Control Research, The Foxboro Company.

[2] Research Engineer, The Foxboro Company.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

[4] Gas holders, steam accumulators, etc., can be subjected to the same reasoning as is here applied to surge vessels for liquid.

Contributed by the Committee on Industrial Instruments and Regulators of the Process Industries Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

A major purpose of this paper is that of demonstrating to the practical industrial engineer, such as those who are actually confronted with averaging control problems, the extreme practicability of some of these simplified formulas. The practical, economic value of the formulas cannot, perhaps, be fully appreciated except by numerical substitution. The astonishing character of soundly derived mathematical results was expressed by Heinrich Hertz:

"One cannot escape the feeling that these mathematical formulas have an independent existence and an intelligence of their own; that they are wiser than we are; wiser even than their discoverers; that we get more out of them than was originally put into them."

The formulas which describe the results under cyclic conditions, as presented in the text. contain only those factors which are necessary to a practical determination of the over-all response. They have been simplified by logical assumptions. Much of the complexity and unimportant detail has been eliminated and emphasis given to those factors which are or may be influential in actual industrial applications.

Consideration of sine-wave disturbances leads to the appearance of trigonometric functions in some of the mathematics. It is normal practice in much applied mathematics to express trigonometric angles in radians. Conventional trigonometric tables, however, are compiled in terms of angular degrees. For this reason a departure is taken from normal practice, in that the final forms are made to appear as dimensionless ratios of an "angle whose tangent is something" to an angle of 90°, or as $(\tan^{-1}X)/90°$.

From the simplified general formulas, some exemplary numerical results have been included in the figures. These results pertain only to the particular dimensions assumed for the surge vessel and to the particular nature and magnitude of the assumed disturbances. It is hoped, however, that these tabulations will serve to rationalize the complexities of the general problem.

The following special nomenclature applies for the simplified text as well as for the formal mathematics.

Notation, Definitions, and Units

V = level above an assumed base; feet above bottom of vessel
V_n = normal or "desired" value of V
b = proportional or throttling band of V, ft
r = reset constant, units per min
Q_s = inflow to vessel (total), gpm
Q = outflow from vessel (total), gpm
Q_m = $^1/_2(Q_{\min} + Q_{\max})$ = mid-value of Q
k = $(Q_{\max} - Q_{\min})$ = band in which Q may be varied by controls, gpm
d = diameter of vessel, assumed upright and cylindrical, ft
A = capacity of vessel, gal per ft (= $5.88\, d^2$)
R = resistance to outflow (linear), ft per gpm
R_e = b/k = equivalent "resistance" under control, ft per gpm
t = time, min
h = half-period of oscillation, min
$(X)'$ = first derivative of X
$(X)''$ = second derivative of X
p = d/dt = differential operator

Numerical Values Assumed Constant in All Examples

V_n = 5 ft (mid-value of allowable range of level variation)
$Q_{\min}$ = 100 gpm
$Q_{\max}$ = 300 gpm
Q_m = $^1/_2(Q_{\min} + Q_{\max})$ = 200 gpm
k = $(Q_{\max} - Q_{\min})$ = 200 gpm
d = two values considered = 4.125 and 8.25 ft
A = two values considered = 100 and 400 gal per ft
h = two values considered = 10 and 20 min

Test Disturbances (in Inflow) Applied for All Modes of Control

To represent a wide variety of disturbances, the inflow is assumed to undergo three different sorts of variation, as follows:

So-Called Condition (a)

In a state of perfect balance, the inflow is assumed to change suddenly from a constant value of 200 gpm to a new constant value of 250 gpm

This condition can be expressed mathematically as follows:

$$Q_s = 200 \text{ for } (t < 0),\ Q_s = 250 \text{ for } (t > 0)$$

So-Called Condition (b_1)

The inflow is assumed to be engaged in a permanent sine-wave oscillation about a value of 200 gpm at an amplitude of 50 gpm and with a half-period of 10 min.

This condition can be expressed mathematically as follows:

$$Q_s = 200 + 50 \sin\left[180° \frac{t}{10}\right] \text{ for } (-\infty < t < \infty)$$

So-Called Condition (b_2)

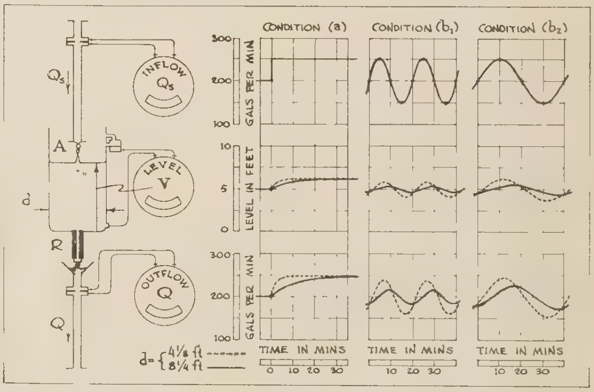
Same as condition (b_1) but with a half period of 20 min.

This condition can be expressed mathematically as follows

$$Q_s = 200 + 50 \sin\left[180° \frac{t}{20}\right] \text{ for } (-\infty < t < \infty)$$

Single Resistance-Capacity Unit as Surge Vessel; Self-Regulation

An elementary resistance-capacity system of the sort described



Numerical Solutions		No Explicit Control $R = 0.025$ feet/(gal/min)		
	$A = 5.88d^2$	Condition (a)	Condition (b_1)	Condition (b_2)
Inflow Q_s gals/min		200 - - - ($t<0$) 250 - - - ($t>0$)	$200 + 50 \sin\left[180° \frac{t-0}{10}\right]$ (h = 10 mins)	$200 + 50 \sin\left[180° \frac{t-0}{20}\right]$ (h = 20 mins)
Level V feet	100 gals/foot - - - -	$6.25 - 1.25 e^{-0.40t}$	$5 + 0.983 \sin\left[180° \frac{t-2.12}{10}\right]$	$5 + 1.164 \sin\left[180° \frac{t-2.38}{20}\right]$
	400 " ——	$6.25 - 50 e^{-0.10t}$	$5 + 0.380 \sin\left[180° \frac{t-4.02}{10}\right]$	$5 + 0.671 \sin\left[180° \frac{t-6.39}{20}\right]$
Outflow Q gals/min	100 " - - - -	$250 - 50 e^{-0.40t}$	$200 + 39.32 \sin\left[180° \frac{t-2.12}{10}\right]$	$200 + 46.54 \sin\left[180° \frac{t-2.38}{20}\right]$
	400 " ——	$250 - 50 e^{-0.10t}$	$200 + 15.17 \sin\left[180° \frac{t-4.02}{10}\right]$	$200 + 26.85 \sin\left[180° \frac{t-6.39}{20}\right]$
Time-Boundaries		($t > 0$)	($-\infty < t < \infty$)	($-\infty < t < \infty$)

Fig. 1 Single Resistance-Capacity Unit as Surge Vessel; Self-Regulation

in an earlier paper (2) by one of the authors can be considered in the role of a surge vessel. Fig. 1[5] shows such a system with indicating instruments on inflow, level, and outflow.

---

[5] In the curves of Figs. 1 to 8, full lines are for one capacity and dotted lines are for one quarter of the capacity (or one half of the diam).

Following the development in the earlier reference we may write

$$(V)' = (Q_s - Q)/A \qquad [1]$$

$$V = RQ \qquad [2]$$

The two following basic equations are obtainable by familiar methods from Equations [1] and [2]

$$AR(V)' + V = RQ_s \qquad [3]$$

$$AR(Q)' + Q = Q_s \qquad [4]$$

Solutions, similar to those in the earlier paper (2), for the response of the level and the outflow, when the inflow is changed suddenly from a constant value of 200 gpm to a new constant value of 250 gpm, are shown by the curves under condition (*a*) of Fig. 1. The numerical equations given in the same figure, for the same assumed conditions, express the deviation of the level V from the normal value of $V_n = 5$ ft and the deviation of the outflow Q from $Q_m = 200$ gpm.

* * *

Operational methods can also be used for solutions of this sort and are especially useful when oscillatory disturbances are to be dealt with. Operational or symbolic calculus has been placed on a rigorous foundation and a number of excellent texts (3, 4, 5) are available which describe its application. From Equations [3] and [4], the following equivalent operational expressions are directly derived

$$V = \frac{R}{1 + ARp} \cdot Q_s \qquad [5]$$

$$Q = \frac{1}{1 + ARp} \cdot Q_s \qquad [6]$$

For a single sudden change in Q_s, simple exponential solutions can be obtained directly from Equations [5] and [6] as well as from Equations [3] and [4].

For steady sine-wave oscillations in the inflow, the amplitude and phase of the resulting oscillations of level and outflow are obtainable by replacing p in the operators with the imaginary angular velocity ($i\pi/h$). Briefly, if the operator then becomes ($u + iv$), the relative amplitude is given by $\sqrt{u^2 + v^2}$ and the phase angle by $\tan^{-1}(v/u)$, while the true[6] lag in time units is $-(h/\pi)\tan^{-1}(v/u)$. Thus for steady oscillations in the inflow the amplitude and lag response of the level and outflow can be obtained from Equations [5] and [6] and are summarized as follows:

$$\frac{\text{Ampl. of } V}{\text{Ampl. of } Q_s} = \frac{R}{\sqrt{1 + G^2}}$$

where

$$G = \pi AR/h$$

$$\text{Lag of } V \text{ versus } Q_s = (h/\pi)\tan^{-1}(G)$$

$$\frac{\text{Ampl. of } Q}{\text{Ampl. of } Q_s} = \frac{1}{\sqrt{1 + G^2}}$$

$$\text{Lag of } Q \text{ versus } Q_s = \text{same as for } V$$

* * *

The equations expressing the values of V and Q under cyclic disturbances of the inflow must contain harmonic functions of time. These can be brought into the equations as sine functions of angular degrees. General forms for the equations of V and Q under the cyclic conditions (b_1) and (b_2) may be written

[6] Time lag, as such, should not be given significance except in the case of sinusoidal oscillations, as here, or in the case of a pure time delay or distance-velocity lag (2).

$$V = V_n + A_v \sin\left[180° \frac{t - T_v}{h}\right] \qquad [7]$$

$$Q = Q_m + A_q \sin\left[180° \frac{t - T_q}{h}\right] \qquad [8]$$

The expressions for use under cyclic conditions, which were developed as previously shown by operational methods, may be used to supply the following formulas for the new constants appearing in Equations [7] and [8].

$$A_v = \text{Ampl. of } V = \frac{R \times (\text{Ampl. of } Q_s)}{\sqrt{1 + G^2}}$$

$$= \text{level variation in feet}$$

$$A_q = \text{Ampl. of } Q = \frac{(\text{Ampl. of } Q_s)}{\sqrt{1 + G^2}}$$

$$= \text{outflow variation in gpm}$$

$T_v = T_q = \dfrac{h}{2}\,\dfrac{\tan^{-1}(G)}{90°}$ = time in minutes by which cycles of V and of Q lag behind the cycles of Q_s

The constant G depends upon the characteristics of the process and upon the half-period of the inflow oscillations. Its numerical value is given by

$$G = 3.14 \frac{AR}{h}$$

The quantities "Ampl. of V," "Ampl. of Q," and "Ampl. of Q_s" are the magnitudes of the maximum variation of these variables on either side of their mean values, i.e., one half of their total variation.

The results of numerical substitution in the general formulas, for the assumed conditions (b_1) and (b_2), are included in Fig. 1, together with the curves of their solutions plotted against time. These curves show that the level and the outflow oscillate exactly in phase with one another, but that they are out of phase with the inflow.

The principal merit of this arrangement as a surge-absorbing system lies in its simplicity. Smoothing of the outflow versus the inflow is not impressive. The level can reach an eventual balance anywhere in the vessel, depending upon the average value of the inflow.

Independent Control of the Outflow

In this case a flow controller is installed directly on the outflow, as illustrated in Fig. 2, and is assumed to be completely successful in maintaining this flow at a constant value.

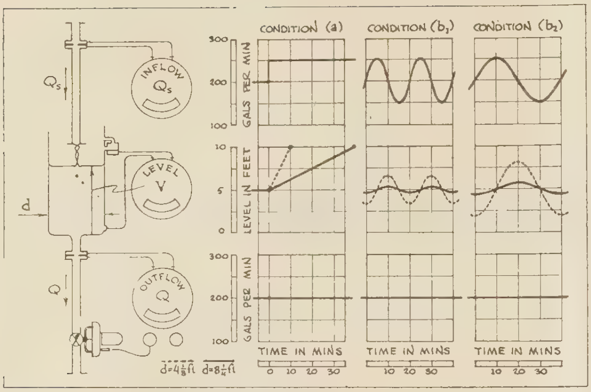
From the universally valid Equation [1]

$$(V)' = (Q_s - Q_m)/A \qquad [9]$$

Where the mean flow Q_m is the constant value at which the outflow happens to be controlled. Equation [9] may be written as the indefinite integral

$$V = \frac{1}{A}\int (Q_s - Q_m)dt$$

which is equivalent to the statement that the level "integrates" the excess of the inflow over the controlled outflow, and does so in inverse proportion to the capacity of the vessel.



Numerical Solutions		Independent Control of Outflow		
	$A = 5.88 d^2$	Condition (a)	Condition (b_1)	Condition (b_2)
Inflow Q_s gals/min		200 ---- ($t<0$) 250 ---- ($t>0$)	$200 + 50 \sin\left[180° \frac{t-0}{10}\right]$	$200 + 50 \sin\left[180° \frac{t-0}{20}\right]$
Level V feet	100 $\frac{\text{gals}}{\text{foot}}$	$5 + 0.5 t$	$5 + 1.59 \sin\left[180° \frac{t-5}{10}\right]$	$5 + 3.18 \sin\left[180° \frac{t-10}{20}\right]$
	400 "	$5 + 0.125 t$	$5 + 0.398 \sin\left[180° \frac{t-5}{20}\right]$	$5 + 0.796 \sin\left[180° \frac{t-10}{20}\right]$
Outflow Q gals/min	100 "	Constant at 200	Constant	Constant
	400 "	" " "	"	"
Time-Boundaries		($t < 0$)	($-\infty < t < \infty$)	($-\infty < t < \infty$)

Fig. 2 Limiting Case; Independent Control of Outflow

For a sudden sustained increase in the inflow Q_s above Q_m, that is for condition (*a*), it is evident that the level assumes a constant rate of increase which depends upon the capacity A, as shown in Fig. 2.

* * *

Operationally

$$V = \frac{1}{Ap} \cdot (Q_s - Q_m) \ldots\ldots\ldots\ldots\ldots [10]$$

In the case of continuous oscillation of the inflow Q_s, under conditions (b_1) and (b_2), the level response may be found by direct integration or by the formal $p = i\pi/h$ substitution already employed. Thus for sine-wave oscillations, we obtain the following response

$$\frac{\text{Ampl. of } V}{\text{Ampl. of } Q_s} = \frac{h}{\pi A}$$

$$\text{Lag of } V \text{ versus } Q_s = (h/\pi) \tan^{-1}(\infty) = h/2$$

$$Q \ldots\ldots\ldots \text{(Constant)}$$

* * *

The general form of the equations for the cyclic conditions (b_1) and (b_2) are

$$V = V_m + A_v \sin\left[180° \frac{t - T_v}{h}\right] \ldots\ldots\ldots\ldots [7]$$

$$Q = Q_m$$

in which

$$A_v = \text{Ampl. of } V = 0.318 \frac{h}{A} \text{ (Ampl. of } Q_s)$$

$$= \text{level variation in feet}$$

$T_v = \dfrac{h}{2}$ = time in minutes by which the cycles of V lag behind cycles of Q_s

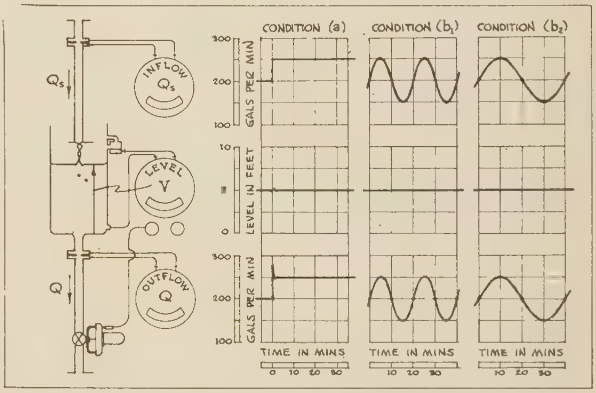
The results of numerical substitution in the general Equation [7], for the assumed conditions (b_1) and (b_2), are given in Fig. 2 together with curves of their time solutions. The behavior in this case under cyclic conditions (b_1) and (b_2) represents the limiting case of "perfect" averaging operation, based on oscillation of the inflow about a constant mean value. It is interesting to note, from the formulas for A_v and T_v, that the level variations are directly proportional to the period of the inflow variations and inversely proportional to the capacity or area of the vessel, and that the cycles of the level are exactly one-fourth period out of phase with the cycles of the inflow. This is evident also from the curves.

With this type of control, perfect smoothing of the outflow with respect to the inflow is made inevitable by the application of the flow controller on the outflow, but no recognition is taken of the level, which will gradually rise or fall, even to limits, depending upon the difference between the accumulated average of the inflow and the value at which the outflow is controlled.

In practical application of this method, periodic manual re-adjustment of the controlled outflow may in some cases be a satisfactory mode of operation, especially when the magnitude or the period of the oscillations encountered compares favorably with the size of the vessel. Such readjustment amounts to matching the controlled outflow to the average of the inflow taken over considerable periods of time. The aim of automatic averaging control is to make such readjustment continuous and automatic, to approach as nearly as possible to perfect smoothing of the outflow versus the inflow, consistent with keeping the level continuously within the vessel. Returning the level to a predetermined central value is also desirable as well, since this will permit optimum absorption both of sustained changes and of sudden surges, irrespective of the direction in which these occur.

Independent Control of the Level

Automatic control of a system involving only a single capacity unit can be carried out to any desired degree of effectiveness, even with types of control which in an operating sense may be



Numerical Solutions		Independent Control of Level		
	$A = 5.88 d^2$	Condition (a)	Condition (b_1)	Condition (b_2)
Inflow Q_s gals/min		200 ...... ($t<0$) 250 ...... ($t>0$)	$200 + 50 \sin\left[180° \frac{t-0}{10}\right]$	$200 + 50 \sin\left[180° \frac{t-0}{20}\right]$
Level V feet	100 $\frac{\text{gals}}{\text{foot}}$	Substantially constant at 5	Constant	Constant
	400 "	" " " "	"	"
Outflow Q gals/min	100 "	Similar to Q_s	Same as Q_s	Same as Q_s
	400 "	" " "	" " "	" " "
Time-Boundaries		($t > 0$)	($-\infty < t < \infty$)	($-\infty < t < \infty$)

Fig. 3 Limiting Case; Independent Control of Level

called elementary. The problem is one exclusively of rapid measurement and manipulation. In Fig. 3 such a control system

is assumed to be applied to maintain a constant level in the vessel. The level controller itself might have, for example, a proportional characteristic with an extremely narrow proportional or throttling band. In this sense the equations which are given later for proportional control may be considered to apply here, but with an extremely small value of proportional band b. Whatever means seem most proper actually to achieve a substantially constant level, we are for the moment concerned only with the effect on the outflow. As shown graphically in Fig. 3, this degree of level control is acquired at the cost of full variation of the outflow. The latter flow essentially duplicates the inflow, even to the point of being in phase with it.

From the point of view of automatic averaging control this example represents a limiting case, opposite to that of Fig. 2, and is brought in only as a logical step in the development.

Theoretically, the magnitude of the outflow variations is independent of the area of the vessel. Only the practical impossibility of reducing the proportional band precisely to zero, or some imperfection in the operation of the controls, could cause any reduction in the amplitude of the outflow cycles.

Cascaded Control

The term "cascaded control" appears appropriate to describe in general a system of control whereby the operating means of one controller automatically adjusts the control-point setting of one or more succeeding controllers, intermediate between the initial or master controller and the final controlling means or manipulated variable. In averaging level control, this would correspond to allowing the operating means of the level controller to "set the control point of" a special flow controller on the outflow.

Such inclusion of an auxiliary flow controller for the outflow has the advantage that it eliminates any direct dependence of the outflow upon the behavior of the level, or on external-pressure relationships such as changes in the drop across the outlet valve. It also eliminates similar dependence of the outflow upon whatever pressure may be impressed on the liquid surface, as shown symbolically in the last two figures of the paper. This method is a recognized procedure in control technique.

In the remaining examples it will be assumed, as in the earlier paper (6), that the cascaded method of control is employed. Thus, it is assumed that the "control point" of the flow controller on the outflow is set throughout its operating range by the operating means of the level controller, and that the relationship thus formed is uniform within that range.

Proportional Control of the Level, Cascaded

If the level instrument is assumed to be a proportional controller, as described in paper (6), we may write the controller equation as a relationship between the level V and the outflow Q, or as

$$Q - Q_m = (k/b)(V - V_n) \qquad [11]$$

where ($Q_{\min} < Q < Q_{\max}$), and in which it is assumed that the proportional band b is so located that V_n is in the middle of that band.

Combining Equation [1] for the "process" with Equation [11] for the controller and making the substitution

$$R_e = b/k \qquad [12]$$

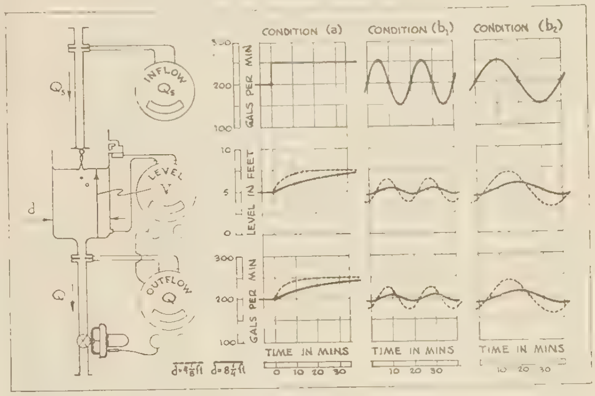
gives for the level and the outflow, respectively

$$AR_e(V - V_n)' + (V - V_n) = R_e(Q_s - Q_m) \qquad [13]$$

$$AR_e(Q - Q_m)' + (Q - Q_m) = (Q_s - Q_m) \qquad [14]$$

Equations [13] and [14] are similar to Equations [3] and [4] for the resistance-capacity unit. This fact is no coincidence as the systems are directly analogous. The ratio (b/k) for automatic control is analogous to the resistance R under self-regulation and may be thought of as an equivalent "resistance" R_e, so designated in the nomenclature in order to emphasize the analogy.

The response of the level and the outflow to the sudden sustained change in the inflow is obtained precisely as in the analo-



Numerical Solutions	Control constants: $b = 10$ ft., $k = 200$ gals/min			
	$A = 5.88d^2$	Condition (a)	Condition (b_1)	Condition (b_2)
Inflow Q_s gals/min		200 ---- ($t<0$) 250 ---- ($t>0$)	$200 + 50 \sin\left[180° \frac{t-0}{10}\right]$	$200 + 50 \sin\left[180° \frac{t-0}{20}\right]$
Level V feet	100 gals/foot	$7.5 - 2.5\, e^{-0.20t}$	$5 + 1.34 \sin\left[180° \frac{t-3.20}{10}\right]$	$5 + 1.90 \sin\left[180° \frac{t-4.23}{20}\right]$
	400 "	$7.5 - 2.5\, e^{-0.05t}$	$5 + 0.393 \sin\left[180° \frac{t-4.50}{10}\right]$	$5 + 0.759 \sin\left[180° \frac{t-8.04}{20}\right]$
Outflow Q gals/min	100 "	$250 - 50\, e^{-0.20t}$	$200 + 26.85 \sin\left[180° \frac{t-3.20}{10}\right]$	$200 + 39.32 \sin\left[180° \frac{t-4.23}{20}\right]$
	400 "	$250 - 50\, e^{-0.05t}$	$200 + 7.86 \sin\left[180° \frac{t-4.50}{10}\right]$	$200 + 15.17 \sin\left[180° \frac{t-8.04}{20}\right]$
Time-boundaries		($t > 0$)	($-\infty < t < \infty$)	($-\infty < t < \infty$)

Fig. 4 Proportional Level Controller

gous case under self-regulation. The results are shown under condition (*a*) in Fig. 4.

* * *

In operational form, Equations [13] and [14] become

$$V - V_n = \frac{R_e}{1 + AR_e p} \cdot (Q_s - Q_m) \qquad [15]$$

$$Q - Q_m = \frac{1}{1 + AR_e p} \cdot (Q_s - Q_m) \qquad [16]$$

Equations [15] and [16] are similar to Equations [5] and [6] for the resistance-capacity unit. The attenuation, or amplitude ratio, and the lag involved in the response of the level and outflow to continuous oscillation of the inflow are also given by analogous expressions and may be written

$$\frac{\text{Ampl. of } V}{\text{Ampl. of } Q_s} = \frac{R_e}{\sqrt{1 + G^2}}$$

where

$$(G = \pi A R_e / h)$$

$$\text{Lag of } V \text{ versus } Q_s = (h/\pi) \tan^{-1}(G)$$

$$\frac{\text{Ampl. of } Q}{\text{Ampl. of } Q_s} = \frac{1}{\sqrt{1 + G^2}}$$

Lag of Q versus Q_s = same as for V

* * *

General equations for V and Q under cyclic conditions (b_1) and (b_2) may again be written

$$V = V_n + A_v \sin\left[180° \frac{t - T_v}{h}\right] \ldots\ldots\ldots\ldots [7]$$

$$Q = Q_m + A_q \sin\left[180° \frac{t - T_q}{h}\right] \ldots\ldots\ldots\ldots [8]$$

The numerical values of the constants in these equations may be determined from the following formulas

$$A_v = \text{Ampl. of } V = \frac{R_e \times (\text{Ampl. of } Q_e)}{\sqrt{1 + G^2}}$$

$$= \text{level variation in feet}$$

$$A_q = \text{Ampl. of } Q = \frac{(\text{Ampl. of } Q_e)}{\sqrt{1 + G^2}}$$

$$= \text{outflow variation in gpm}$$

$T_v = T_q = \dfrac{h}{2} \dfrac{\tan^{-1}(G)}{90°}$ = time in minutes by which cycles of V and of Q lag behind cycles of Q_e

in which

$$G = 3.14 \frac{AR_e}{h}$$

For the numerical examples considered

$$R_e = 0.005\, b \text{ and}$$

$$G = 0.0157 \frac{Ab}{h}$$

Results of numerical substitution in the general equations, for the assumed conditions (b_1) and (b_2), are included in Fig. 4, together with curves of the time solutions.

It is evident that the remarks already made on the performance of the simple resistance-capacity system, Fig. 1, apply almost equally well here. The use of the proportional type of level controller in this application merely imparts to the vessel a definite, mechanical, self-regulating property similar to that of the resistance-capacity system shown in Fig. 1, while the use of "cascaded control," as described, prevents alteration, by pressure changes in any form, of the already limited averaging characteristics of the system. In the case illustrated in Fig. 4, the proportional or throttling band is made equal to the full allowable range of the level. For proportional bands narrower than this value, the smoothing of the outflow is even less effective. Wider proportional bands, on the other hand, would not permit balance of the level within the allowable range, or within the confines of the vessel, for all values of inflow, even under steady conditions.

When the range of the instrument is so selected that it fits the allowable range of level variation, a proportional band having a width equal to this range, such as that chosen in Fig. 4, is generally referred to as a "100 per cent throttling range." From the viewpoint of averaging control this so-called 100 per cent throttling controller has a very limited ability toward smoothing of the outflow. Some of the limitations are shown by the following observations: (*a*) If the outlet resistance R of Fig. 1 had been located 5 ft below the bottom of the vessel, the value of R to give the same level in balance would have been equal to that of R_e in Fig. 4, and the results of self-regulation and of the 100 per cent throttling control would have been identical. (*b*) If, in such a system as is illustrated in Fig. 1, a constant static pressure of approximately 2 psi had been exerted on the liquid surface, the results of self-regulation and those of 100 per cent throttling control would have been identical. (*c*) The square-root characteristics of an ordinary valve, which could replace the resistance R in the system of Fig. 1, and which could be adjusted to give a level of 5 ft for a flow of 200 gpm, would provide the same averaging effect at the center of the level range as does the 100 per cent throttling control, although it would give less averaging at levels below the center and more above it.

Methods (*a*) and (*b*) of the preceding paragraph could be extended to increase the averaging effect throughout the allowable level range. This would be accomplished, however, at the cost of limitation of the range of inflow variations which would permit the level to remain within the allowable range. Adjustment of the outflow resistance in connection with any of methods (*a*) to (*c*) permits establishment of the value of the level for a given outflow and a given pressure drop across the resistance but does not permit adjustment of the range of level variation for a given variation of the inflow. The really practical advantages in using an automatic control instrument with adjustable proportional or throttling band lie in the fact that the level variation may be retained within a definite range for any specified variation in the outflow, and regardless of the pressure drops existing across the valve. The maximum capacity of the valve is the only factor limiting the range of outflow variation.

Proportional-Plus-Floating Control of the Level, Cascaded

For automatic averaging control, it is evident that there is a real advantage in the use of a level controller which controls to a single ultimate value rather than to within a band of values, i.e., in the use of a controller which has point-stability rather than band-stability alone. The severity of the corrective measures set up by such a controller may be moderated without simultaneously spreading out the band in which the level can ultimately balance, as is the case with the proportional form of instrument. The proportional-plus-floating type of controller, known to be a versatile form in other applications, fits this requirement and will be considered in an installation similar to that of the preceding section. The level controller, this time with a proportional-plus-floating characteristic, is again assumed to operate by setting the "control point" of a controller on the outflow.

As described in the authors' previous paper (6) and for the present installation, the proportional-plus-floating controller may be identified by the following equation

$$(Q)' = (k/b)[(V - V_n)' + r(V - V_n)] \ldots\ldots\ldots [17]$$

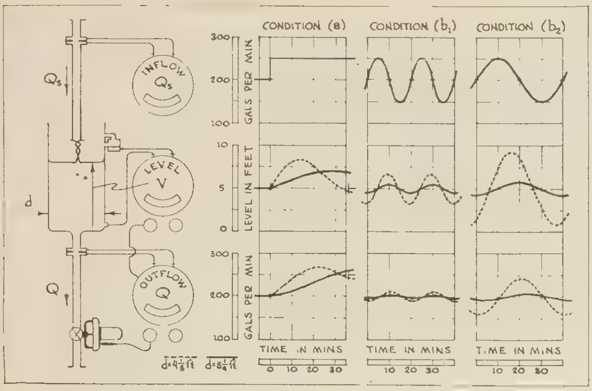
in which k and b have already occurred, and in which r is the so-called reset constant.

It should be pointed out that the proportional or throttling band b, as defined in paper (6), need not exist in an entirely tangible form. The expressed value of this band may be considerably greater than the full available range of the level, in which case the controls act as though the full extent of such a band were really effective. This places no permanent restriction on the performance of the proportional-plus-floating controller since in operation this band is automatically moved in such a way that the level returns to the normal value for balanced conditions.

To determine the properties of the system under this form of control, we may combine Equation [17] for the controller with the "process" Equation [1]. By methods described in detail in the earlier paper (6), one obtains for the level V

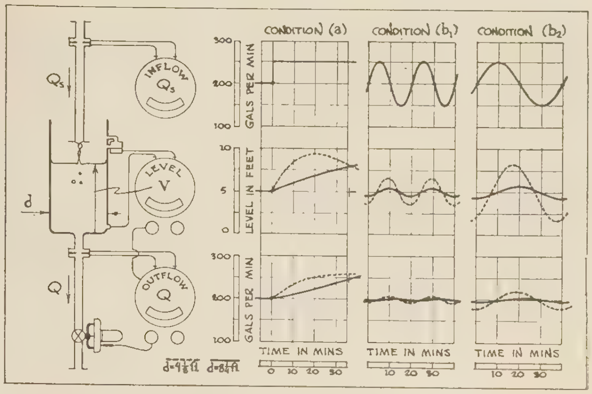
$$AR_e(V - V_n)'' + (V - V_n)' + r(V - V_n) = R_e(Q_e - Q_m)' \ldots . [18]$$

and for the outflow Q



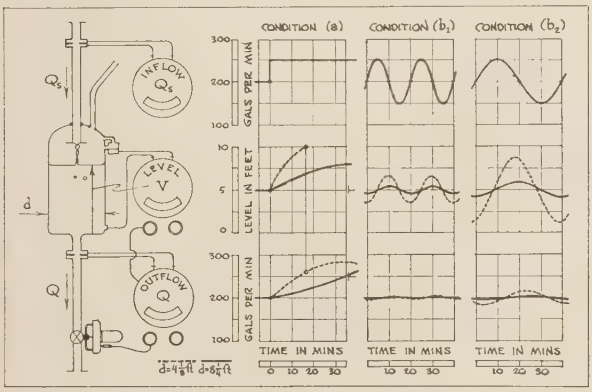
NUMERICAL SOLUTIONS		CONTROL CONSTANTS: $b = 30$ ft, $r = 0.15$ 1/min, $k = 200$ gals/min		
	$A = 5.88d^2$	CONDITION (a)	CONDITION (b_1)	CONDITION (b_2)
INFLOW Q_s gals/min		200 ---- ($t<0$) 250 ---- ($t>0$)	$200+50\sin\left[180°\frac{t-0}{10}\right]$	$200+50\sin\left[180°\frac{t-0}{20}\right]$
LEVEL V feet	100 gals/foot ----	$5+5.30e^{-0.0333t}\cos\left[180°\frac{t-16.7}{33.3}\right]$	$5+1.72\sin\left[180°\frac{t-4.26}{10}\right]$	$5+4.35\sin\left[180°\frac{t-6.05}{20}\right]$
	400 " ——	$5+2.54e^{-0.00833t}\cos\left[180°\frac{t-31.9}{63.7}\right]$	$5+0.408\sin\left[180°\frac{t-4.83}{10}\right]$	$5+0.879\sin\left[180°\frac{t-9.85}{20}\right]$
OUTFLOW Q gals/min	100 " ----	$250-53.03e^{-0.0333t}\cos\left[180°\frac{t+3.6}{33.3}\right]$	$200+12.73\sin\left[180°\frac{t-6.68}{10}\right]$	$200+40.05\sin\left[180°\frac{t-10.91}{20}\right]$
	400 " ——	$250-50.71e^{-0.00833t}\cos\left[180°\frac{t+3.4}{63.7}\right]$	$200+3.01\sin\left[180°\frac{t-6.25}{10}\right]$	$200+8.11\sin\left[180°\frac{t-14.11}{20}\right]$
TIME-BOUNDARIES		($t>0$)	($-\infty<t<\infty$)	($-\infty<t<\infty$)

FIG. 5 PROPORTIONAL-PLUS-FLOATING LEVEL CONTROLLER; CASE I



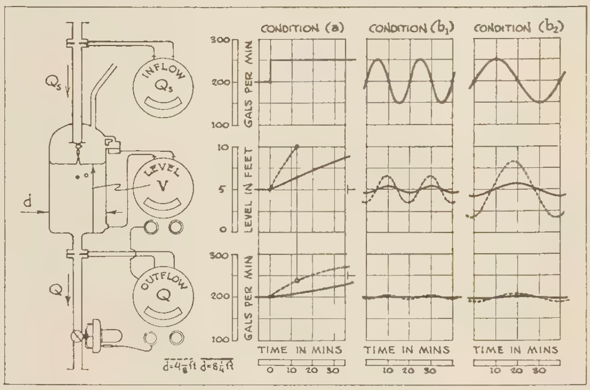
NUMERICAL SOLUTIONS		CONTROL CONSTANTS: $b = 30$ ft, $r = 0.05$ 1/min, $k = 200$ gals/min		
	$A = 5.88d^2$	CONDITION (a)	CONDITION (b_1)	CONDITION (b_2)
INFLOW Q_s gals/min		200 ---- ($t<0$) 250 ---- ($t>0$)	$200+50\sin\left[180°\frac{t-0}{10}\right]$	$200+50\sin\left[180°\frac{t-0}{20}\right]$
LEVEL V feet	100 gals/foot ----	$5+10.61e^{-0.0333t}\cos\left[180°\frac{t-33.3}{66.7}\right]$	$5+1.61\sin\left[180°\frac{t-4.31}{10}\right]$	$5+3.30\sin\left[180°\frac{t-7.10}{20}\right]$
	400 " ——	$5+4.52e^{-0.00833t}\cos\left[180°\frac{t-56.8}{113.6}\right]$	$5+0.401\sin\left[180°\frac{t-4.83}{10}\right]$	$5+0.819\sin\left[180°\frac{t-9.30}{20}\right]$
OUTFLOW Q gals/min	100 " ----	$250-61.24e^{-0.0333t}\cos\left[180°\frac{t+13.0}{66.7}\right]$	$200+10.86\sin\left[180°\frac{t-4.81}{10}\right]$	$200+23.12\sin\left[180°\frac{t-9.60}{20}\right]$
	400 " ——	$250-52.22e^{-0.00833t}\cos\left[180°\frac{t-10.6}{113.6}\right]$	$200+2.70\sin\left[180°\frac{t-5.33}{10}\right]$	$200+5.73\sin\left[180°\frac{t-11.27}{20}\right]$
TIME-BOUNDARIES		($t>0$)	($-\infty<t<\infty$)	($-\infty<t<\infty$)

FIG. 6 PROPORTIONAL-PLUS-FLOATING LEVEL CONTROLLER; CASE II



NUMERICAL SOLUTIONS		CONTROL CONSTANTS: $b = 60$ ft, $r = 0.15$ 1/min, $k = 200$ gals/min		
	$A = 5.88d^2$	CONDITION (a)	CONDITION (b_1)	CONDITION (b_2)
INFLOW Q_s gals/min		200 ---- ($t<0$) 250 ---- ($t>0$)	$200+50\sin\left[180°\frac{t-0}{10}\right]$	$200+50\sin\left[180°\frac{t-0}{20}\right]$
LEVEL V feet	100 gals/foot ----	$5+7.28e^{-0.0167t}\cos\left[180°\frac{t-22.9}{45.7}\right]$	$5+1.67\sin\left[180°\frac{t-4.65}{10}\right]$	$5+3.86\sin\left[180°\frac{t-8.34}{20}\right]$
	400 " ——	$5+3.64e^{-0.00417t}\cos\left[180°\frac{t-45.7}{91.4}\right]$	$5+0.414\sin\left[180°\frac{t-4.92}{10}\right]$	$5+0.837\sin\left[180°\frac{t-9.64}{20}\right]$
OUTFLOW Q gals/min	100 " ----	$250-61.84e^{-0.0167t}\cos\left[180°\frac{t+9.2}{45.7}\right]$	$200+6.15\sin\left[180°\frac{t-6.06}{10}\right]$	$200+17.78\sin\left[180°\frac{t-13.20}{20}\right]$
	400 " ——	$250-53.21e^{-0.00417t}\cos\left[180°\frac{t+10.1}{91.4}\right]$	$200+1.53\sin\left[180°\frac{t-6.33}{10}\right]$	$200+3.86\sin\left[180°\frac{t-14.50}{20}\right]$
TIME-BOUNDARIES		($t>0$)	($-\infty<t<\infty$)	($-\infty<t<\infty$)

FIG. 7 PROPORTIONAL-PLUS-FLOATING LEVEL CONTROLLER; CASE III



NUMERICAL SOLUTIONS		CONTROL CONSTANTS: $b = 60$ ft, $r = 0.025$ 1/min, $k = 200$ gals/min		
	$A = 5.88d^2$	CONDITION (a)	CONDITION (b_1)	CONDITION (b_2)
INFLOW Q_s gals/min		200 ---- ($t<0$) 250 ---- ($t>0$)	$200+50\sin\left[180°\frac{t-0}{10}\right]$	$200+50\sin\left[180°\frac{t-0}{20}\right]$
LEVEL V feet	100 gals/foot ----	$5+21.21e^{-0.0167t}\cos\left[180°\frac{t-66.9}{133.7}\right]$	$5+1.60\sin\left[180°\frac{t-4.66}{10}\right]$	$5+3.22\sin\left[180°\frac{t-8.62}{20}\right]$
	400 " ——	$5+9.05e^{-0.00417t}\cos\left[180°\frac{t-113.7}{227.3}\right]$	$5+0.399\sin\left[180°\frac{t-4.92}{10}\right]$	$5+0.801\sin\left[180°\frac{t-9.66}{20}\right]$
OUTFLOW Q gals/min	100 " ----	$250-117.3e^{-0.0167t}\cos\left[180°\frac{t+48.1}{133.7}\right]$	$200+5.34\sin\left[180°\frac{t-4.91}{10}\right]$	$200+10.86\sin\left[180°\frac{t-9.43}{20}\right]$
	400 " ——	$250-67.42e^{-0.00417t}\cos\left[180°\frac{t+53.2}{227.3}\right]$	$200+1.33\sin\left[180°\frac{t-5.17}{10}\right]$	$200+2.70\sin\left[180°\frac{t-10.67}{20}\right]$
TIME-BOUNDARIES		($t>0$)	($-\infty<t<\infty$)	($-\infty<t<\infty$)

FIG. 8 PROPORTIONAL-PLUS-FLOATING LEVEL CONTROLLER; CASE IV

$$AR_e(Q - Q_m)'' + (Q - Q_m)' + r(Q - Q_m) = r(Q_s - Q_m) + (Q_s - Q_m)' \quad \ldots [19]$$

From the integration of these differential equations, we may determine the response of the level V and of the outflow Q when sudden changes occur in the inflow Q_s. The curves under condition (a) in Figs. 5, 6, 7, and 8 represent the response of the level and the outflow following the usual sudden disturbance, when various magnitudes of proportional band b and reset constant r are assumed for the proportional-plus-floating level controller. The numerical equations from which these curves were computed are included in the figures.

* * *

In operational form, Equations [18] and [19] become

$$V - V_m = \frac{R_e p}{r + p + AR_e p^2} \cdot (Q_s - Q_m) \quad \ldots [20]$$

$$Q - Q_m = \frac{r + p}{r + p + AR_e p^2} \cdot (Q_s - Q_m) \quad \ldots [21]$$

When a sudden change occurs in the inflow Q_s, the response of the level V and of the outflow Q may be found by classical methods from Equations [18] and [19] or by standard operational methods from Equations [20] and [21]. Under equilibrium conditions, after all transients have faded out and all derivatives have become zero, it is evident that $V = V_n$ and that $Q = Q_s$. Thus the level will ultimately balance out at the desired value for all of the values of flow.

As before, the response under permanently oscillatory conditions may be found by setting $p = i\pi/h$ in the operators of Equations [20] and [21]. For level and outflow, respectively, the operators yield the following complex expressions, where $G = \pi AR_e/h$ and $H = rh/\pi$

$$\frac{[1 - i(G - H)]R_e}{1 + (G - H)^2}$$

$$\frac{1 - H(G - H) - iG}{1 + (G - H)^2}$$

From these complex expressions, the amplitude ratios and the relative time lags may be found by methods already described. This information is completely descriptive of the behavior of level and outflow when the inflow is assumed to follow a given permanent harmonic oscillation about a constant mean value. Thus we find the following

$$\frac{\text{Ampl. of } V}{\text{Ampl. of } Q_s} = \frac{R_e}{\sqrt{1 + (G - H)^2}}$$

$$\text{Lag of } V \text{ versus } Q_s = (h/\pi) \tan^{-1} (G - H)$$

$$\frac{\text{Ampl. of } Q}{\text{Ampl. of } Q_s} = \sqrt{\frac{1 + H^2}{1 + (G - H)^2}}$$

$$\text{Lag of } Q \text{ versus } Q_s = (h/\pi) \tan^{-1} [G/(1 - GH + H^2)]$$

* * *

As in the previous cases, the general equations for V and Q under the cyclic conditions (b_1) and (b_2) may be written

$$V = V_n + A_v \sin\left[180° \frac{t - T_v}{h}\right] \qquad [7]$$

$$Q = Q_m + A_q \sin\left[180° \frac{t - T_q}{h}\right] \qquad [8]$$

The equations for the constants in these equations are again taken from the operational development, as outlined, and can be given as

$$A_v = \text{Ampl. of } V = \frac{R_e \times (\text{Ampl. of } Q_s)}{\sqrt{1 + (G - H)^2}}$$

= level variation in feet

$$A_q = \text{Ampl. of } Q = (\text{Ampl. of } Q_s) \times \sqrt{\frac{1 + H^2}{1 + (G - H)^2}}$$

= outflow variation in gpm

$$T_v = \frac{h}{2} \frac{\tan^{-1} (G - H)}{90°}$$ = time in minutes by which cycles of V lag behind cycles of Q_s

$$T_q = \frac{h}{2} \frac{\tan^{-1} [G/(1 - GH + H^2)]}{90°}$$ = time in minutes by which cycles of Q lag behind cycles of Q_s

in which $G = 3.14 \dfrac{AR_e}{h}$ and $H = 0.318rh$

Compared to those for the case of proportional control, these equations have become more complex, due to the inclusion of the reset constant r, but it is interesting to note the nature of the changes and the fact that the equations will reduce to those for proportional control on substituting $r = 0$. An important difference is that the cycles of the level and those of the outflow are no longer in phase with one another.

In Figs. 5 through 8 are shown numerical and graphical examples of the application of the general formulas obtained. Four different cases are taken, covering four different sets of adjustments incorporated in the proportional-plus-floating level controller. Otherwise the conditions assumed are the same as were those for the previously considered system. The values of (effective) proportional band b and of reset constant r assigned in the various cases are given in tabular form as follows:

	Proportional band (b)	Reset constant (r)
Case I (Fig. 5)	30 Ft (300 per cent)	0.15 Inverse min
Case II (Fig. 6)	30 Ft (300 per cent)	0.05 Inverse min
Case III (Fig. 7)	60 Ft (600 per cent)	0.15 Inverse min
Case IV (Fig. 8)	60 Ft (600 per cent)	0.025 Inverse min

With proportional bands wider than 100 per cent, it is necessary to consider the effect of sustained changes in the inflow. Figs. 5 through 8, under condition (a), show the response of the level and the outflow following a sudden sustained change in the inflow. After such a change, the duty of the installation is to bring the outflow as smoothly as possible into equality with the new inflow, and also to return the level to the normal value. The more time allowed for these operations, the better the duties of smoothing may be performed. Shown in the figures are the initial portions of the level and outflow transients following the instantaneous disturbance of condition (a).

In case III, Fig. 7, and in case IV, Fig. 8, the size of the sudden change in the inflow is such that (for the low-capacity vessel) the level reaches its high limit in about $16^1/_2$ and $12^1/_2$ min, respectively. The practical design of proportional-plus-floating control instruments, capable of utilizing such excessive magnitudes of proportional band, must include mechanical means for decreasing the effective "throttling range" in the immediate region of the high and low limits. Details of such mechanism and of its operation are discussed in the paper (1) by J. B. McMahon.

It is readily evident that the general equations for the cyclic conditions can be put to practical use in determining many of the important relationships in actual averaging-control installations. The substitution of known factors permits concrete determination of other factors or relationships between them, as in connection with (a) vessel areas to give desired smoothing of the outflow for various types of instruments; (b) periods and magnitudes of oscillation which could be tolerated in existing installations; (c) economic considerations of instrument investment against increased equipment costs, etc.

The particular process and conditions selected for consideration in this paper illustrate many of the common circumstances met with in commercial installations. Much general information can be gained by exploring the hidden "intelligence" of these equations. Space will only permit us a brief discussion of one series of observations which appears to shed light on the nature of desirable instrument adjustments.

Under case I, Fig. 5, the response for the smaller vessel and the longer period of oscillation shows that both the level and the outflow variations have been increased over those for the 100 per cent throttling control, Fig. 4, although the proportional band has been trebled. Furthermore, the level variation has been increased beyond that of Fig. 2, where the outflow was perfectly constant. This circumstance is a result of the fact that the reset

constant is too great for the existing conditions of vessel area and period of oscillation.

In case II, Fig. 6, the reset constant is made one third of its value in case I, Fig. 5, but the proportional band is kept at the same value. For the same vessel area and period of oscillation, a marked improvement is discernible in the variation of the outflow. Some reduction is also made in the level variation, although this variation is still in excess of that for the ideal case of Fig. 2.

In case III, Fig. 7, the proportional band is increased to 6 times that used in the 100 per cent throttling control, Fig. 4, or to twice that used in cases I and II, Figs. 5 and 6, but the same reset constant is applied as in case I. A still further reduction in outflow variation is obtained, but the variation of the level is greater than that of case II, Fig. 6. This means that the reset constant could still be reduced.

In case IV, Fig. 8, the same proportional band is used as in case III, Fig. 7, but the reset constant is reduced to one sixth that of case III, Fig. 7, or to one half that of case II, Fig. 6. Another marked improvement is evident in the smoothing of the outflow variations, as well as a further reduction in those of the level. It is interesting to observe, in this case, for both of the vessel areas and for both periods of inflow oscillation, how closely the magnitude and lag of the level variations have approached those seen under the ideal case of constant outflow, Fig. 2.

Even this brief introductory treatment and these few observations seem to have established certain of the characteristic properties of the proportional band and reset adjustments in connection with automatic averaging control. The authors feel that a considerable amount of investigation remains to be done in this direction and that such work could be of tremendous practical value to those in industry who are faced with averaging-control problems. We have only endeavored to point out a possible approach. Even from the quantitative material presented here, tables could be compiled or charts prepared which would facilitate the engineering of installations.

Acknowledgment

Acknowledgment is wholeheartedly made to all those who have contributed, through their experiences and discussion, to the developments here presented. Appreciation is especially due to R. A. Rockwell of The Foxboro Company for his valuable assistance and advice concerning a rational development of the substance of the text. The authors are grateful also to many members of the Society, and in particular to P. W. Keppler for his encouragement in the preparation of the paper.

BIBLIOGRAPHY

1 "Control of Liquid Level in Vessels Under Pressure," by J. B. McMahon, *Industrial and Engineering Chemistry*, vol. 29, 1937, pp. 1219–1224.

2 "Quantitative Analysis of Process Lags," by C. E. Mason, Trans. A.S.M.E., vol. 60, 1938, pp. 327–334.

3 "Mathematical Methods in Engineering," by Th. von Kármán and M. A. Biot, McGraw-Hill Book Company, Inc., New York, N. Y., 1940.

4 "Fourier Integrals for Practical Application," by G. A. Campbell and R. M. Foster, Bell Telephone Laboratories, New York, N. Y., Monograph B-584, September, 1931.

5 "Operational Methods in Mathematical Physics," by H. Jeffreys, Macmillan Company, New York, N. Y., second edition, 1931.

6 "Automatic Control in the Presence of Process Lags," by C. E. Mason and G. A. Philbrick, Trans. A.S.M.E., vol. 62, May, 1940, pp. 295–308.

Discussion

E. S. Smith.[7] From the paper it appears that the use of a proper controller makes it possible to reduce simultaneously both the outflow and level variations, a conclusion which warrants discussion since the level variations in the reservoir are the means for reducing the outflow variations. The following consideration of phase differences shows that this conclusion is correct.

Let us investigate the phase angle or lag with a harmonic variation of inflow to a tank or reservoir for several cases, some of which were noted in the paper. A simplifying assumption is that changes in level do not appreciably affect either the inflow or the outflow, i.e., that there is negligible self-regulation.

Case I. *Outflow Valve Held Steady.* The outflow rate is held steady at the average rate of inflow. The level variations lag 90 deg behind those of the inflow and their amplitudes vary inversely with the area of the tank. Let

q_1 = inflow variation from mean rate, cfs
q_2 = outflow variation from mean rate, cfs
h = level variation from mean position, ft
t = time, sec

$$q_1 = \sin t \quad \text{[22]}$$

$$h = \int \frac{1}{A} \sin t \, dt = -\frac{1}{A} \cos t = \frac{1}{A} \sin\left(t - \frac{\pi}{2}\right) \quad \text{[23]}$$

or h lags $\frac{\pi}{2}$ sec (for a period of 2π sec) or 90 deg behind q_1. This situation is shown in Fig. 9 (freehand curves being used as adequate for this discussion).

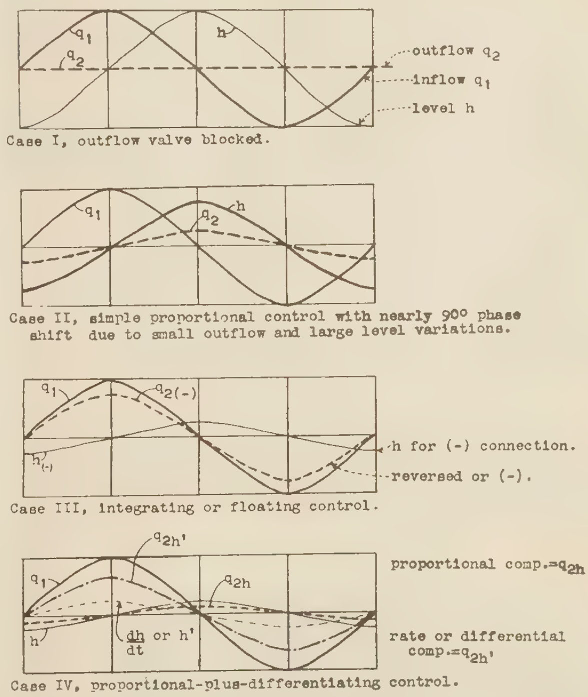


Fig. 9 Diagrams Illustrating Phase Relations With Various Controllers, Cases I–IV

To hold the level constant requires that the outflow be controlled at a point exactly equal to and in phase with the inflow. Then there is zero instantaneous net inflow or difference of inflow and outflow at any moment. Since there would then be no change in the level, it cannot be used to control the outflow to regulate perfectly the level; instead, for perfect level control, the

[7] Patent Agent, C. J. Tagliabue Mfg. Co., Brooklyn, N. Y. Mem. A.S.M.E.

outflow can only be controlled by an inflow meter. However, in practice the level can be controlled very well by the use of high-sensitivity controllers which are governed by the level.

Case II. *Proportional Control With Large Level Variations.* The outflow is varied in phase with the level in simple corresponding (i.e., proportional or throttling) control. As long as there is negligible effect of the outflow on the phase of the level, both the level and the outflow will lag 90 deg behind the inflow.

Case II*a*. *Proportional Control With Large Outflow Variations.* As in case II, the outflow is varied in phase with the level. However, as shown in Fig. 10 of this discussion, the outflow changes so

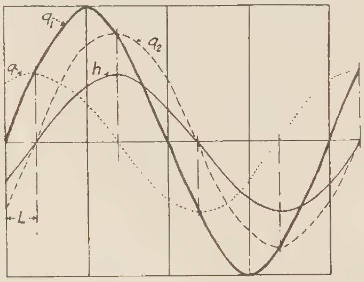


Fig. 10 Simple Proportional Control With Less Than 90 Deg Phase Shift Due to Large Outflow and Small Level Variations, Case II*a*

much that it causes the instantaneous net inflow q (which equals $q_1 - q_2$) to lead the inflow. Both the level and the outflow lag 90 deg behind the net inflow q and lag behind the inflow by L, which is appreciably less than 90 deg.

If an unconventional device be used so that the outflow lags 90 deg behind the inflow and the amplitude of both flows is identical, it is evident from inspection that the net inflow q would lead the inflow by 45 deg and the level h would lag the inflow by 45 deg. Of course this device is different from the proportional controller which has the outflow in the same phase as the level.

With proportional control, the lag depends upon the ratio b of the flows q_2 to q_1 and the area of the reservoir. With unity area of reservoir and the size of the outflow valve such that the amplitude of the outflow variation is in the ratio b to that of the inflow variation, and b is less than unity, the net inflow q is

$$q = q_1 - q_2 = \frac{dh}{dt} \qquad [24]$$

The term q_2 is directly and continuously proportional to h in a simple proportional controller and, from an axiom to be stated later, is a simple harmonic function of the same period as q_1 so that

$$\frac{dh}{dt} = \sin t - b \sin (t - L) \qquad [25]$$

from which h and L can be obtained by integration, taking vectorial account of the following considerations. Since the lag L is, loosely, the time required for the level to change enough to operate the outflow valve upon a change of the inflow rate, the lag increases with an increase of the area and of the amplitude of the inflow variation. An increase in the size of the outflow valve gives an increase in the outflow variations and causes a decrease in the amplitude of the level variations, and hence of the lag.

Case III. *Integrating or Floating Control.* A brief consideration of the effect of adding a differentiating component best shows the limitations of an integrating control. If the regulator be provided with a differentiating device so that the outflow varies continuously with the rate of change of the level, each outflow variation obviously can be brought more nearly into phase with each inflow variation and the level variation may be made as small as will serve to actuate the differentiating device so that it governs the controlling of the outflow. The differentiating device may be a leak-shunted differential bellows, a disk-driven threaded roller, or any equivalent means.

Offhand, an integrating device with a reverse-acting outflow valve might seem to be an equivalent of a differentiating device, since the valve lags 90 deg behind the level variations (which lag up to 90 deg behind the inflow variations) and, hence, 180 deg behind the inflow variations, neglecting the effect of outflow variations upon the phase of the level variations. For this case (Fig. 9, case III), the change of sign brings the outflow variations into phase with the inflow variations. With this change of sign, with inflow $q_1 = \sin t$ as in Equation [22], and with the sensitivity equal to the area A, to have the amplitude of the outflow nearly equal to that of the outflow so that the net inflow is nearly in phase with q_1, then following Equation [23]

$$q_2 = \int -Ahdt = \int -A \frac{1}{A} \sin \left(t - \frac{\pi}{2}\right) dt = -\sin (t - \pi)$$
$$= \sin t \qquad [26]$$

Since a sudden change of inflow causes a serious lag and possibly a controlling impulse in the wrong direction, a reverse-acting integrating device is not a practical equivalent of the differentiating device. With a direct-acting outflow valve of the same sensitivity, the changes of level are much larger but the device is reliable.

The servomotor of a floating-type controller acts as an integrating device. An integrating effect is produced by metering lag with a simple corresponding (or proportional) regulator. Some years ago, the writer was required to make such a level regulator work on the settling basin of the waterworks at Red Lion, Pa., where the required sensitivity was high and there was considerable metering lag with the regulator as originally installed. After testing the device, a mechanic had hooked the valve up backwards so that level control within fairly narrow limits was obtained for a short time. Occasionally, however, the regulator would open wide or close off the outflow valve so that this empirical "solution" was, of course, unsatisfactory since closing this valve shut off the town's entire water supply. In this case, the regulator operated satisfactorily as soon as the metering lag was greatly reduced and the outflow valve was made direct-acting.

Case IV. *Proportional-Plus-Differentiating Control.* The customary types of reset regulators include a proportional follow-up of some sort to insure initial correspondence and, hence, to prevent overtravel of the final controlling element following a sudden change of the sensed or measured variable. They also include a differentiating device to advance the phase of the controlling element toward that of the pertinent variable. In addition to the metering lag, some lag is bound to exist with such regulators for level control since there is always an appreciable effect of the follow-up component. In other words, while the proportional follow-up component does not produce lag as regards the following of level variations, it does as regards the following of inflow variations. From this it is evident that an appreciable differentiating component is essential for the control of level within minimum limits.

In the common case of "averaging level control," the outflow variation is kept within minimum limits and as much "slack" as possible is taken up by level changes within the capacity of the tank or reservoir. Even though, offhand, it might seem to be impossible to alter the adjustment of a reset-type controller to reduce the variations of both the outflow and the level, this can be accomplished as shown in Fig. 9, case IV, by changing the phase

relation between the variations of the level and of the outflow. This follows from the vectorial axiom that the sum of harmonic curves of the same period but of different phase produces a resultant simple harmonic curve of an intermediate phase and an amplitude which depends upon the algebraic sum of the components, i.e., when the phase difference between the component curves is from 0 to 90 deg, the resultant amplitude is increased but, when it is from 90 to 180 deg, the resultant amplitude is decreased. No further mathematical analysis is believed to be necessary to establish the functioning of such a phase change.

In regulation, generally, the function of reset is simply to destroy gradually the momentary correspondence, by which the follow-up gives stability, in order to restore asymptotically the sensed variable precisely to its set value. In the regulation of level (or for other cases involving single-capacity systems), the reset has the particular function of determining the inflow rate from the rate of change of the level in addition to that of returning the level to the same point. Such a reset acts much as does an inflow meter. This point was originally brought out in an A.S.M.E. paper prepared by R. P. Lowe and E. S. Smith on level control by an asymptotic reset controller, which paper was read at the Providence meeting of October, 1938, and showed that such a controller acted much as does one which follows the net inflow or difference between separately metered values of the inflow and outflow rates.

Where the initial response is less than the total response ultimately required, the reset must act to move the outflow valve further in the same direction; and, where the initial response is more than the total response ultimately required, the reset must act to move the outflow valve in the reverse direction from that of its initial response. However, in both cases the reset will return to rest at a particular point and act to restore the level to the same point. Averaging level control ordinarily has an inadequate initial response, and hence must have the delayed response in the same direction as the initial response.

The value of the paper and the excellence of its mathematical treatment are self-evident. Such mathematics is of course, necessary for a definitive treatment of the subject. However, the curtailed mathematical presentation of this discussion is submitted without apology in an effort to place the treatment of this subject on a plane which is completely understandable to those who, like the writer, work with it only occasionally. While this discussion has adopted the authors' convention of a steadily hunting inflow to allow lag to be expressed as a phase angle, it is appreciated that a yet simpler or less artificial analysis may be possible through the use of the authors' sudden single change of the inflow instead of the cyclical change.

The references at the end of this discussion may prove helpful in a study of the subject. In addition to those mentioned, the works on oscillating phenomena in connection with heat transfer by Ivanoff and by De Juhasz are well known. The treatment of the subject in all the references cited, like that of the paper itself, is more involved than the writer's which is intended to be limited to the effect of phase shifts in level controlling.

REFERENCES

"Heaviside's Operational Calculus as Applied to Engineering and Physics," by E. J. Berg, eight installments, *General Electric Review*, vol. 30, 1927, pp. 586–589; vol. 31, 1928, pp. 93–96, 143–146, 212–222, 267–278, 395–398, 444–451, 504–509.

"On the Oscillations of Certain Electrical or Mechanical Systems Due to a Periodic Impressed Force," I. Herlitz, *General Electric Review*, vol. 25, 1922, pp. 686–689.

"Design Factors Controlling the Dynamic Performance of Instruments," by C. S. Draper and G. P. Bentley, Trans. A.S.M.E., vol. 62, July, 1940, pp. 421–432.

"Elementary Theory of Automatic Temperature Control," by C. O. Fairchild, *Instruments*, Nov., 1940, pp. 334–339.

E. W. Yetter[8] and J. C. Peters.[9] In a previous paper by A. F. Spitzglass,[10] the feasibility of calculating the results to be expected in applying floating or proportional position control to a single-capacity process was demonstrated. The authors have now extended this to proportional-plus-floating control, which mode they show to be advantageous for control of level when it is desired to keep the outflow as steady as is possible without permitting undue variations in the level.

The authors' consideration of the effect of a sine-wave disturbance of the inflow has suggested to the writers the possible advantage of considering this as an electrical problem by mathematical analogy. Such analogy has been found to be very useful in other branches of applied physics, because of the high degree of development of mathematics applied to electrical problems.

It is found that, for the case of proportional-plus-floating control and a sine-wave disturbance of the inflow, the ordinary mathematics of alternating currents suffices. The nature of this approach will be briefly indicated since it may prove useful in a further development of the subject. In what follows, the readily obtainable solutions for phase angles will be omitted because it appears that they have little or no practical significance in this connection.

The combined process-and-control equation may be written in the form

$$A\ddot{x} + k_p\dot{x} + k_f x = \dot{q}_1 \ldots\ldots\ldots\ldots\ldots [27]$$

where x = deviation of level from set point, ft
A = area of surge tank, sq ft
k_p = constant for proportional control, cfm per ft
k_f = constant for floating control, cfm per ft per min
q_1 = inflow rate cfm

The dots indicate time derivatives.

An electrical equation analogous to Equation [27] is

$$L\ddot{i} + R\dot{i} + \frac{1}{C}i = \dot{e}_1 \ldots\ldots\ldots\ldots\ldots [28]$$

where i = current
L, R, and C are electrical inductance, resistance, and capacitance, respectively
e_1 is the impressed voltage

This is the equation of the simple R, L, C circuit shown in Fig.

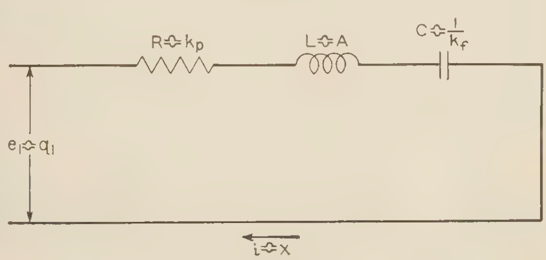


Fig. 11 Diagram Showing Electrical Analogy for Case of Proportional-Plus-Floating Control

11 of this discussion. Equivalents from Equation [27] are indicated in Fig. 11. The steady-state solution for the electrical case when

$$e_1 = E_1 \sin \omega t$$

[8] Research Department, Leeds and Northrup Company, Philadelphia, Pa.

[9] Research Department, Leeds and Northrup Company, Philadelphia, Pa. Mem. A.S.M.E.

[10] "Quantitative Analysis of Single-Capacity Processes," by A. F. Spitzglass, Trans. A.S.M.E., vol. 62, no. 1, Jan., 1940, p. 51.

is

$$I = \frac{E_1}{\sqrt{R^2 + \left(\omega L - \frac{1}{\omega C}\right)^2}} \qquad [29]$$

The amplitude for level variations is, by analogy

$$X = \frac{Q_1}{\sqrt{k_p^2 + \left(\omega A - \frac{k_f}{\omega}\right)^2}} \qquad [30]$$

where $\omega = \frac{2\pi}{T}$ and T = period of sine-wave disturbance, min.

(Capital letters uniformly represent amplitudes of sine-wave variations of quantities represented by corresponding small letters.)

The solution for the amplitude Q_2, of the variation in outflow, q_2 is obtained by noting that this is merely the result of the mode of control working in accordance with the variations of level given by Equation [30]. It is, therefore, the result of impressing the current i on a circuit representing the law of control. The output flow is analogous to the voltage which appears across this circuit, as a result of the impressed current. The law of the controller is

$$k_p\dot{x} + k_f x = \dot{q}_2 \qquad [31]$$

An analogous electrical equation is

$$R\dot{i} + \frac{1}{C}i = \dot{e}_2 \qquad [32]$$

This is the equation of the series R, C circuit shown in Fig. 12. Equivalents from Equations [31] and [32] are indicated in Fig. 12.

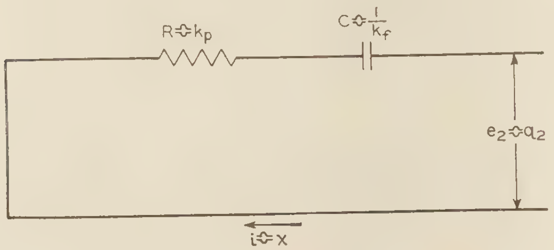


Fig. 12 Diagram of Electrical Circuit Analogous to the Law of the Controller

The impressed current is $i = I \sin \omega t$ and the amplitude of voltage e_2 is given by

$$E_2 = \sqrt{R^2 + \left(\frac{1}{\omega C}\right)^2}\, I \qquad [33]$$

Substituting for current I its value from Equation [29] and putting the result in the form of the ratio of the two voltages gives

$$\frac{E_2}{E_1} = \sqrt{\frac{R^2 + \left(\frac{1}{\omega C}\right)^2}{R^2 + \left(\omega L - \frac{1}{\omega C}\right)^2}} \qquad [34]$$

The ratio of the amplitudes of the sine waves of output and input flow is then, by analogy

$$\frac{Q_2}{Q_1} = \sqrt{\frac{k_p^2 + \left(\frac{k_f}{\omega}\right)^2}{k_p^2 + \left(\omega A - \frac{k_f}{\omega}\right)^2}} \qquad [35]$$

Equations [30] and [35] are equivalent to equations found in the paper under discussion. Written in this form, it is immediately evident that the equivalent of an electrical or mechanical "resonant" effect may be obtained by varying the period of the input flow. A maximum amplitude of level variation will take place when

$$\omega A - \frac{k_f}{\omega} = 0$$

and the period is

$$T = 2\pi\sqrt{\frac{A}{k_f}} \qquad [36]$$

It will be noted that the equation for E_2, obtained from Fig. 12, is also obtainable from Fig. 11 as the voltage across the resistance and capacitance in series. A single circuit may therefore be used to write equations for both level and outflow.

Authors' Closure

Due to the absence of controversial issues in the discussions to this paper, the authors feel that there is no real necessity for detailed supplementary comments. Their thanks are due to the separate discussers, however, for the elaborations which they have contributed from their respective viewpoints.

With regard to the practicality of the approach taken in the paper, it might not be inappropriate to include here in the closure an example in which the assumed condition of the inflow is different from either of the two "ideal" types of disturbance postulated in the paper. Such a condition is shown in Fig. 13 of this closure, in which the inflow, after having existed for an indefinite period under equilibrium with level and outflow, begins suddenly to execute a periodic sawtooth variation involving an infinitely steep wave front. This corresponds to the practical assumption made to represent a circumstance which was encountered in an actual industrial application. The resulting level and outflow behavior, shown in Fig. 13, was calculated on the basis of the same control constants and so on as were assumed in Fig. 6 (Case II) of the paper itself. The behavior of the variables seen in Fig. 13, it may be noted, involves both transient and steady-state periodic variations. It is interesting to observe that the response of level and outflow to this special disturbing condition might almost have been predicted from the response with the same equipment to the idealized step and periodic disturbances.

Included in the discussion by E. S. Smith there is the following quite elegant single-sentence definition of reset: "In regulation generally, the function of reset is simply to destroy gradually

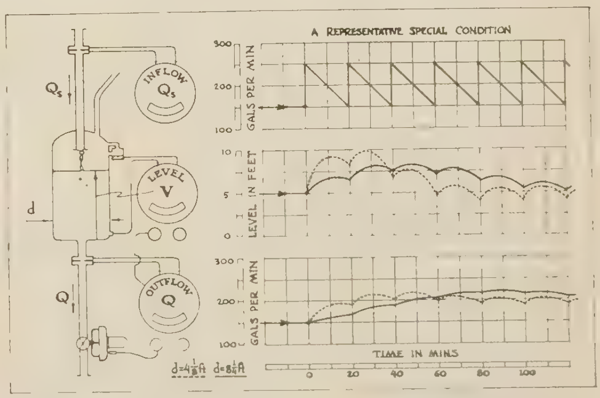


Fig. 13 Result for Special Disturbance; Same Control System as in Fig. 6, Case II of Paper

the momentary correspondence[11] . . . . in order to restore . . . the sensed variable precisely to its set value." This definition, carefully framed so as to apply in general, is worthy of note at a time when terminology is still in a fluid state. If anything there has been too little tendency to generalize—apparent in the literature of automatic control (or automatic regulation). Until placed on an independent footing and freed of all specializing concepts, this subject will never attain recognition as a branch of knowledge in its own right as we know it deserves to be.

The discussion of E. W. Yetter and J. C. Peters is devoted principally to electrical analogs of the hydraulic system and controlling equipment treated in the paper. Such analogy may lead to a quicker perception, by many electrical engineers, of the dynamic phenomena described, but it may be remarked that the process of analogy is traditionally the other way around; hydraulic analogs serving to make more tangible the functional performance of electric-circuit elements and circuits. A great many engineers we feel sure, mechanical engineers for example, would not agree to the relatively higher development of mathematics in electrical as compared with other technical fields. Then too, the rudimentary sort of wave mechanics which is represented in the standard alternating-current theory is not limited in applicability to electric circuits. It is significant that the operational treatment, although actually no more involved than any other, is versatile to the extent that it yields the wave-mechanics, or frequency "spectrum," solution if interpreted one way and the complete transient solution if interpreted another, both solutions coming down from the same operational form.

An entire series of mechanical, thermal, pneumatic, and electrical analogs may be placed in correspondence with, and will adequately represent, the hydraulic system assumed in the paper. Beyond those introduced in the discussion, a number of other electrical analogs are possible and could suffice as models for the hydraulic prototype. A mechanical interpretation, in which the flows become displacements and the capacitance of the vessel is replaced by a massive body, can be traced out in complete detail and is easy to visualize. Under this latter analogy the problem of automatic control is seen as a true problem in shock-absorbing.

[11] The authors would add the words "or proportionality."

Graphical Methods for Plotting Time-Speed-Distance Curves for Railway Trains

By A. I. LIPETZ,[1] SCHENECTADY, N. Y.

The paper reviews briefly the interest displayed some years ago in Europe, particularly in Russia, in graphical methods for plotting time-speed-distance curves for railway trains and develops methods devised by the author for plotting such curves. Analytical methods and the graphical method are compared and the results are tabulated, and the author's methods are applied to data from runs of high-speed trains in this country.

About thirty years ago great interest was displayed in Europe, especially in Russia, in graphical methods by which speed versus time, or speed versus distance, or distance versus time for trains between two stations could be determined. Special methods had been worked out and were in use in some Russian railway offices; and the development of these methods became sort of a fad in which railway officials and young engineers vied with each other. Naturally, these developments were reflected in the Russian technical press. A well-known railway official and college teacher, Prof. G. V. Lomonossoff, and some of his pupils and friends became active in this game and greatly contributed to the current literature of that period (1, 2, 3, 4).[2]

Likewise, a similar interest was displayed in Germany, with a corresponding reflection in the German press (5, 6). Various methods had been developed in Germany, which later were reviewed in a symposium by Dittmann (7). In this work five methods were described.

In the United States, analytical methods are mainly in use, although in some cases graphical methods have been resorted to for auxiliary calculations (8, 9, 10).

It so happened that after the Russian revolution the men who were active in this development in Russia were scattered all over the world—Lomonossoff emigrated first to Germany and later to England; Chechott first to Poland and then to Persia (Iran); Lipetz, the author of this paper, to this country, and others to France, Germany, and elsewhere. Thus little has been published in English; most of the publications appeared in Russian, Polish, German, and French (11, 12, 13, 14). Owing to pressure of business and preparation of other articles, the author has not made his method known in this country, although it had been in exclusive use in Russia, partially in Poland, in Germany, under the name Lipetz-Strahl (12, p. 29), and France, where it was later used by Cremer (13) and others (24). In this country it is used by the author and some of his associates (H. Cregier and S. Slastenin) in calculations needed for designing and investigation of some high-speed locomotives in the offices of the American Locomotive Company. However, it has never been fully published, although it was mentioned and used as an illustration in the author's discussion of C. T. Ripley's paper (17). The latter omission is now corrected by the presentation of this paper.

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[2] Numbers in parentheses refer to Bibliography at end of paper.

Contributed by the Railroad Division and presented at the Annual Meeting, New York, N. Y., Dec. 2–6, 1940, of The American Society of Mechanical Engineers.

Integration of Equation of Train Movement

The time and running distance of a train under the influence of various forces applied to it are defined by the fundamental equation of the movement of the train. If a train, as a body with mass M, is covering an elementary distance ds under the influence of locomotive tractive effort T and train resistances R (forces which may be applied to different parts of the train, like friction of brake shoes to wheels, axle resistance to journals, air resistance to car bodies, gravity to centers of mass of cars and locomotives), the elementary change of energy dE in time dt on a distance ds is

$$dE = Mvdv = (T - R)ds$$

if v is the momentary and dv the differential of speed.

As every train, in addition to the translatory movement of its car bodies, has rotating parts (wheels, motors, gears), this equation should be amplified. Let the polar moments of inertia of each rotating part around its axis be I; the increment of energy is then (12, p. 10; 16, p. 10)

$$dE = vdv \times \left(M + \sum \frac{I}{\rho^2}\right)$$

Hence

$$\left(M + \sum \frac{I}{\rho^2}\right) vdv = (T - R)ds \qquad [1]$$

This is the fundamental equation of the movement of a train. It is usually simplified by referring the members with moments of inertia of rotating parts $\sum \frac{I}{\rho^2}$, which has the dimension of mass, to the total mass of the respective equipment, locomotive and cars. Depending upon their dimensions, these ratios γ fluctuate from 0.04 to 0.30 for different equipment; for instance, for steam locomotives it is about 5 per cent, i.e.

$$\gamma = \sum \frac{I}{\rho^2} : M = 0.05$$

going up to 0.06 for high-speed steam locomotives with large wheels. In this formula I is the moment of inertia of every driving wheel and axle of the locomotive; ρ the respective outside radii of the wheels, and M, the mass of the locomotive. For electric locomotives with motors geared to the axles $\gamma = 0.3$ to 0.4, including all these parts. For loaded freight cars $\gamma = 0.03$; for empty freight cars $\gamma = 0.11$ (12, p. 10; 6, p. 142).

For the whole train the influence of the rotating parts is of secondary importance, for instance, for a train consisting of a steam locomotive of 300 tons and passenger cars of 500 tons, the ratio γ is

$$\gamma = \frac{300 \times 0.06 + 500 \times 0.04}{800} = 0.0475$$

For another sort of equipment, a light electric locomotive and loaded freight cars

$$\gamma = \frac{200 \times 0.35 + 1000 \times 0.03}{1200} = 0.0833$$

For heavy Diesel locomotives and streamline passenger cars

$$\gamma = \frac{460 \times 0.35 + 1250 \times 0.04}{1710} = 0.1233$$

The average of these three cases gives $\gamma = 0.0847$.

The case of empty freight cars has not been considered, as it is unlikely that empty freight cars would be transported by modern high-speed locomotives.

Equation [1] is written in absolute units of foot, pound, and second—see Lionel S. Marks, Mechanical Engineer's Handbook, first edition, 1916, page 73 (symbols in lower-case letters). For units which are customary in railroad engineering, capital letters are used; for miles of length, one mile equals 5280 ft; for speed V in miles per hour $V = \frac{3600}{5280} \times v = \frac{1}{1.467} \times v$, where v is in fps. For acceleration in miles per hour per second $A = \frac{1}{1.467} \times a$, where a is in fpsps; and for mass M in tons of 2000 lb of weight $= \frac{32.17}{2000} = \frac{1}{62.17}$ tons of weight.

For these latter units, a constant C must be introduced in the right side of Equation [1] (10, p. 19; 25, p. 49) equal to $C = \frac{1}{62.17} \times \frac{1}{1.467} = \frac{1}{91.18}$, if V is in miles per hour, t in seconds, $\frac{dV}{dt}$ in miles per hour per second, $(T - R)$ in lb of weight, and M in tons of weight.

Remembering that $v = \frac{ds}{dt}$ and canceling ds, Equation [1] will read

$$\frac{dV}{dt} = \frac{T - R}{91.18(1 + \gamma)M} \quad \text{[1a]}$$

where V is speed in miles per hour, and t time in seconds; $\frac{dV}{dt}$ acceleration in miles per hour per second.

Since γ for trains of different consist varies from 0.04 to 0.1233, with an average of 0.0847, the author assumed that Equation [1a] can, for average conditions, be rewritten

$$\frac{dV}{dt} = \frac{t - r}{98.69} \quad \text{[2]}$$

where $\frac{dV}{dt}$ is acceleration in miles per hour per second; t and r are tractive effort and train resistance in pounds per ton of their weight, and the coefficient corresponds to $91.18(1 + \gamma) = 91.18 \times 1.0847 = 98.69$, or approximately 100.

We should consider then the approximate formula

$$\frac{dV}{dt} = \frac{t - r}{100} \quad \text{[2a]}$$

as close to the average conditions of modern trains. However, if the consist and γ of the train are known in advance, a more accurate coefficient can be figured out and formula [1a] should be used instead of [2a].

If the time-speed curve, namely, speed versus time, is the one sought, then $\frac{dV}{dt}$ is the tangent to this curve. Equation [2] shows that the tangent is represented by a simple relation, the difference between tractive effort and train resistance in pounds per ton of train weight divided by 100 for American units, mile, hour (respectively, second), and ton. If curves of Fig. 1 represent tractive effort of the locomotive and R the train resistance versus speed V, then the ordinates of the shaded area $ABCDA$ represent in a certain scale the right-hand side of Equation [2]. The desired acceleration curve will be found if a curve is so built that the tangents to it are equal to the ordinates divided by 98.69 or 100, as the case may be.

Suppose that a train stands in a station and the locomotive starts to accelerate it from standstill on a level. The excess of tractive effort over the train resistance on a level at low speeds from zero will be represented by the shaded area, and under the influence of this difference of forces the train will be accelerated from zero speed until the balanced speed V_0 is reached (Fig. 1), at which point the two forces (tractive effort and train resistance)

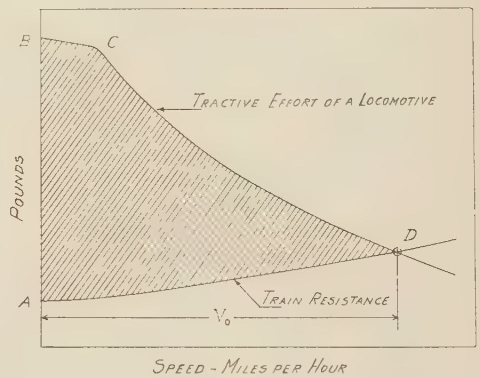


FIG. 1 TRACTIVE EFFORT AND TRAIN RESISTANCE

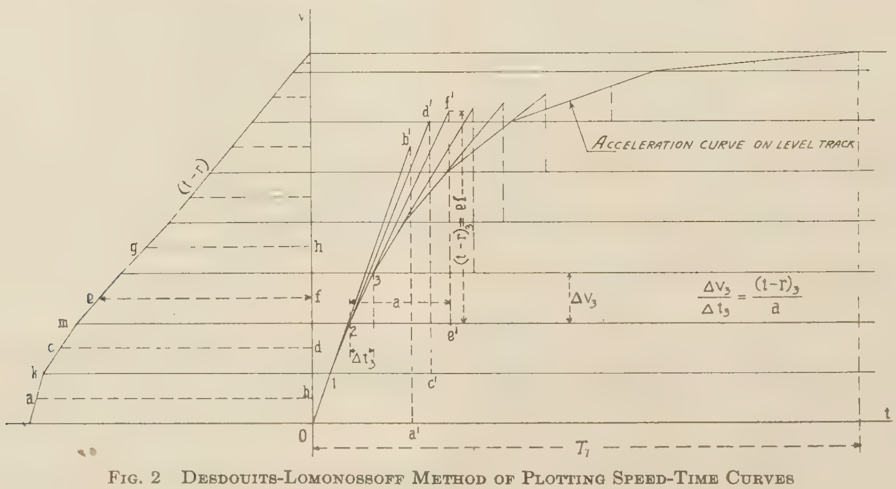


FIG. 2 DESDOUITS-LOMONOSSOFF METHOD OF PLOTTING SPEED-TIME CURVES

are equalized. The train will continue to move at the balanced speed V_0 as long as the forces (the locomotive and the profile) remain unchanged.

Acceleration curve from 0 speed to V_0 is the one we are trying to plot. After what has been said the plotting can be easily done. Draw first on the chart the difference by subtraction of the two curves t and r in pounds per ton of train weight[3] to a certain scale, say 5 lb per ton equals one inch (Fig. 2). Then mark intervals for speeds, say for every ten miles per hour from 0 to V_0, except the last one, which will be thus automatically defined; draw average ordinates corresponding to the middle of each interval (ab, cd, ef, gh, etc.); build for the first interval a right-angle triangle on base $0a^1$ = unit a to a certain scale for the constant a, for instance, 2 in., with the other side $a^1b^1 = ab$, the average ordinate of the first interval; draw the hypotenuse $0b^1$ through the first interval, 01, until it intersects with the ordinate $1K$ through the end of the interval. Build another triangle from point 1 with a base $1c^1$ = unit a for the constant (2 in.), and side $c^1d^1 = cd$, the average ordinate of the second interval. Draw the hypotenuse $1d^1$ through the second interval until it intersects $2m$, the end of the interval. In the same way build for the third interval with $e^1f^1 = ef$, and so on. Then the broken line 0 1 2 3 will be tangent to the acceleration curve at the prolongation of the ordinates through the middle points, ab, cd, ef, etc.

An important, if not the most important, operation in graphical calculations is the determination of scale, as the curve must be numerically read after it has been plotted. The length of the chart, for instance, T_1 in Fig. 2, represents to a certain scale the total time of the acceleration from 0 to balancing speed V_0 (Fig. 1).

The curve should represent Equation [2a], and the graphical construction is based on the ratios of increments denoted by differentials in formula [2a] and shown graphically in a certain scale on Fig. 2. If the Greek letters ν, θ, and φ represent the scales for speed, time, and force, respectively, namely, if

ν be the scale for speed, ν miles per hour = 1 in.
θ be the scale for time, θ seconds = 1 in.
φ be the scale for forces, φ lb per ton = 1 in.
a be a constant = 2 in.

Comparing the geometry of the construction of Fig. 2 with Equation [2a], it can be seen that

$$\frac{dV/\nu}{dt/\theta} = \frac{(t - r)/\varphi}{a}$$

or putting all members to one side

$$\frac{dV\theta a\varphi}{\nu dt(t - r)} = 1$$

But from Equation [2a]

$$\frac{dV \times 100}{dt(t - r)} = 1$$

Consequently

$$\frac{\theta a\varphi}{\nu} = 100$$

or

$$\theta = \frac{100\nu}{a\varphi} \qquad [3]$$

[3] Use any reliable formula for train resistance, including gravity, 20 lb per ton for each per cent of grade (20, p. 17), air resistance for speed (19), and curve resistance (20, p. 35).

This defines the scale for time θ.

In the chart, Fig. 2, representing the acceleration under the influence of forces in Fig. 1, the following scales for full-size chart have been chosen

ν = 10 mph = 1 in.
φ = 5 lb per ton = 1 in.
a (parameter) = 2 in.

Consequently

$$\theta = \frac{100 \times 10}{2 \times 5} = 100 \text{ sec} = 1.667 \text{ min} = 0.0278 \text{ hr} = 1 \text{ in.}$$

It should be remembered that while the speed is given in this case in miles per *hour*, the time scale is in *seconds*, because the acceleration, which is the basis of Equation [2a], is in miles per hour per *second.*

It should also be borne in mind that the coefficient 100 is an approximation for all high-speed trains, exactly equal to 100 for a case when γ in Equation [1a] corresponds to

$$91.18(1 + \gamma) = 100$$

or

$$\gamma = 0.0967$$

When γ is less, the coefficient is smaller than one hundred. A convenient case is when $\gamma = 0.0528$. In this case $91.18(1 + \gamma) = 96$ and the scale for time, when other scales are the same as just given, will be

$$\theta = \frac{96 \times 10}{2 \times 5} = 96 \text{ sec} = 1.6 \text{ min} = 1 \text{ in.}$$

In the speed-time curve the distance covered by the train during acceleration, say from zero to balanced speed V_0, is determined by the area of the curve and can be figured as the definite integral

$$s = \int_0^{V_0} Vdt$$

if we know the law of the curve V. Graphically, it can be determined as the area on squared paper or by planimetering. The scale for the area s can be easily determined, if the linear scales are known. If

ν is the scale for speeds in miles per hour = 1 in.
θ is the scale for time in hours = 1 in.

then σ, the scale for distances in miles, $= \nu\theta = 1$ sq in. For instance, in Fig. 2, $\nu = 10$ mph = 1 linear inch and $\theta = 0.0278$ hr per linear inch; then σ, the scale for distance, equals 10 mph times 0.0278 hr = 0.278 miles distance per square inch.

When a run over a certain profile is being investigated, it is necessary to watch constantly the changes in the profile, because if conditions change the speed and time are also changing. It is therefore necessary to resort frequently to planimetering of the areas while the curve is being plotted. This is the disadvantage of any speed-time curve plotting which prompted the author in 1911, when he was occupied with this kind of calculations, to develop the speed-distance method for plotting curves of this type (4, 18).

Method of Speed-Distance Curves

If we revert to Equation [1] we can write it in the following form, if we choose distance s in miles as the independent variable

$$\left(M + \sum \frac{I}{\rho^2}\right) \frac{vdv}{ds} = T - R$$

where, as before

M = total mass of the train, including locomotive and cars
I = moment of inertia of each rotating mass
ρ = outside radius of rotating wheels
v = linear speed of the train in miles per hour
s = distance covered by the train in miles
T and R = respectively, the tractive effort and the resistance applied as forces to different parts of the train

or

$$\frac{dv}{ds} = \frac{T - R}{M + \sum \frac{I}{\rho^2}} \frac{1}{v} = \zeta \frac{t - r}{v} \qquad [4]$$

where, also as before, t and r are the previous T and R referred to one ton of train weight and ζ is a constant, different, though, from the previous constants, 98.69 or 100. Constant ζ in Equation [4] should be

$$\frac{3600}{96} = 37.5$$

because speed is in miles per hour = 3600 sec.

Generally it should be

$$\frac{3600}{91.18(1 + \gamma)}$$

and accordingly as per the following table:

γ	C	ζ
.0967	100	36
.0823	98.69	36.5
.0528	96	37.5

The forces t and r are functions of speed and are the same as used before for the speed-time method. Their difference $(t - r)$ is shown in Fig. 2 for plotting the Desdouits-Lomonossoff curve. For the author's method it is shown in Fig. 3. Imagine that the acceleration curve is plotted as a function of distance and consider a certain element a_0b_0 of the acceleration curve. It is evident from Equation [4] that the tangents to the speed-distance (acceleration) curve should form the same angles with the distance axis $0S$, as the radii vectors of the force curve form with the speed axis $0V$ (Fig. 3) because the tangents to the acceleration curve are equal to $\frac{dv}{ds}$, while the tangents of the radii-vector angles are equal to $\frac{t - r}{v}$, and these two quantities are alike, according to Equation [4]. In other words, the tangent to the acceleration curve should be perpendicular to the radii vectors of the force curve, provided they are drawn to the proper scale and located at right angles to each other, as in Fig. 3.

Therefore, the following construction is suggested: Plot the force curve (Fig. 3) so that the speed axis $0V$ should be the vertical axis. Draw radii vectors 01, 02, 03, etc., from the center 0 to the middle points 1, 2, 3, etc., of intervals on the force curve, which is intersected by horizontal lines representing speeds in a certain scale; draw through the first interval a line $0a_0$ perpendicular to first radius vector 01; then through the second interval a_0b_0 perpendicular to 02; then through the third interval b_0c_0 perpendicular to 03, etc. The broken line $0a_0b_0c_0d_0$ is the acceleration curve.

As every graphical method, the foregoing construction is subject to certain inaccuracies. If the broken line should be tangent to the acceleration curve in the middle points for which Equation [4] holds good, it would represent the acceleration fairly accurately. Therefore, the smaller the intervals, the more accurate the method. Especially it is true when the radii vectors and the intersecting lines begin to form acute angles and the intersection points are not quite definite. Nevertheless, with some skill, it is possible to get fairly accurate results.

After the acceleration curve versus distance has been plotted, the time curve can also be drawn, and in fact, very simply. Consider a certain increment of distance on Fig. 4, the upper part of

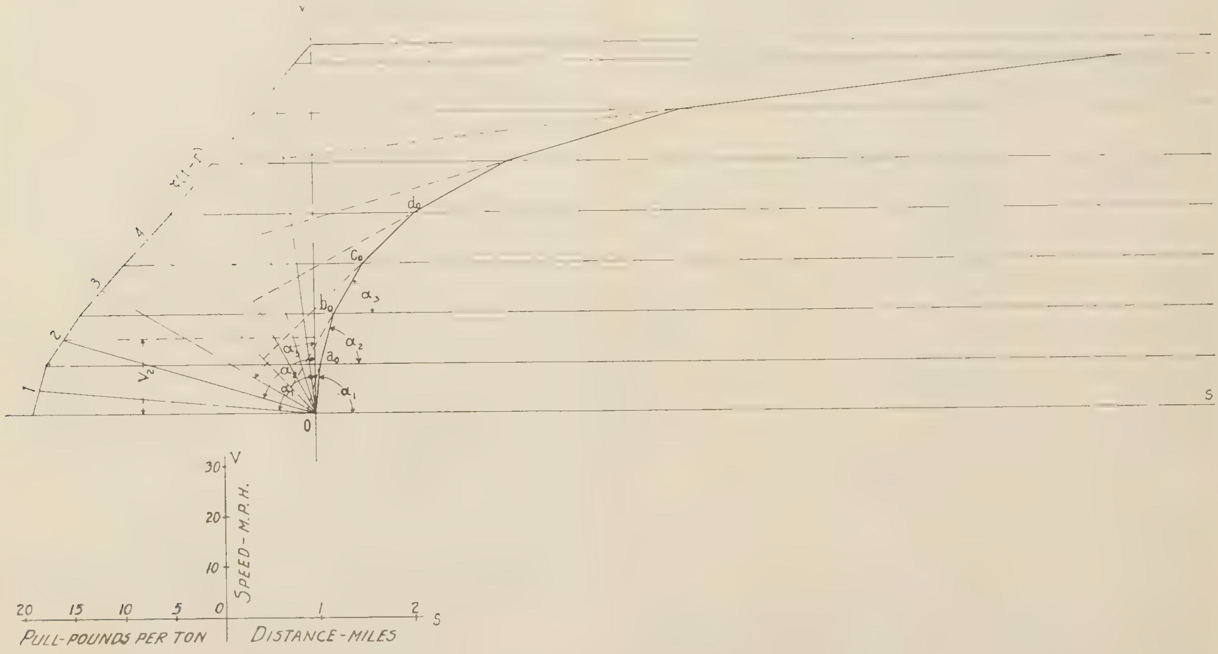


Fig. 3 Lipetz' Method for Speed-Distance Curve

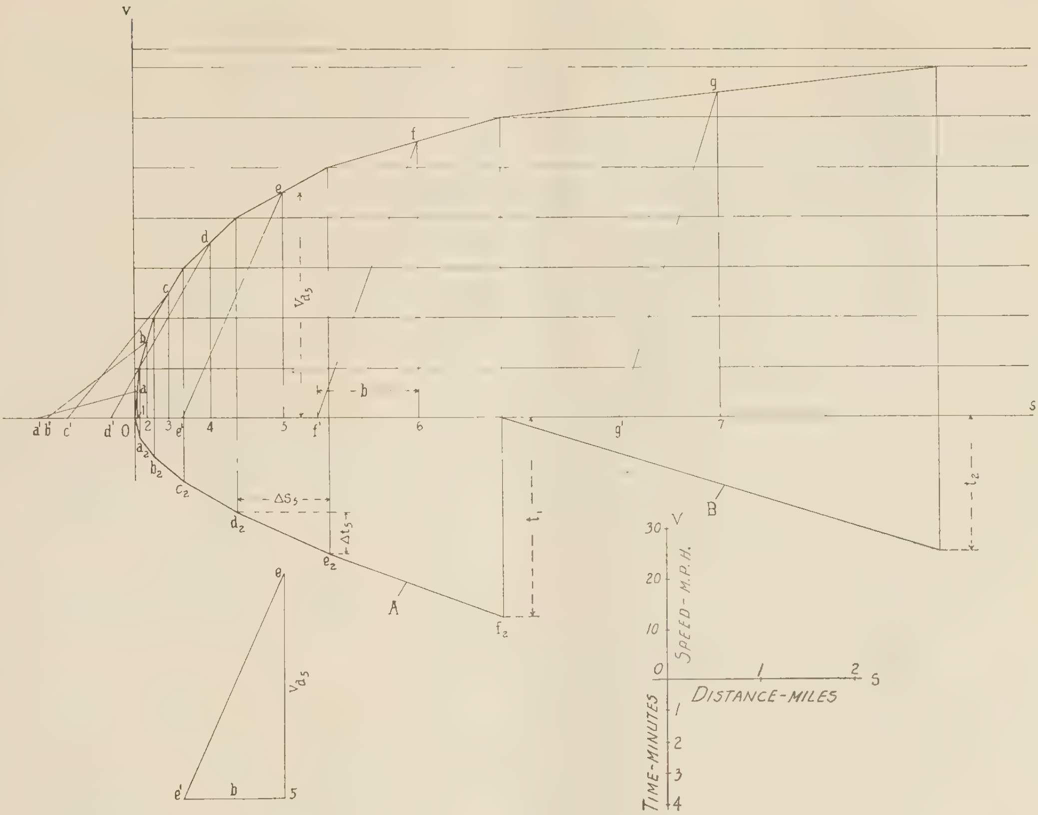


Fig. 4 Lipetz' Method for Speed-Time Curve

which is the acceleration curve of Fig. 3 redrawn somewhat differently. The time needed for covering e distance of an elementary increment will be

$$\Delta t = \frac{\Delta s}{V_a} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

where V_a is the average speed for the distance increment. This suggests a simple method for construction of the time curve.

Divide the speed-versus-distance curve into elements. The easiest way is to take the elements already drawn in Fig. 3 through the same steps of speeds. Mark the middle points of the elements a, b, c, d, e, f, etc., (Fig. 4); drop perpendiculars from these points to the distance axis $0S$ — 1, 2, 3, 4, etc.; measure up a constant parameter b equal 2 in. from each foot of the perpendiculars to points a^1, b^1, c^1, d^1, etc.; draw radii vectors a^1a, b^1b, c^1c, d^1d, e^1e, f^1f, etc., to the middle points, and direct $0a_2$ perpendicularly to a^1a through the length of the first interval; a_2b_2 perpendicularly to b^1b through the second interval; b_2c_2 perpendicularly to c^1c through the third interval and so on. All these constructions are identical with the construction shown in Fig. 4 for the fifth interval, as an example, from which it is evident that

$$\frac{\Delta t_5}{\Delta s_5} = \frac{5e^1}{5e} = \frac{b}{V_{a5}}$$

or

$$\Delta t_5 = \frac{\Delta s_5}{V_{a5}}\, b \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [5a]$$

in accordance with what is required by Equation [5] in a certain scale. The broken line $0a_2b_2c_2d_2$, etc., is the desired time curve. The total time is equal to the height of the last construction. For convenience, the broken line can be divided into branches A, B, etc., and their heights, measured to a certain scale, added together, will represent the total time.

As to the scales by which the distance and time should be measured and read, the procedure is the same as the one used before, namely, the distance was based on the construction shown in Fig. 3 and determined by Equation [4], or

$$\frac{dv}{ds} = 37.5\,\frac{t - r}{v} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4a]$$

if v is in miles per hour, s in miles, $t - r$ in lb per ton of weight, ζ is the coefficient equal 37.5, for $\gamma = 0.0528$. If we should use for the variables the scales given previously, namely

ν = 10 mph = 1 in.
φ = 5 lb per ton = 1 in.
σ = miles distance = 1 in., which is sought,

then the relation from the geometric construction is

$$\frac{dv/\nu}{ds/\sigma} = \frac{(t-r)/\varphi}{v/\nu}$$

or

$$\frac{dv}{ds}\,\frac{\sigma}{\nu} = \frac{t - r}{\varphi v/\nu}$$

Putting, as before, all variables on one side of the equation

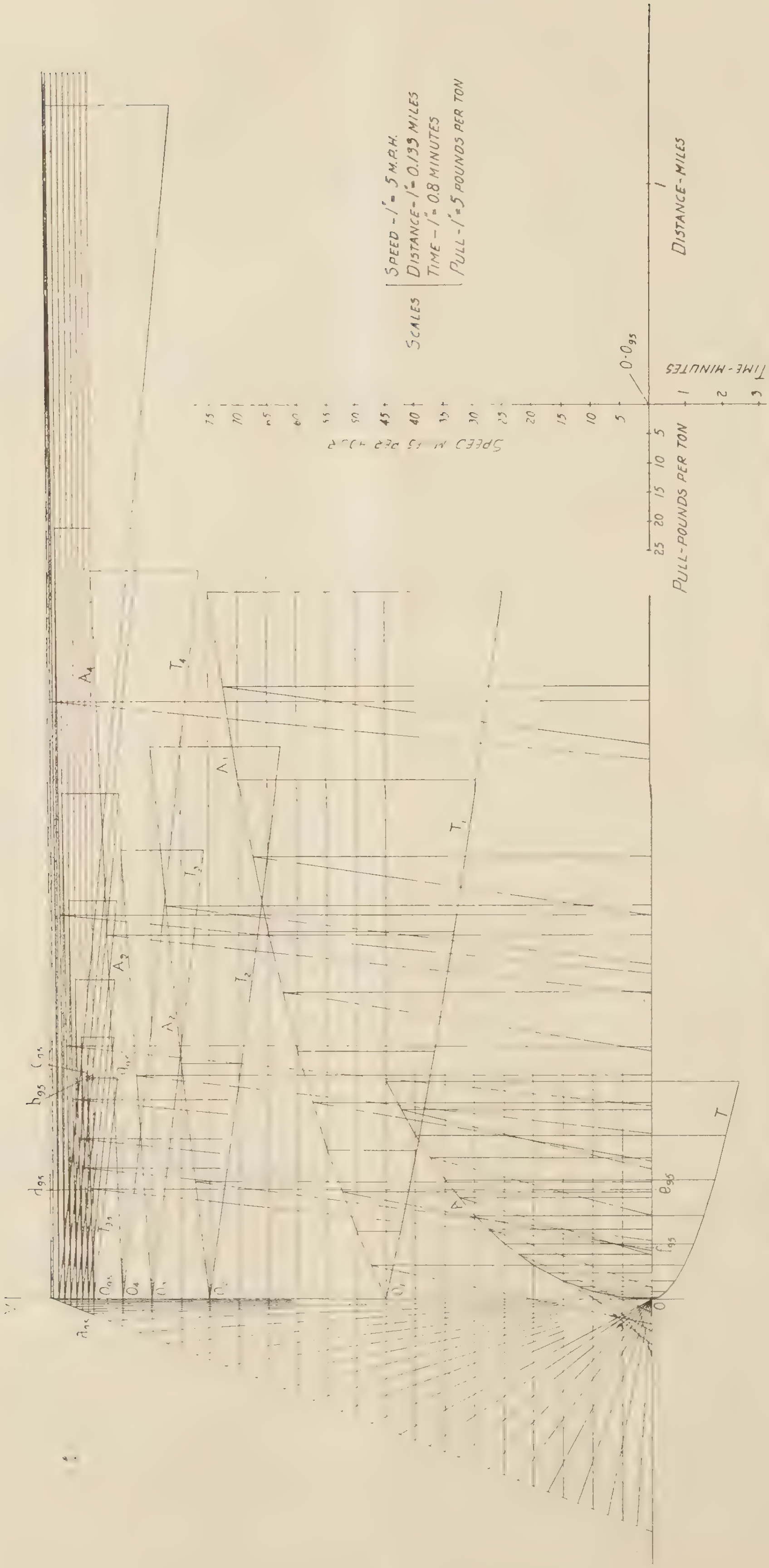


Fig. 5 Speed-Distance-Time Curves for N. Y. Central Locomotive J-3a Tractive Effort—Straight Line

$$\frac{dv}{ds}\frac{\sigma}{\nu}\frac{\varphi v}{\nu(t-r)} = 1$$

Substituting the right-hand side of Equation [4] for $\frac{dv}{ds}$

$$37.5\frac{t-r}{v}\frac{\sigma}{\nu}\frac{\varphi v}{\nu(t-r)} = 1$$

and after cancellation,

$$\sigma = \frac{\nu^2}{37.5\varphi} \qquad [6]$$

Inserting the values for scales in formula [6]

$$\sigma = \frac{10^2}{37.5 \times 5} = \frac{20}{37.5} = 0.533 \text{ mile per inch,}$$

if the other scales are as previously chosen.

As to scale for time when figured by the method in Fig. 4 on the basis of formula [5a], if we follow the same procedure

$$\frac{\Delta t/\theta}{\Delta s/\sigma} = \frac{b}{V/\nu}$$

Comparing this with [5], it is again evident that

$$\frac{\sigma}{\theta} = b\nu$$

or

$$\theta = \frac{\sigma}{b\nu} = \frac{\nu^2}{37.5\varphi b\nu} = \frac{\nu}{37.5b\varphi} \qquad [7]$$

For the previously used scales and constants

$$\theta = \frac{10}{37.5 \times 2 \times 5} = \frac{1}{37.5} = 0.0267 \text{ hr}$$

which is true, at the same scales, or for any train with $\gamma = 0.0528$ and $\zeta = 37.5$.

Generally

$$\theta = \frac{91.18(1+\gamma)\nu}{3600b\varphi} \text{ hr per in} \qquad [7a]$$

and

$$\sigma = \frac{91.18 \times (1+\gamma)\nu^2}{3600 \times \varphi} \text{ miles per inch} \qquad [6a]$$

where

ν = scale for speed, miles per hour per inch
φ = scale for forces, pounds per ton per inch
b = parameter, units

as used in the described method of plotting speed, distances, and time curves.

By using the graphical method, we are able to read a definite answer as to distance and time, because the lines, no matter how acute the angles become, will finally intersect each other. It would be different if we should attempt to solve the problem analytically. By looking at Figs. 2 and 4, it is evident that the numerators of Equations [2] and [4] are nearing zero at balanced speed and the integrals for time and distance reach infinity. However, we know that trains are reaching their destination in finite time. It is easy to see why it is so. A slight increase in the pull caused by the engineer of the locomotive, or a slight reduction in train resistance, when the speed is nearing the balanced speed, will establish a slightly greater speed for the next period of running. So, for instance, in a train which the author has investigated, with speed balancing out at 102.5 mph, the distance and time determined by the author's graphical method were plotted (in Fig. 5) from 95 to 102 mph for one-mile intervals. The total distance from summing up the 1 mile-per-hour intervals was then 16.25 miles and the time 9.8 min, and the average speed was 99.5 mph. Then if this be covered in one run with one acceleration from 95 to 100 mph and two accelerations of 1 mph each (from 100 to 101 and 101 to 102 mph), the total distance from 95 to 102 mph would be 15.45 miles in 9.25 min with an average speed of 100.2 mph. If the whole increase in speed of 7 mph from 95 to 102 mph is run through in one interval, then, as found graphically, we need 8.35 min to cover 14.0 miles with an average speed of 100.5 mph. Consequently, it does not make much difference how we divide the intervals in speed, provided they are small, not over 5 mph for the higher speeds. The average speeds of the last elements, nearing the balanced speed, are about the same, although the distances and times may seem to be quite different. This is very gratifying, as the speeds of the runs will be thus practically not affected.

Analytical Method and Examples

If we knew the law of the difference between tractive effort and resistance curve, namely, if we knew in Equation [2a] the functional relationship of $t - r$, the integration of this equation would be possible in some cases. It is seldom that this difference can be expressed in a function which would be easily integrated. However, in a modern steam locomotive the tractive effort less streamline air resistance can, with a close approximation, be assumed to be a straight line, or a system of straight lines, as shown in Fig. 7. In such a case an integration, although tedious and cumbersome, is possible.

Suppose we take Equation [2a] for which we assume a linear function for $t - r$ of the type

$$\frac{t-r}{a} + \frac{V}{b} = 1$$

or

$$(t-r)b + Va = ab$$

and

$$\frac{(t-r)b}{96} = \frac{ab - Va}{96}$$

Consequently, from Equation [2a]

$$\frac{dt}{dV} = \frac{96b}{ab - Va}$$

$$T = \int_0^{V_0} \frac{96b\,dV}{ab - Va} + C \qquad [8]$$

A more accurate coefficient 96 corresponding to $\gamma = 0.0528$ instead of the approximate coefficient 100 has been assumed for all of these calculations.

This is a logarithmic integral which, when developed with all the constants and between the proper limits, gives a long, complicated formula. The calculation requires tedious work. The advantage of the graphical method becomes at once apparent after the first attempt is made to solve the equation analytically; and this only with the approximation of one straight line for the tractive effort, which is not always possible.

The author, however, did it once for the New York Central J-3 locomotive, for which the tractive effort less train resistance

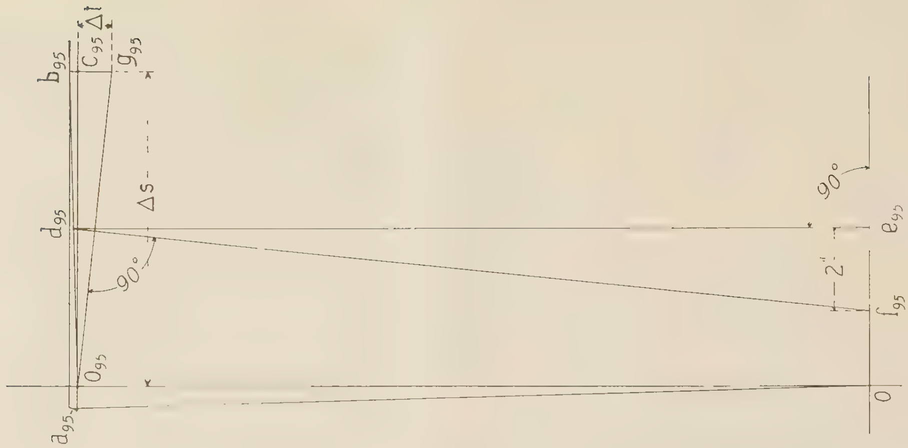


Fig. 6 Plotting Distance and Time Increments for 95–96 Mph Speed Interval

was shown in Fig. 9 of the author's discussion of C. T. Ripley's paper (17, p. 360). It is evident that $t - r$ curve differs slightly from a straight line. This curve is also shown in Fig. 5 for a somewhat lighter train, as a straight line. The equation of this line is

$$\frac{t - r}{a} + \frac{V}{b} = 1$$

Consequently, following Equation [2a]

$$T = \int_0^{V_0} \frac{96bdV}{ab - Va}$$

The result of integration, in seconds, is

$$T = 96 [-2.57 \log_e (39.87 - 0.389V)]_0^{102} + 909.317 \ldots [9]$$

The formula for distance is even more complex. It can be found

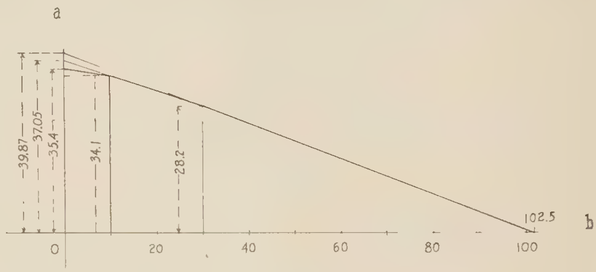


Fig. 7 Tractive Effort—Broken Line

by solving an exponential function for V and making a second integration of the expression $V = \frac{ds}{dt}$. This also has been done by the author for the foregoing case of the New York Central J-3

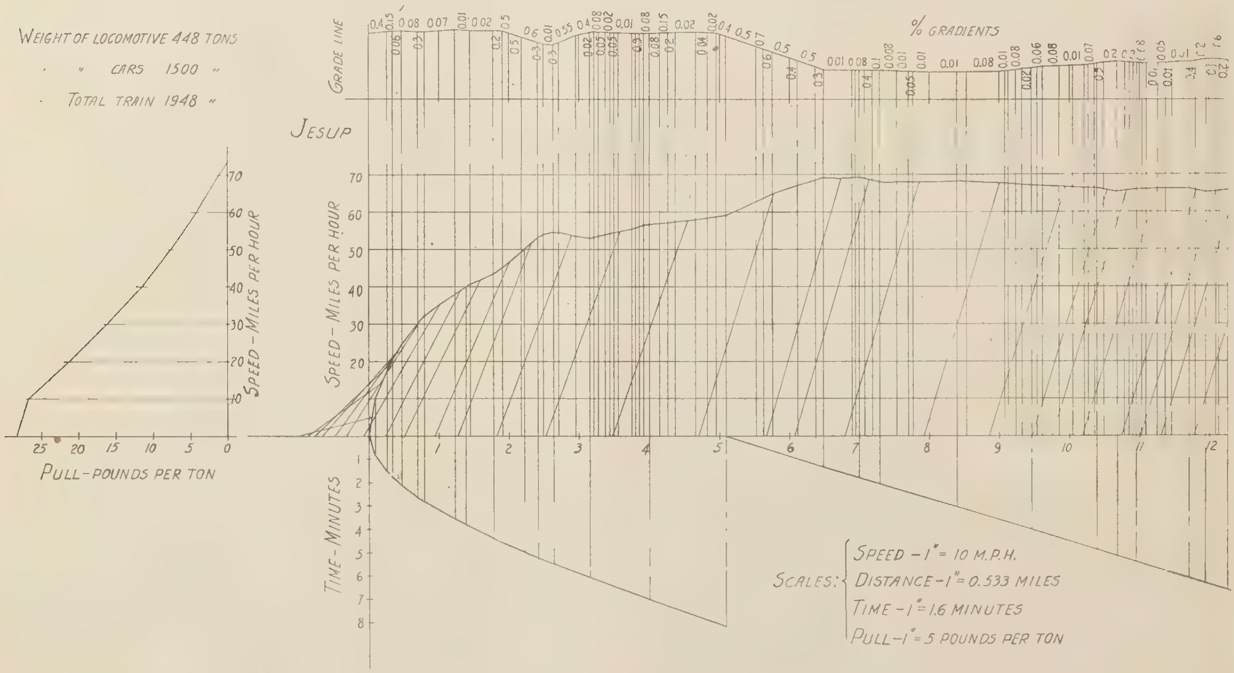


Fig. 8 Test Run With Locomotive

locomotive, with a straight-line tractive curve. The results are given in Table 1 in comparison with the graphical method for this New York Central locomotive J-3.

The right-hand side of Fig. 5 shows the construction of the distance and time curves of the author. For the sake of space, the acceleration line, speed versus distance, is shown in several branches, A, A_1, A_2, etc., starting out from point 0, 0_1, 0_2, etc. They are all drawn according to the author's method.

A is the acceleration curve from start at zero speed to 45 mph (the scales are given on the chart); A_1 is the acceleration curve branch from 45 to 75 mph; A_2, A_3, A_4 are accordingly the acceleration branches from 75 to 85 mph, from 85 to 90 mph, and from 90 to 95 mph. The curves after 95 mph are drawn for speed intervals of one mile per hour. (See example Fig. 6.)

The corresponding branches of the time curve for the same intervals are also shown on this chart (Fig. 5) T, T_1, T_2, T_3, etc., drawn from the same centers, 0, 0_1, 0_2, 0_3, and so on. From 0_{95} starts the T_{95} curve, the construction of which was given separately on Fig. 6.

TABLE 1

Speed at end of interval, mph	Graphical method (Fig. 5): Distance covered, miles	Graphical method (Fig. 5): Time from 0 to end of interval, sec	Analytical method formulas [8] and [9]: Distance covered, miles	Analytical method formulas [8] and [9]: Time from 0 to end of interval, sec
5	.007	12.3	.0071	12.369
10	.038	25.0	.041	25.329
15	.086	40.0	.090	39.055
20	.155	53.5	.154	53.558
25	.255	68.0	.254	68.994
30	.385	84.0	.386	85.440
35	.543	102.0	.540	103.086
40	.740	122.0	.740	122.062
45	.991	142.5	.990	142.636
50	1.280	164.0	1.270	164.000
55	1.690	192.5	1.639	189.317
60	2.120	219.0	2.124	219.000
65	2.650	249.0	2.643	249.000
70	3.360	288.0	3.339	287.000
75	4.080	324.0	4.079	324.000
80	5.165	374.0	5.164	374.000
85	6.600	437.0	6.604	437.000
90	8.550	516.0	8.544	516.000
95	11.900	646.0	11.884	645.361
96	12.850	681.8	12.822	680.626
97	13.940	722.4	13.932	721.958
98	15.320	773.0	15.277	771.532
99	17.100	838.0	16.981	833.638
100	19.200	913.7	19.277	916.719
101	22.820	1043.5	22.817	1043.286
102	28.150	1232.0	30.517	1316.651
			+ Infinity	+ Infinity

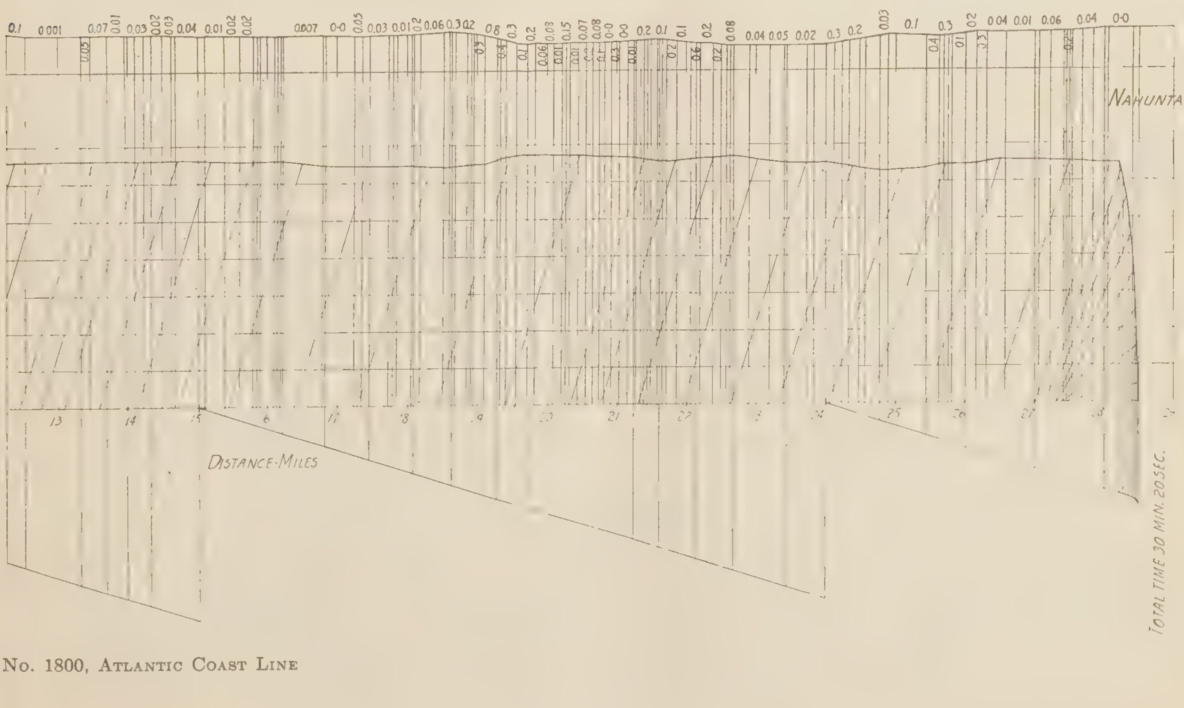
The distances and times were figured both ways—analytically by integration of Equation [8] and double integration of the expression for speed; also graphically as shown in Fig. 5. The results of these calculations are given in Table 1.

This construction was made in order to compare the results of the author's method with the analytical method as to its accuracy, and at the same time to check the scales and the correctness of the proposed formulas [3], [6], [6a], [7], and [7a]. The coincidence of the results is very gratifying.

The next example is the test run of the Atlantic Coast Line steam locomotive, built by the Baldwin Locomotive Works (21). From the information published in this article and additional information kindly supplied by the builders, it was possible to plot a curve for the train from Jesup to Nahunta, a distance of 28.5 miles. A chart was made for this section of the road for a train of a weight given in the publication (1948 tons). The power of the locomotive was figured to the author's formulas of 1934 (22); the train resistance was taken in accordance with Davis' formula (23); the air resistance with formulas of the author (19). The average speed of the run, as found from the chart over the profile (Fig. 8), was 56.5 mph. The actual average speed on test was not given for the investigated division, but for the whole distance (648 miles) it is shown in the Baldwin publication as 53.8 mph. As signal stops and other retardations are included in the entire run, the agreement between speeds is very good.

An acceleration curve plotted from test data on a distance basis, and another curve on a time basis, are given in the Baldwin article (21, p. 19). Both are redrawn in the author's chart (Fig. 9) and they show good agreement, two almost coinciding curves.

The author's method was used for the first time in this country when the *Hiawatha* train was built for the Chicago, Milwaukee, St. Paul and Pacific in 1935. The entire run from Milwaukee to St. Paul was investigated for the *Hiawatha* with seven cars



No. 1800, Atlantic Coast Line

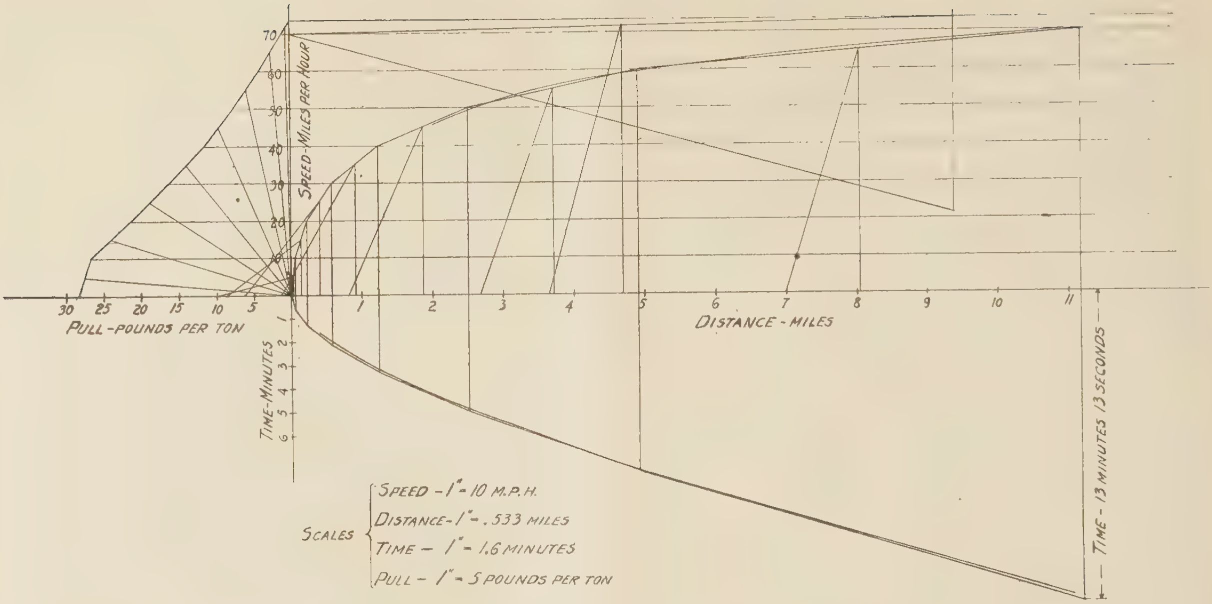


Fig. 9 Comparison of A.C.L. Test Curve and Lipetz' Curve of Acceleration

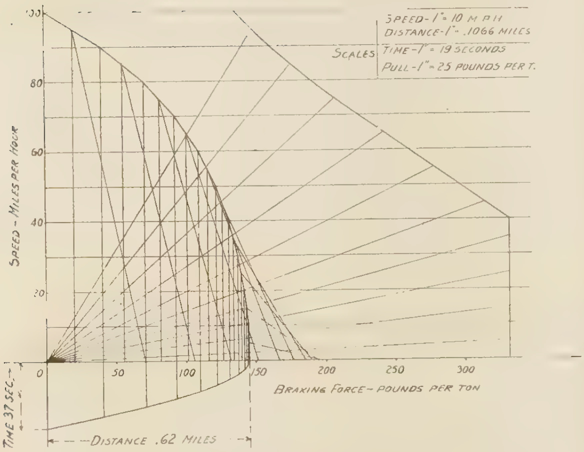


Fig. 10 Distance-Time Curve for *Hiawatha* 7-Car Train at Braking

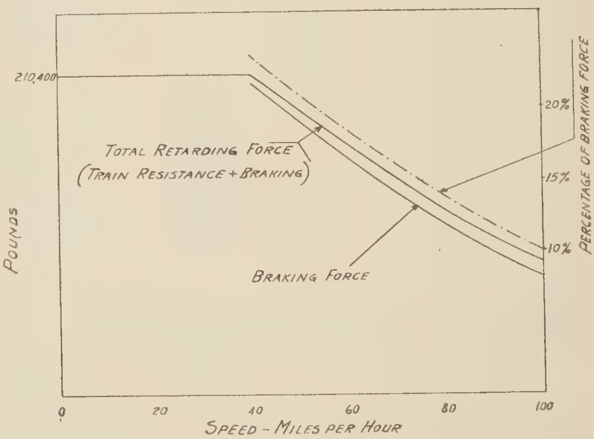
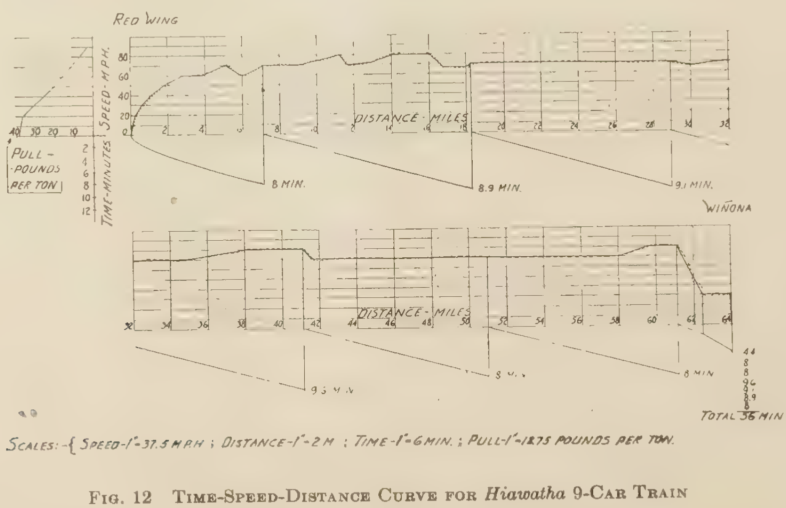


Fig. 11 Retardation Force for *Hiawatha* 7-Car Train at Braking



SCALES:- { SPEED-1" = 37.5 M.P.H. ; DISTANCE-1" = 2 M ; TIME-1" = 6 MIN. ; PULL-1" = 12.75 POUNDS PER TON.

Fig. 12 Time-Speed-Distance Curve for *Hiawatha* 9-Car Train

of a total weight of 358 tons and the time was found to be 6 hr, 30 min for the entire distance (409 miles). At present the *Hiawatha* runs with nine cars having a total weight of 440.4 tons, making the total weight of cars and locomotive 688.4 tons. Fig. 12 represents the time-speed-distance curve for this new train and the *Hiawatha* locomotive, the tractive effort of which was built according to the author's 1934 tractive-effort moduli (22), his resistance curves for equipment (19), and his time-distance graphical method, as exemplified in the foregoing. Between stations Red Wing and Winona the time thus found was 56 min, while the tape taken off the train, shown in Fig. 12 in dotted lines, and the actual performance give 55 min. After this, many more high-speed Diesel-electric and turbine-driven trains were investigated. From the examples which have already been given, the method and procedure are obvious. The only curve which might not be amiss to present, as an example, is the braking curve, which has been once shown at the end of chart of Fig. 8, for the Atlantic Coast Line train.

The braking curve was drawn for the *Hiawatha* train of a total weight of 632 tons, consisting of a locomotive with a weight, including tender, of 274 tons. The braking power of locomotive and tender have been assumed as follows: Engine truck 45, drivers 78, trailer 60, and tender 100 per cent of the light weight on the wheels, and cars 90 per cent of their weight.

The retarding force (augmented by train resistance) friction between braking shoes and wheels is shown in Fig. 11. On the basis of this assumption the braking force was built as given in Fig. 1. The retardation curve was plotted in the same way as the acceleration curves, for both speed and time.

As can be seen, the time of braking was determined as 37 sec for bringing the *Hiawatha* train to a stop from 100 mph; the distance for braking has been found to be 0.62 mile.

Conclusion

The practicability of the proposed method has been proved during many years in many countries and for different trains, by international research, as it were. It may be found useful for application to our trains, especially those for high speed, for which acceleration and retardation speeds and times require exact determination.

BIBLIOGRAPHY

1 "Investigation of Movement of Electric Railway Cars," by G. E. Dubelir, Kiev, 1908, doctor's dissertation, Kiev Polytechnic Institute (in Russian).

2 "Locomotives," by G. V. Lomonossoff, a mimeographed privately issued textbook for students of Warsaw and Kiev Polytechnic Institute, 1901-1902 (in Russian).

3 "New Methods of Calculation of Train Propulsion Times," by A. O. Chechott, St. Petersburg, 1911, doctor's dissertation (in Russian).

4 "Simplified Methods of Evaluating Train Running Times," by A. I. Lipetz, St. Petersburg, Viestnik Obstchestva Technologov, 1913 (in Russian).

5 "Die Berechnung der Fahrzeiten und Geschwindigkeiten von Eisenbahnzügen aus den Belastungsgrenzen der Lokomotiven," by Strahl, *Glasers Annalen*, vol. 73, 1913, no. 869, pp. 86–91.

6 E. E. Seefehlner, Elektrische Zugförderung, Berlin, 1922.

7 Anweisungen für die Ermittlung der Fahrzeiten der Züge nach den Zeichnerischen Verfahren von Unrein; Dr. Ing. Müller; Oberregierungsbaurat Strahl; Regierungsbaurat; Dr. Ing. Velte; Oberregierungsbaurat Caesar, by Geheimer Ober Caurat Dittmann; Organ für die Fortschritte des Eisenbahnwesens, June 15, 1924, pp. 117–129.

8 "Notes on the Plotting of Speed-Time Curves," by C. O. Mailloux, Trans. A.I.E.E., vol. 19, 1902, p. 1035.

9 "Graphic Method for Speed-Time and Distance-Time Curves," by E. C. Woodruff, Trans. A.I.E.E., vol. 33, 1914, p. 1689.

10 "Some Graphical Solutions of Electric Railway Problems," by A. M. Buck, University of Illinois Bulletin No. 47, vol. 13, July 24, 1916 (Engineering Experiment Station Bulletin No. 90).

11 "Ujednostajnienie we Wszystkich Dyrekcjach P.K.P. Sposobu Okreslania Obciazenia i Czasów Jazdy Pociagów Osobowych i Towarowych," by A. O. Chechott, Warsaw (Poland), 1933.

12 "Über rechnerische und zeichnerische Ermittlungen der Fahrzeiten von Eisenbahnzügen," by W. Lubimoff, Berlin, 1932.

13 M. M. Crémer, *Électricité et Méchanique*, Thomson-Houston, May-June, 1925, pp. 6, 13, 15.

14 "The Working Out of Electric Traction Problems," by Thomas Ferguson, *The Metropolitan Vickers Gazette*, August, 1926, pp. 298–309.

15 "Locomotive and Train Acceleration," by L. H. Fry, *The Engineer*, May 2 and 9, 1913, pp. 462 and 483.

16 "Traction Problems," by G. V. Lomonossoff, Berlin, 1922 (in Russian).

17 "High-Speed Lightweight Trains," by C. T. Ripley, Trans. A.S.M.E., vol. 62, 1940, pp. 347-360-366.

18 "Traction Problems," by G. V. Lomonossoff, Berlin, 1922, p. 137; reference to minutes of 19th meeting of Russian Master Mechanics' Association, 1912, p. 326; reference in "Zheleznodorozhnoe Delo," 1911, p. 134; also first edition (1912) of "Traction Problems," p. 62, where Lipetz' method was described, and reference (12), p. 29, footnote 33.

19 "Air Resistance of Railroad Equipment," by A. I. Lipetz, Trans. A.S.M.E., vol. 59, 1937, p. 617.

20 Alco Handbook, 1917, pp. 17 and 35.

21 "Test Runs of Atlantic Coast Line Locomotive No. 1800," *Baldwin Locomotives*, February, 1938, p. 18.

22 "Tractive Effort of Steam Locomotives (Locomotive Ratios—II)," by A. I. Lipetz, Trans. A.S.M.E., vol. 56, 1934, pp. 923–945.

23 "The Tractive Resistance of Electric Locomotives and Cars," by W. J. Davis, *General Electric Review*, October, 1926, pp. 685–707.

24 Calcul des Temps de Parcours des Trains, *Revue Générale des Chemins de Fer*, April, May, 1931.

25 "Locomotive Operation," by A. O. Wood, New York, N. Y., 1925.

26 "Locomotive Data Book," The Baldwin Locomotive Works, Philadelphia, Pa., 1939.

Discussion

R. P. Johnson.[4] Modern train operation, with its fast schedules and heavy tonnages, presents a serious problem for railroad-operating staffs and mechanical departments, and also the preliminary engineering departments of locomotive builders. Very often the running time is so reduced and the train consist so changed as to make prior experience of railroads along these lines of diminished value, hence, theoretical considerations become increasingly important. This paper, which indicates careful research and attention to detail, should prove of considerable interest to those concerned, being both timely and informative.

While the development of the basic analytical formulas in this paper is along rational and orthodox lines, the graphical method shown is ingenious and apparently a close approximation of actual results, based on the comparisons shown for the Atlantic Coast Line and *Hiawatha* locomotives.

To compute tractive effort for locomotives operating at ultrahigh speeds, approximating 100 mph, is still a problem, as is also train resistance, all due to the limited amount of test data available. These items render difficult an accurate calculation of the accelerating power available. At such high speeds, much of the existing test data is extended by extrapolation, an expedient which must serve until more definite information is obtained.

The writer's company uses a combination analytical-and-graphical method, based on formulas published about 30 years ago, and fundamentally identical with the method shown in the paper in so far as the analytical method is concerned. Tractive effort is computed in the conventional manner; locomotive tender and train resistances in accordance with the Davis formulas; and the difference represents the accelerating power

[4] Chief Engineer, The Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

available. From this force, curve and grade resistances are deducted, as may be necessary, and the remainder used in plotting speed-distance or speed-time curves as may be desired. The remainder is substituted in the following conventional formulas:

For distance in feet necessary to accelerate

$$D = \frac{70}{A}(V_2^2 - V_1^2)$$

For time necessary to accelerate

$$T = \frac{95.6}{A}(V_2 - V_1)$$

where V_1 = lower speed
V_2 = higher speed
A = accelerating force per ton available

The speed increments $(V_2 - V_1)$ are usually taken at 5 mph.

By planimetering the area under a speed-distance curve, an average speed is determined from which the running time may be computed.

In working out such problems, consideration must be given to speed restrictions on curves, bridges, through towns, and those imposed by the short distances occasionally existing between station stops.

It would seem to us that the graphical method thus outlined is somewhat simpler than shown in this paper, particularly if long runs are involved. We had occasion not so long ago to make a time study of this nature over a profile more than 2200 miles long. We did not have an opportunity to check actual results against our theory, except in a few places, but these checks were gratifyingly close.

Modern high-speed operation presents problems in deceleration as well as acceleration, but the paper does not say much about this. On some railroads, such as the New Haven, station stops may be rather close. If, on such a road, the decelerating force were relatively low, the train could not, within the distance, be permitted to attain the maximum speed of which it is capable. Such a condition would require special consideration.

In working out a number of time studies, especially over a period embracing the last 10 years, Baldwin engineers have developed a routine procedure along the lines discussed in the foregoing, which provides a reasonably rapid determination of the problem involved. Basic formulas and tabulations are given in a booklet[5] published by the writer's company. We note the bibliography appended to the paper makes no mention of this booklet.

It is hoped that not only will more rational analyses of these problems be possible, but that the plotting of such graphs in preliminary considerations will more closely approximate results to be expected in actual service.

R. T. Sawyer.[6] This interesting paper covers the subject in such great detail, there is very little the writer can add to it in a discussion, except to point out that the author's method actually is one of several, all of which have proved to be quite satisfactory. The principal methods are as follows:

1 Calculate each step separately.

2 Arrange calculations in an orderly manner, such as in tabular form, and then fill in this table progressively.

(a) Time for computations may be shortened by using a slide rule.

(b) Yet more time can be saved by doing the simpler parts of the calculations mentally, completing the more complicated equations with the aid of the slide rule.

3 The graphical method described by the author.

4 A graphical method carried out with special calculating machines, such as that developed by Perkinson of the General Electric Company, Erie, Pa. This method is only practical for calculating long runs as, for example, New York to Chicago, because of the time required to set up the machine. This method gives a very uniform result, but the writer cannot state that it is more accurate than other methods.

[5] "Locomotive Data Book," Eleventh edition, published by The Baldwin Locomotive Works, Philadelphia, Pa., 1939.

[6] Sales Engineer, Diesel Locomotives, American Locomotive Company, New York, N. Y. Mem. A.S.M.E.

Author's Closure

From the introduction to this paper, the reader must have already noticed that the railroad engineers in Europe were the most interested in the author's method of time-speed-distance calculations. It is strange, judging by the paucity of the discussions of the present paper and by the total absence of discussions by railroad engineers, to see that in this country only locomotive builders revealed some interest in the paper. Mr. Johnson speaks in his discussion for the railroad engineers, when he points out that "modern train operation, with its fast schedules and heavy tonnages, presents a serious problem for railroad operating staffs and mechanical departments, and also for the preliminary engineering departments of locomotive builders." The absence of discussion from railroad engineers in this country may probably be explained by the pressure of business which the present state of the country's defense has imposed on the available time of railroad engineers and employees.

Reverting to the substance of Mr. Johnson's discussion, the author does not agree with the statements that the combined analytical and semigraphical method of the discusser's company mentioned in the Baldwin booklets, "Locomotive Data" (26),[7] is fundamentally identical with the method shown in the paper and that "the graphical method thus outlined [Baldwin's] is somewhat simpler than shown in this paper [Lipetz'], particularly if long runs are involved," this because the author's graphical method has not been known so far, and is not similar to any other method so far used in this country. As a rule graphical methods are simpler than analytical, or semianalytical methods, and in this circumstance is to be found the justification of their existence. This is true for many branches of engineering, probably for the graphical calculation of stresses in bridges, and is more true for the plotting of the time-distance curves for railroad trains. If the discusser, Mr. Johnson, wanted to prove the opposite statement, he should have taken an example, for instance, the Atlantic Coast Line Train, or the *Hiawatha*, analyzed in the author's paper graphically, which are also running over long distances, and make the calculations by the combined analytical-graphical method. This would give him an opportunity to show in detail what the method consists of and prove that it is simpler. This has not been done by Mr. Johnson, but it has been done now by the author, and he has found that for a distance of only 3.2 miles the analytical method by the Baldwin formulas occupied the time of at least $3^1/_2$ hr of a very skilled calculator, whereas the author's graphical method required only 45 min by the same calculator for the same distance.

Before closing the reply to Mr. Johnson's discussion, may the author also call the reader's attention to the statement made by Mr. Johnson, namely, "by planimetering the area under a speed-distance curve, an average speed is determined from which the running time may be computed." This is incorrect. A reference to the author's paper, page 604 right-hand column, para-

[7] In all editions up to 1939, a Baldwin method is hardly mentioned; only in the 1939 edition, the formulas on pages 40 and 41 are given for acceleration of trains on level track, but not for calculation of time tables.

graph beginning, "Suppose that a train stands in a station" and the one that follows, will make this clear. As the speed-time curve will therefore represent distance because it is equal to

$$A = \int_{t_1}^{t_2} V dt = \int_{t_1}^{t_2} \frac{dS}{dt} dt = \int_{S_1}^{S_2} dS = S_2 - S_1$$

If this is divided over the length of the diagram $t_2 - t_1$, then the average speed is found. If we should attempt to draw the average ordinate under the curve $V = f(S)$, as the discusser suggests, the ordinate would represent the average speed as function of distance, which is not the average speed as we understand it; the latter must be referred to *time* in order to be *speed*, and it will be impossible by any constants to convert one average into the other. The author does not know these constants and the method of conversion of one into the other, unless by going through the determination of distance S and time t. In other words, the determination of time is needed for a method, the object of which is the same—determination of time. This is not very helpful.

With regard to R. T. Sawyer's discussion, the method which he advocates requires the coincidence of the elements of speed and time with elements of the profile; otherwise, it would become very complicated, calling for a great deal of calculation, and if followed for the example cited, regarding the Baldwin method, it would require much more time than their method for the length of $3^1/_2$ miles. The author suggests that Mr. Sawyer make a comparison of his and the author's methods of calculation applied to a certain stretch of a profile and check the time needed for the calculation in each case. He would find that while the author's graphical method required only 45 min, his method would require at least 6 hr. In addition, both Mr. Johnson and Mr. Sawyer should, in figuring time, consider the accuracy of the results, comparing time and distance with figures of actual experience, as the author did in his paper.

Power Losses in High-Speed Journal Bearings

By F. C. Linn[1] and D. E. Irons,[1] Lynn, Mass.

The effect and interrelationship of the various factors which influence bearing losses have been the subject of many experiments and a vast amount of theoretical work since the basic physical principles of lubrication were outlined by Osborne Reynolds. In the half century following Reynolds' work, the hydrodynamic theory of lubrication has undergone considerable development. Like all theory, it needs to be examined and re-examined in the light of experimental data. In this paper, several novel contributions in the form of experiment and theory are presented. The paper deals with the results of tests made to determine the power losses of bearings of the type used on turbines manufactured by the company with which the authors are associated.

Nomenclature

The following nomenclature is used in the paper:

W	=	total bearing load, lb
p	=	unit bearing load, psi
l	=	axial length of bearing, in.
d	=	journal diameter, in.
F	=	frictional force at journal surface, lb
L	=	power loss, kw
N	=	speed of journal, rpm
ω	=	angular velocity of journal, radians per sec
U	=	peripheral velocity, in. per sec
f	=	coefficient of journal friction
t	=	temperature, F
Δt	=	temperature rise, F
Q	=	rate of oil flow to bearing, gpm
δ	=	specific heat of oil, Btu per gal per deg F
Z	=	absolute viscosity, centipoises $\frac{\text{dyne sec}}{\text{cm}^2}$
ρ	=	specific gravity of oil
x	=	exponent
C, K, K_0	=	constants
ϕ_1, ϕ_2, etc.	=	functions
c	=	diametral clearance, in.
T	=	torque, in-lb

Loss Characteristics of Journal Bearings of Design Similar to Those Tested

The most interesting result of the tests described in this paper is the power-loss formula which was established. This formula, the derivation of which is given later, is

$$L = 2.81 \times 10^{-3}\, d^{1.55}\, l^{0.55} \left(\frac{N}{1000}\right)^{1.43} Z^{0.43}\, Q^{0.43} \ldots \quad [1]$$

A number of other interesting results were obtained as follows:

$$(a) \quad f = \phi_1\left(\frac{ZN}{p}, p\right)$$

for a given bearing when Q, l/d, and d are constant

$$(b) \quad f = \phi_2\left(\frac{ZNQ}{p}, p\right)$$

for a given bearing when l/d and d are constant

$$(c) \quad f = \phi_3\left(\frac{ZNQ}{p}, ld\right)$$

when p and l/d are constant for bearings of similar design.

(*d*) The loss changes only slightly with changes in unit load.

(*e*) The clearance ratio (mils clearance per inch diameter) should be increased with speed to obtain minimum power loss.

(*f*) The actual width of the groove over the top half of the bearing, so long as there is a liberal cross section, has little or no effect upon the loss.

(*g*) Curves resulting from the test data can be extrapolated to predict the loss of bearings of similar design beyond the range of those tested.

Table 1 gives the important dimensions of the bearings which were tested and the range of test conditions for each bearing. As shown in the table, the smallest bearing tested was 3 × 3 in. and the largest 8 × 6 1/4 in.; the over-all pressure range was 51 to 775 psi and the speed range was from 3600 rpm to 12,000 rpm.

Fig. 1 shows the general features of the type of bearing tested.

Method of Testing

All the different methods of determining bearing loss were investigated before beginning the tests. After weighing the relative advantages and disadvantages of these, the heat-balance method was selected as being the best for the purpose. The test-stand setup is illustrated in Fig. 2.

The bearing housings, inlet, and discharge pipes were insulated with 1-in-thick high-temperature insulating material.

The test shaft was driven through a flexible coupling by a variable-speed turbine which was provided with a speed governor and its own oiling system. A turbine-driven oil pump supplied oil to the bearings being tested, and the quantity of oil supplied was measured by means of displacement flowmeters in the lines to the bearings.

Calibrated Fahrenheit thermometers and thermocouples were placed in each inlet and outlet oil line. The thermocouples were used to check the temperature rise. In order to measure accurately the average outlet-oil temperature, the thermometers were placed in a tee where the direction of flow changed from the horizontal to the vertical. An orifice placed in the discharge line had a thermocouple located directly back of it.

Oil coolers were provided to maintain a constant inlet-oil temperature. A constant head of water on the cooler was maintained by means of a gravity-flow system.

Load was applied to the top of one of the bearings, as illustrated in Fig. 2, through a beam which was pivoted on rollers at the bearing, pivot point, and applied load.

Discussion of Tests and Method of Presenting Results

The values of specific heat δ, absolute viscosity Z, and specific gravity ρ, of the oil were based on the average of the inlet and outlet temperatures.

In the early part of the testing work, the flow of oil to the bearings was maintained constant so as to eliminate the effect of that

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[1] Turbine Engineering Department, General Electric Company.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

TABLE 1 PRINCIPAL DIMENSIONS AND RANGE OF TEST CONDITIONS FOR BEARINGS TESTED

Test no.	Dimensions $l \times d$, in.	Clearance ratio, mils per in. diam — Vertical	Horizontal	Top-half grooves, width × depth, in.	Feed type	Drain type	Range of test conditions — p, psi	N, rpm	Z, centipoises	Q, gpm
1	3 × 3	2.0	3.0	None	On up-coming side	Through ends of lining	57–521	8000	10–13	0.6–1.0
2	3 × 3	2.0	3.0	$1^3/_4 \times {}^1/_{32}$	On up-coming side	Through ends of lining	57–521	4000–12000	11.5–18	0.5–1.0
3	3 × 3	2.0	5.0	$1^3/_4 \times {}^1/_{32}$	On up-coming side	Through ends of lining	57–521	8000	12–18	2.0–3.5
4	$3 \times 3^1/_4$	2.5	5.0	$1^3/_4 \times {}^1/_{32}$	On up-coming side	Through ends of lining	52–480	4000–12000	6–15	0.6–3.5
5	$3 \times 3^1/_4$	2.0	5.0	$1^3/_4 \times {}^1/_{32}$	On up-coming side	Through ends of lining	52–480	8000–12000	11.5–18	1.2–2.85
6	$3 \times 3^1/_4$	3.0	4.7	$1^3/_4 \times {}^1/_{32}$	On up-coming side	Through ends of lining	52–480	4000–12000	6–20	0.5–3.5
7	4 × 4	2.5	3.0	$2 \times {}^1/_{16}$	On up-coming side	Through ends of lining	59–775	4000–12000	15–19	0.95–6.6
8	4 × 4	2.5	5.0	$2 \times {}^1/_{16}$	On up-coming side	Through ends of lining	59–775	4000–12000	10.6–17.4	0.9–5.5
9	$6 \times 4^3/_4$	1.3	2.5	$1 \times {}^1/_{16}$	On up-coming side	${}^1/_2 \times {}^1/_4$ orifice	86–500	3600–6000	7–20	1.7–9.0
10	6 × 8	1.3	2.5	(2) $1 \times {}^1/_{16}$	On up-coming side	${}^1/_2 \times {}^1/_4$ orifice	51–355	3600	7.5–16	2.5–9.0
11	$8 \times 6^1/_4$	1.3	5.0	$1^1/_4 \times {}^1/_{32}$ (diagonal)	On up-coming side	Through ends of lining	59–416	3600–6000	6.5–19.5	3.0–15.5
12	$8 \times 6^1/_4$	1.3	2.5	$1^1/_4 \times {}^1/_{32}$ (diagonal)	On up-coming side	Through ends of lining	60–500	3600–6000	7–15.5	5.0–16.0

NOTE: Arc of babbitt of loaded surface = 120 deg — 130 deg.

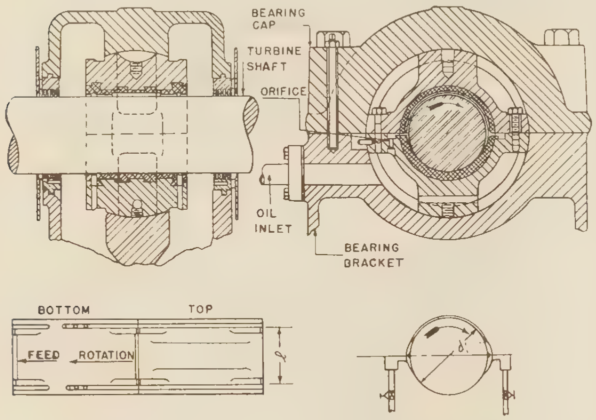

FIG. 1 SHAFT BEARING

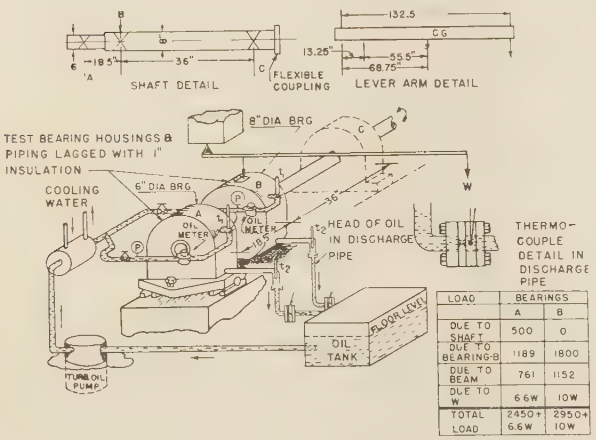

FIG. 2 BEARING TEST STAND

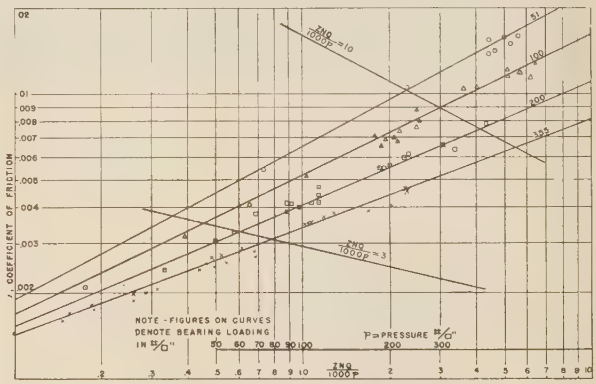

FIG. 3 ZNQ/p VERSUS f CURVES FOR 6 × 8-IN. 130-DEG LINING

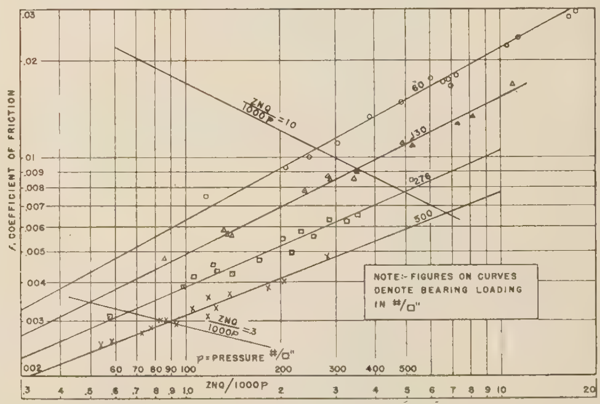

FIG. 4 ZNQ/p VERSUS f CURVES FOR $8 \times 6^1/_4$-IN. 120-DEG LINING

variable, and it was found that, for each value of bearing pressure, the coefficient of friction varied exponentially with ZN/p; in other words a straight-line curve on log-log paper was obtained for f versus ZN/p at each value of unit pressure.

Later, when the total flow of oil to the bearings was varied, it was found that for each value of pressure the coefficient of friction varied exponentially with ZNQ/p. (It should be understood that the flow of oil Q is the total amount supplied to the bearing, which may be more or less than the amount of oil which flows between the journal and lining on the loaded side.) The resulting values of f were then plotted against ZNQ/p for various values of p, Figs. 3 and 4. It was also found that a cross plot of f versus p at constant values of ZNQ/p gave a straight-line curve on log-log paper. Thus, we are led to believe that

$$f = \phi_2\left(\frac{ZNQ}{p}, p\right)$$

for a given bearing when l/d and d are constant, as illustrated in Figs. 3 and 4.

The cross plot of f versus p for constant values of ZNQ/p gave a check upon the accuracy of the test results.

A detailed discussion of the results of the tests on all the bearings need not be given here. However, the results on one of the 3 × 3-in. bearings are of particular interest. The first test on this bearing was with 0.006-in. clearance in the vertical plane and 0.009-in. clearance in the horizontal plane and with no groove

in the top half but with oil fed to both sides at the horizontal joint. The second test was run on the same bearing with a $^1/_{32} \times 1^3/_4$-in. circumferential groove over the top half and the feed was on the upcoming side of the shaft. In the third test, the horizontal clearance was increased to 0.015 in. Fig. 5 is a plot of the results. These tests clearly show that a circumferential groove in the top half and relatively large clearances are required to give minimum loss in high-speed bearings.

The following formulas were used in working up test results:

Relationship of total load and unit pressure

$$W = pld \qquad [2]$$

Relationship of frictional force and power loss

$$F = \frac{33000}{0.746} \times 12 \times \frac{L}{r \times 2\pi N}$$

$$= \frac{84400\ L}{rN} \qquad [3]$$

By definition

$$f = F/W = \frac{84400\ L}{rNW} \qquad [4]$$

Heat-balance formula

$$L = \frac{\Delta t \delta Q}{56.87} \qquad [5]$$

The values of specific heat δ, absolute viscosity Z, and specific gravity ρ, are shown in Fig. 6 for the oil used in test.

Most of the tests were made on bearings whose length was not equal to the diameter. In order to compare bearings properly, they should be geometrically similar. Results of tests, therefore, have been transposed, by means of the following method, to bearings whose l/d ratio equals unity.

Gümbel[2] expresses the loss in a bearing as

$$L = K_0 \omega d \sqrt{Z(l + 4d)\omega d(lpd)} \qquad [6]^3$$

from Equations [3] and [4]

$$L = CfNpld^2 \qquad [7]$$

So for variable bearing length and constant values of p, Z, ω, and d

$$\frac{L_1}{L_2} = \sqrt{\frac{(l_1 + 4d)l_1}{(l_2 + 4d)l_2}} \qquad [8]$$

and

$$\frac{f_1}{f_2} = \sqrt{\frac{(l_1 + 4d)l_2}{(l_2 + 4d)l_1}} \qquad [9]$$

Although Equation [6] does not contain an expression for the

[2] "Steam and Gas Turbines," by A. Stodola (English translation by Lowenstein). McGraw-Hill Book Co., Inc., New York, N. Y., 1927, vol. 1, p. 477.

[3] The factor $(l + 4d)$ was originally used by Gümbel[2] to take account of the end leakage in a bearing.

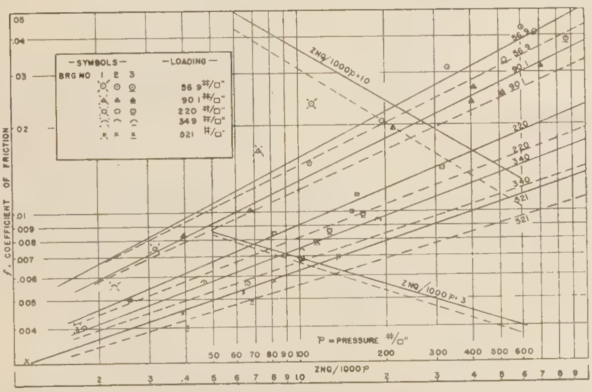

Fig. 5 *ZNQ/p* Versus *f* Curves for 3 × 3-In. 130-Deg Insert Lining

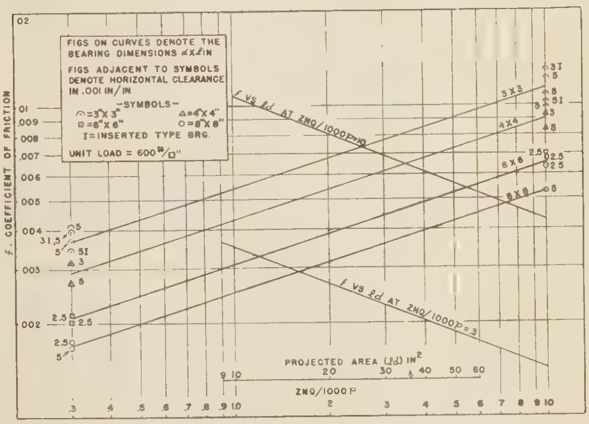

Fig. 7 *ZNQ/p* Versus *f* Curves for Bearings With $l/d = 1$

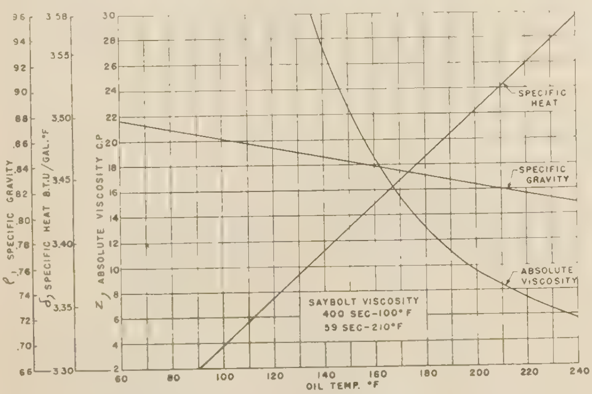

Fig. 6 Specific Heat, Specific Gravity, Viscosity Versus Temperature of Oil Used

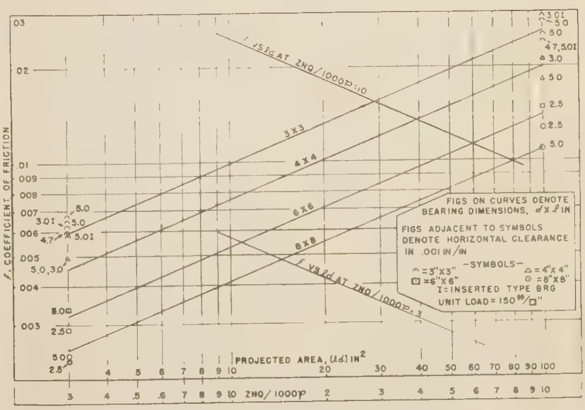

Fig. 8 *ZNQ/p* Versus *f* Curves for Bearings With $l/d = 1$

flow of oil Q, the assumption has been made that Equation [9] is true at constant values of ZNQ/p. Since transformations from one value of the coefficient of friction to another were made only for small changes in bearing dimensions, it is felt that errors due to this assumption are small.

In order to determine the relationship of f and ZNQ/p for the bearings of various size, curves at constant pressures of 150 psi and 600 psi were drawn as shown in Figs. 7 and 8.

Hersey[4] has shown by means of dimensional analysis that

$$f = \phi_4 \left(\frac{ZN}{p}, \frac{c}{d}, \frac{l}{d} \right)$$

and

$$\frac{x}{c} = \phi_5 \left(\frac{ZN}{p}, \frac{c}{d}, \frac{l}{d} \right)$$

thus removing the requirement for geometrical similarity so far as the clearance-diameter and length-diameter ratios are concerned. The tests herein reported also indicate that in addition

$$f = \phi_3(ld)$$

when l/d is constant.

This relationship is illustrated in Figs. 7 and 8, in which a family of f versus ZNQ/p curves, drawn at constant values of p, give straight lines on log-log paper with ld as the parameter. The curves are drawn parallel to each other in order to simplify the mathematical derivation of the power-loss formula. Some slight shifting of the actual test curves was necessary to make them parallel.

A unit load of 150 psi was selected as the value on which to base the loss formula, since in turbine and gear work the bearing pressures vary from 50 to 200 psi, and the loss variation with load within this range is not very great. Fig. 9 illustrates the variation at 5000 rpm for loads from 50 to 600 psi for a 6 × 6-in. bearing.

[4] "The Theory of Lubrication," by M. D. Hersey, John Wiley & Sons, Inc., New York, N. Y., 1938, pp. 86, 87, and 88.

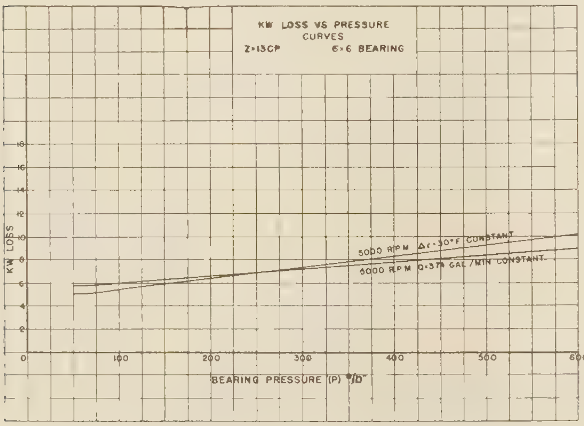

Fig. 9 Kilowatt-Loss Versus Pressure Curves
(Z = 13 centipoises; 6 × 6-in. bearing.)

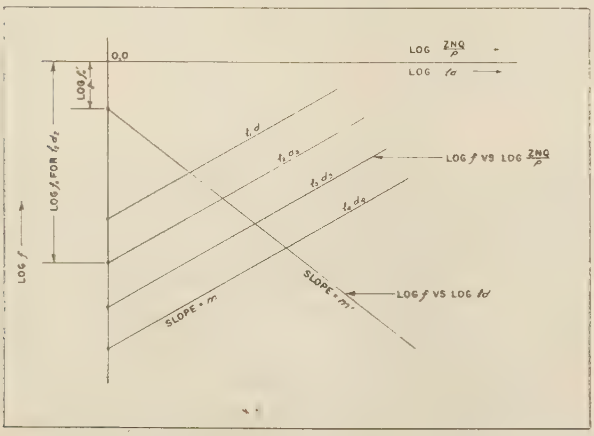

Fig. 10 Log f Versus Log ZNQ/p at Constant Pressure

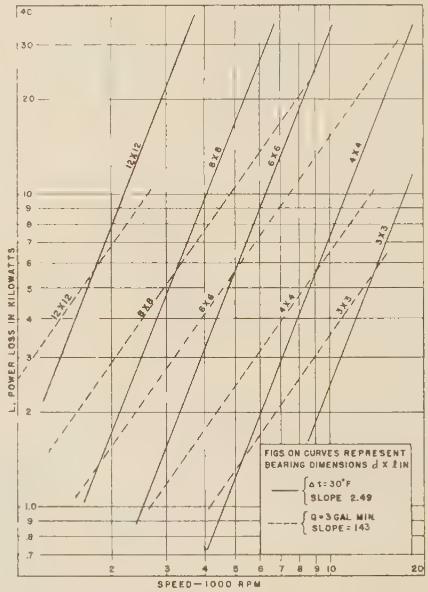

Fig. 11 Bearing Loss Versus Speed for Overshot Lubrication; 120-Deg Bearings
(p = 150 psi; Z = 13 centipoises.)

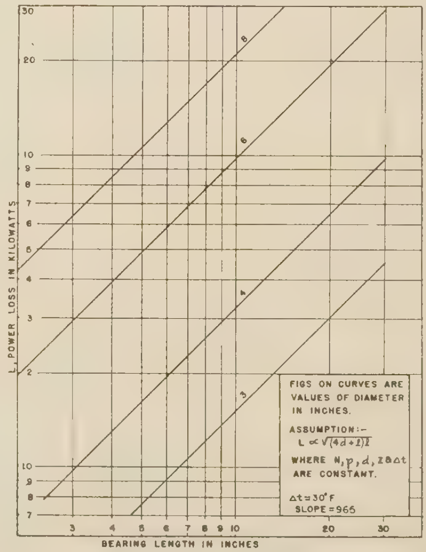

Fig. 12 Kilowatt Loss Versus Length
(Z = 13 centipoises; p = 150 psi; speed, 5000 rpm.)

Derivation of Power-Loss Formula

In order to demonstrate the interrelationship of the primary variables affecting the power loss in bearings, Equation [1] was derived from the test data. The formula is entirely empirical and it should be noted is based on data which have been made consistent within itself. An analysis of the derivation follows:

Referring to Fig. 10, log f, plotted against log ZNQ/p at constant pressure, gives a series of straight-line curves with the bearing size ld as a parameter. These straight lines may be expressed mathematically by an expression of the form

$$\log f = \log f_0 + m \log ZNQ/p \qquad [10]$$

where

f_0 = function of p and ld
m = function of p

Now

$$\log f_0 = \log f_0' + m' \log (ld) \qquad [11]$$

where

f_0' = function of p
m' = function of p

Hence

$$\log f = \log f_0' + m' \log (ld) + m \log ZNQ/p$$
$$\log f = \log [f_0'(ld)^{m'}(ZNQ/p)^m]$$

or

$$f = f_0'(ld)^{m'}(ZNQ/p)^m \qquad [12]$$

From the definition of the coefficient of friction

$$L = \frac{\dfrac{\pi dNWf}{12 \times 33000}}{0.746} = 0.00000592\, pNd(ld)f \qquad [13]$$

Substituting Equation [13] in Equation [12]

$$L = 0.00000592\, p^{1-m} f_0' l^{1+m'} d^{2+m'} N^{1+m} Z^m Q^m \qquad [14]$$

From the master curves of which Figs. 7 and 8 are examples at 150 and 600 psi, respectively, the values of f_0' and the slopes m and m' may be determined. Choosing p equal to 150 psi and substituting the values of f_0' and m and m' in Equation [14] gives the power-loss equation

$$L = 2.81 \times 10^{-3} d^{1.55} l^{0.55} \left(\frac{N}{1000}\right)^{1.43} Z^{0.43} Q^{0.43} \qquad [1]$$

As derived, Equation [1] applies strictly to 150-psi bearing pressure. For other values of pressure, a slight change occurs in the value of the constant and exponents.

Typical curves of speed versus loss, Fig. 11, and bearing length versus loss, Fig. 12, have been plotted by use of this formula. Curves illustrating the effect of other variables on loss can be plotted.

Effect of Top-Half Grooving on Loss

The addition of a groove over the top half of a journal bearing reduces the loss somewhat, but the size of the groove used has little effect on the value of the loss. As recorded in Table 1, the width of the groove was varied from $^1/_5$ to $^2/_3$ of the bearing length. The effect on the power loss was not measurable in these tests, the reason for this being that a vacuum occurs over a large portion of the upper half.

Appendix

Comparison With Other Investigators

(a) Tests made by Metropolitan-Vickers[5] at Trafford Park, Manchester, England.

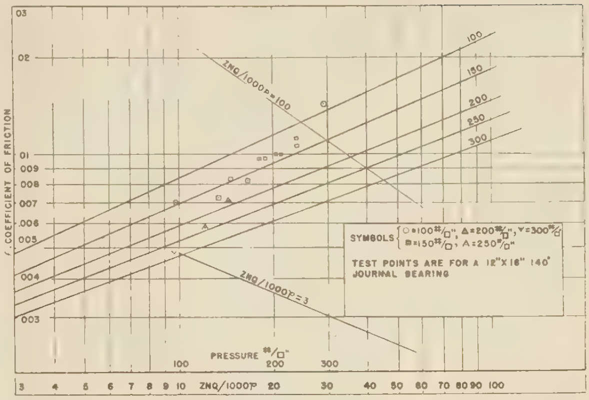

Fig. 13 ZNQ/p Versus f Curves for 12 × 16-In. 130-Deg Lining

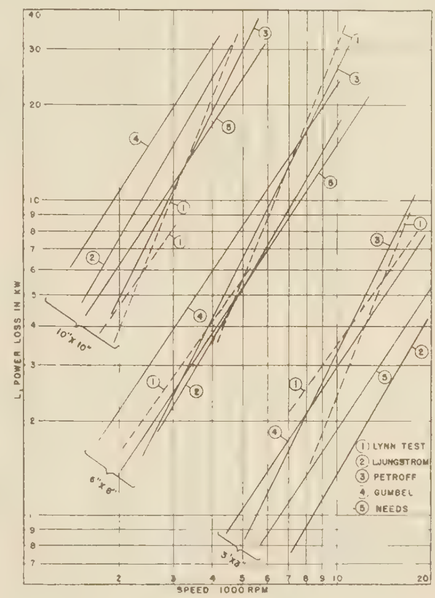

Fig. 14 Comparison of Lynn Bearing Test With Theoretical Calculations
(p = 150 psi; Z = 13 centipoises.)

The first test was made on a 12 × 16-in. cylindrically bored overshot bearing with 0.0025 in. per in. clearance and a $9^1/_2$-in. circumferential groove 0.015 in. deep in the top half. The arc of contact was approximately 140 deg.

The data given for this test were worked up to obtain values of f and ZNQ/p.

Curves of f versus ZNQ/p and f versus p were drawn for a 12 × 16-in. bearing, based on the authors' results.

The Metropolitan-Vickers test points were then plotted as shown in Fig. 13 and very good agreement was obtained with the Lynn test curves.

This close agreement indicates that the loss formulas developed from the tests and presented in this paper are correct. It also illustrates that the method of working up and presenting the

[5] Tests made by the Metropolitan-Vickers Company, Trafford Park, England; partially reported in "Steam Turbine Journal Bearings," by H. L. Guy and D. M. Smith. General Discussion on Lubrication and Lubricants, The Institution of Mechanical Engineers, London, England, vol. 1. American edition published by The American Society of Mechanical Engineers, May, 1938, pp. 115–121.

data gives a direct means of comparison of test data on various sized bearings of similar design.

(b) A comparison of the bearing-loss formula developed at Lynn with results of other investigators has been made. Fig. 14 shows the results of these comparisons for 3 × 3-in., 6 × 6-in., and 10 × 10-in. bearings with a viscosity of 13 centipoises and unit pressure of 150 psi. All of the formulas have been transposed and simplified in order to have the same nomenclature as is used in this paper and to be directly comparable.

1 Curves based on the tests made at Lynn are drawn for $\Delta t = 30$ F and $Q = 3$ gpm for each bearing.

2 The equation as given by Ljungstrom is based upon a Δt of 30 F and in terms of our notation is

$$L = 0.01565ld^{2.7}\left(\frac{N}{1000}\right)^{1.7}(t_1)^{-0.905}\ldots\ldots\ldots[15]$$

It is evident from Fig. 14 that the Ljungstrom formula gives losses that are slightly high for the 10 × 10-in. bearing and 200 to 300 per cent low for the 3 × 3-in. bearing, in the high-speed range (10,000 to 15,000 rpm).

3 In the work by Hersey,[4] Newton's law of viscosity is applied to Petroff's equation for a concentric, full journal bearing; that is, one which is slightly enough loaded and running at high enough speed so that the journal is well centered in the bearing.

The moment of friction, or torque, is

$$T = 2.38 \times 10^{-8}\,(d/c)(l/d)ZNd^3\ldots\ldots\ldots[16]$$

The loss may be expressed as

$$L = \frac{2\pi NT}{12 \times 60} \times \frac{0.746}{550}\ldots\ldots\ldots[17]$$

Then

$$L = \frac{2\pi N}{12 \times 60}\,(d/c)(l/d) \times 0.145 \times 10^{-6}\,Z\left(\frac{N}{60}\right)d^3 \times \frac{0.746}{550}$$

or

$$L = 0.283 \times 10^{-6}\,Z(N/1000)^2\,d^3(d/c)(l/d)\ldots\ldots[18]$$

It is interesting to note the close agreement of this simple fundamental equation with our results at a viscosity of 13 centipoises. At other viscosities, the agreement will not be quite as good.

4 An analysis of bearing theory by Gümbel[2] gives the following expression for loss:

$$\text{Loss in lb-ft per sec} = K_0U\sqrt{Z(l + 4d)\,UW}\ldots\ldots[19]$$

where

$$K_0 = \text{a function of } p,\ Z,\ U,\ l,\ d$$

Assuming an average value of $K_0 = 2.2$, this reduces to

$$L = 35.7 \times 10^{-6}Z^{0.5}(N/1000)^{1.5}p^{0.5}l^{0.5}d^2(l + 4d)^{0.5}\ldots[20]$$

The factor $(l + 4d)$ takes care of the end-leakage effect for various ratios of l to d.

5 S. J. Needs[6] has developed a loss formula which includes the effect of end-leakage and clearance ratio. For the purpose of comparison, the formula has been transposed on the assumption of $l/d = 1$ and for minimum coefficient of friction.

Under these conditions

$$L = 49.8 \times 10^{-6}Z^{0.5}(N/1000)^{1.5}\,p^{0.5}\,d^2l\ldots\ldots\ldots[21]$$

[6] "Effects of Side Leakage in 120-Deg Centrally Supported Journal Bearings," by S. J. Needs, Trans. A.S.M.E., vol. 56, 1934, pp. 721–732.

ACKNOWLEDGMENT

The authors are indebted to Messrs. A. L. Kimball and M. E. Prohl of the General Electric Company for their assistance in preparing the theoretical developments of this paper. They are also grateful for the valuable suggestions and criticisms of other associates.

Discussion

H. D. Emmert.[7] The present state of disagreement that exists between the various published bearing-loss formulas is amply illustrated by Fig. 14 of the paper. The degree of variation between the formulas used by the authors for comparison is of the order of 100 per cent, and the writer is aware of other loss equations in common use which would increase this discrepancy considerably.

Many papers have been published in recent years covering the theoretical phases of journal- and thrust-bearing design, but few of these have given results which could be reduced to a simple formulation of the pertinent measurable variables. The authors' presentation of the test results of bearings as used in steam turbines manufactured by their company is a notable addition to the published literature on practical bearing performance.

The writer has found that, in the great majority of cases, the power loss of a lightly loaded bearing may be calculated with a fair degree of accuracy by a formula of the Petroff type, i.e., based upon the simple shear forces in an evenly distributed oil film, with due allowance for any relief in the bearing surface. For this reason, it has been felt that any formula derived from practical tests would best be expressed as a correction to the loss for an unloaded concentric bearing. This becomes logical when it is considered that the major portion of the loss must result from the average viscous friction around the bearing circumference, regardless of the degree of eccentricity and side leakage.

This conception is borne out from the theoretical standpoint by a short analysis made by the writer of the data presented by S. J. Needs[8] as a result of solutions by electrical integration for 120-deg bearings of finite width. The method of calculation outlined by Needs is quite lengthy, and it was felt that the data could be formulated for easier solution by plotting the results in a manner which could be closely approximated by simple mathematical curves. The resulting formula, valid for 120-deg top and bottom bearings, of the dimensional range encountered in turbine-bearing design, may be expressed in the form

$$HP = 1.75\,d^2l\,\frac{\mu}{c}\left(\frac{N}{1000}\right)^2\left[1 + 1.82\,\frac{pc^2}{\mu N}\left(1 + 1.27\,\frac{d}{l}\right)\right]$$

where HP = bearing loss, hp
d = bearing diameter, in.
l = bearing length, in.
μ = absolute viscosity, psi per sec
c = clearance ratio, diametral clearance/journal diameter
N = rpm
p = unit load, psi

The first part of the formula is easily recognized as an expression for the loss in an unloaded centrally running bearing. This expression is modified by a term proportional to the Sommerfeld variable which in turn includes a side-leakage factor. Therefore

[7] Steam Turbine Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis.

[8] Ref. (6) of paper.

the amount of total loss in a given bearing in excess of the loss when running concentrically is indicated in the numerical solution by the value of the modifying term.

This formula is not presented by the writer as necessarily representing true values of the losses encountered in commercial bearings, but rather as a type of equation giving an immediate picture of the degree of loading on a bearing, as a function of the measurable bearing dimensions. As a matter of interest, however, the agreement with the authors' formula is quite close for the two smaller bearings indicated in Fig. 14.

M. D. Hersey.[9] While the authors' power-loss equation is of exceptional interest as a step in establishing the laws of lubrication of journal bearings, it is not dimensionally homogeneous. Consequently, its use is limited to the range covered by the tests. This valuable study might yield information of wider applicability if expressed in dimensionless variables. See, for example, Edgar Buckingham's study of windage losses in steam turbines,[10] as well as his paper[11] of 1915.

Equation [1] of the paper might be reconstructed by formulating the coefficient of friction for geometrically similar bearings as a function of the appropriate dimensionless variables, for example ZN/p, Q/Nd^3, hNd^2/k, and ap/h. In the foregoing h denotes the heat capacity of the lubricant per unit volume and k its thermal conductivity, while a is the temperature coefficient of viscosity, or fractional change in viscosity per degree rise of temperature.

The second of the four variables was introduced[12] to provide for forced lubrication, a condition excluded when ZN/p is used alone. The third and fourth are more familiar in thin-film lubrication, but applicable to thick films when the viscosity is nonuniform.[13] If the viscosity differences are very great a fifth variable such as a'/a^2 may be required, in which a' denotes the rate of change of a with temperature, assuming our object is to set up charts or equations valid for more than one lubricant. These complications can be avoided to some extent if the viscosity Z is estimated from the mean temperature of the bearing surface, rather than by averaging the inlet and outlet oil temperatures.

It is to be hoped that the tests may be continued in order to check some of the present results both with and without insulation, using a transmission or cradle dynamometer, and with thermocouple junctions in the bearing metal.

[9] Research Director, Morgan Construction Company, Worcester, Mass. Fellow A.S.M.E.

[10] "Windage Resistance of Steam Turbine Wheels," by Edgar Buckingham, *Bulletin*, U. S. Bureau of Standards, vol. 10, 1913, pp. 191–234.

[11] "Model Experiments and the Forms of Empirical Equations," by Edgar Buckingham, Trans. A.S.M.E., vol. 37, 1915, pp. 263–296.

[12] "On the Laws of Lubrication of Journal Bearings," by M. D. Hersey, Trans. A.S.M.E., vol. 37, 1915, pp. 167–202.

[13] Ref. (4) of paper, p. 84.

G. B. Karelitz.[14] Reliable test data on the friction in high-speed bearings are rather scarce, and the published test results of this paper are very welcome. The authors decided to present the coefficients of friction as functions of ZNQ/p, apparently because for each individual bearing these functions give a straight line on log-log paper for each individual value of p, the pressure on the projected area.

The interpretation of the test data may be criticized on the ground that the friction losses in the bearings, i.e., the values of f (coefficient of friction), are caused and determined by the actual viscosity of the oil in the film. The viscosity of the film depends upon the temperatures obtaining actually in the film itself. These temperatures vary from point to point in the films, but not much. The film is so thin that the temperatures cannot differ appreciably from the temperature of the journal itself. It is, therefore, permissible to consider a uniform average temperature of the film in all computation of frictional losses. But this temperature is not the "average of the inlet and outlet temperatures" used by the authors for the basis of their analysis. The film is the source of heat and its temperature is the highest in the bearing. The oil leakage from the bearing ends mixes with cooler oil flowing direct from the relief into the end grooves or into the pedestal, the temperature of the mixture being the outlet temperature. In the writer's experiments, the temperature of the film was measured by thermocouples installed either flush with the bearing or at a depth of a few thousandths of an inch from the bearing surface. With forced lubrication this temperature was from 5 to 20 F higher than the outlet temperature in turbine-type bearings. The difference is smaller in larger bearings where the oil can flow through the clearance with greater ease than in smaller bearings where the direct crossover is more pronounced from the relief groove into the end grooves.

The writer noted that the coefficients of friction given in the chart published by McKee, where they are plotted against ZN/P, and determined empirically on small automobile bearings, applied sufficiently well to turbine bearings of usual design. The paper does not tabulate data on the oil flow, temperatures of inlet and outlet oil, revolutions per minute, and load for individual runs; nevertheless, the writer attempted to apply the McKee chart to the 3 × 3-in. and to the 8 × 6¼-in. bearings used in the plots, Figs. 4 and 5 of the paper. It was assumed that the cooling water and oil flow were regulated so that the inlet-oil temperatures were 150, 165, and 180 F, while the temperature rise of the oil through the bearing was 15 deg for the larger and 30 deg for the smaller bearing. The required oil flow Q was determined on the basis of McKee's chart. The value $\frac{Z'NQ}{1000\,p}$ was then found, where Z' is the viscosity based on

[14] Professor of Mechanical Engineering, Columbia University, New York, N. Y. Mem. A.S.M.E.

TABLE 2 COMPARISON OF DATA ON TURBINE BEARINGS FROM McKEE CHART WITH THOSE FROM AUTHORS' FIGS. 4 AND 5

Q Computed on basis of McKee's chart							Authors plotting		
Inlet temp, F	Outlet temp, F	Film temp, F	Viscosity Z Cp	$\frac{ZN}{p}$	f (McKee)	Oil flow Q, gpm	Base temp, F	Viscosity, Z' Cp	$\frac{Z'NQ}{1000p}$
Bearing No. 2: 3 in. by 3 in.; c/d = 0.0025 (avg); p = 220 psi; W = 1980 lb; N = 6000 rpm; Δt = 30 F; T of film = T_{out} + 20 F; δ = 3.42 Btu per gal per deg F; Q = 116 f, Eqs. [4] and [5]									
150	180	200	9.5	258	0.0060	0.69	165	16.7	0.32
165	195	215	7.7	210	0.0051	0.59	180	13.0	0.22
180	210	230	6.4	174	0.0044	0.51	195	10.3	0.14
Bearing No. 2: 3 in. by 3 in.; c/d = 0 0025 (avg); p = 220 psi; W = 1980 lb; N = 9000 rpm; Δt = 30 F; T_{film} = T_{out} + 20 F; δ = 3.42 Btu per gal per deg F; Q = 116 f, Eqs. [4] and [5]									
150	180	200	9.5	388	0.0085	1.48	165	16.7	1.01
165	195	215	7.7	315	0.0071	1.23	180	13.0	0.65
180	210	230	6.4	252	0.0059	1.03	195	10.3	0.43
Bearing No. 12: 6¼ in. by 8 in.; c/d = 0.002 (avg); p = 276 psi; W = 13,800 lb; N = 3600 rpm; Δt = 15 F; T_{film} = T_{out} + 5 F; δ = 3.46 Btu per gal per deg F; Q = 2020 f, Eqs. [4] and [5]									
150	165	170	15.4	201	0.0059	11.9	157.5	18.3	2.86
165	180	185	11.8	154	0.0048	9.7	172.5	14.9	1.88
180	195	200	9.5	124	0.0041	8.3	187.5	11.2	1.22

the average of the inlet and outlet temperature; this, with f, was entered into Figs. 4 and 5 and compared with the corresponding curves. The procedure is given in Table 2 of this discussion.

It was found that the six points obtained in Fig. 5 and the three entered in Fig. 4, were very close to the 220-psi and 276-psi curves, respectively. In fact, the deviation was of the same order as that of the points plotted by the authors.

It might be suggested that the authors supplement the paper with a table giving the test data, p, t_{in}, t_{out}, N, and Q, pertaining to the points plotted in Figs. 3, 4, and 5. This would help engineers to interpret the tests to suit themselves. This complete record would enhance the usefulness of the contribution.

The authors and their Company should be congratulated on the publication of really valuable test data; those who work in this field realize the amount of time and effort represented by the reported experiments.

F. Nagler.[15] The authors' presentation is extremely commendable, combining, as it does, basic analyses of the factors inherent in the successful lubrication and operation of bearings, the correlation of various test programs, and even the inclusion of practical mechanical details such as the nature of the grooving.

It would be of further particular interest to have comments on the length of the bearing. Fig. 2 of the paper, shows a length slightly greater than the diameter. At what point of length-to-diameter ratio does the self-aligning feature cease to become operative? For example, it is rather difficult to imagine a bearing of a length equal to one half the diameter of the shaft having a very effective self-aligning feature. At some point where this occurs, there must be such high pressures at one end of the bearing as to make average figures of pressure distribution rather fictitious.

If the total flow of oil to the bearing Q is large in comparison to that which is actually effective between the bearing surfaces, is it not fair to assume that the excess oil is merely a cooling medium, more or less inefficiently supplied to the bearing housing and having very little to do with the bearing itself? If such is the case, is it not a little out of order to use the total Q, as is done in the text immediately below Fig. 2, as a basis for a study of the bearing action itself?

Some comment by the authors would be of interest as to the effect of water cooling in practical application of the design data presented. Is it not possible to obtain very much more effective cooling by circulating a cooling medium directly in the shell itself, than by cooling the oil which goes over and around that shell?

Comparison of some of the old rules of some methods as, for example, requiring water cooling when the heating factor (pressure per square inch $\times$ velocity) is above 50,000 and not requiring it when the heating factor is below, say, 25,000, with intermediate determinations based on the duty in question, would be of interest from the standpoint of the authors' analysis. This suggestion is ventured rather diffidently in the full realization that mentioning such rules, against the background of the authors' analysis, places the writer in a decidedly vulnerable position.

All users of bearings, high-speed and otherwise, will be particularly grateful to the authors for their excellent presentation.

S. J. Needs.[16] In its present stage of development journal-bearing theory is not complete, hence the necessity for tests. Even if not entirely satisfactory, however, it has been found that theory generally gives power losses in fair agreement with actual conditions. For this reason, the writer was impressed by the apparent discordance between theory and test results, indicated by the lack of agreement between the curves in the authors' Fig. 14. Here, the disagreement is of the order of 100 per cent, the upper curve in each group indicating power losses about twice as great as the lower curve.

Perhaps the reason for this is the basis upon which the comparisons are made. The authors' tests were made with a loaded-bearing arc of 120 to 130 deg, and a cap, presumably of equal arc, with grooves of various widths and depths. A fair approximation to total useful bearing area might be 50 per cent that of a 360-deg bearing of similar size. Various clearance ratios were used in the tests and, in each bearing, the clearance ratio was less in the vertical than in the horizontal diameter. Among others, these bearings are compared with (a) 360-deg bearings assumed to be so lightly loaded that the shaft runs exactly in the center of the bearing, and (b) a 120-deg bearing without a cap, running under optimum friction conditions. Since the assumed load of 150 psi is relatively light, it is quite likely that at higher speeds the assumption of concentric running is justified. Hence, a comparison under (a) should be made for equal bearing areas and equal clearance ratios. Since the eccentricity is small, comparison under (b) should not be made at the optimum eccentricity ratio, which for the case in point is 0.612. The clearance ratio also enters this picture, although it has been neglected in the authors' Equation [21].

It was interesting to note that power loss varies with the quantity of oil supplied to the bearing. In lightly loaded bearings the power loss is closely proportional to the mean film viscosity. Increase in oil supply results in better cooling, thus increasing the mean film viscosity. From this point of view the addition of Q to the parameter ZN/p might be unnecessary, since, presumably Z represents mean film viscosity.

Experimental data on power losses in high-speed journal bearings are seldom found. The authors are to be congratulated on their practical contribution.

F. W. Kavanagh.[17] This paper is a definite contribution to our knowledge of journal-bearing lubrication, and will be of real value both to designing and lubrication engineers. The extremely high speeds used and the practical commercial bearing sizes employed add both to the interest and the value of the paper.

The authors have introduced a new function ZNQ/p, which is shown by their data to have greater utility than the function previously used ZN/p. It is of interest that ZNQ/p versus f forms straight-line curves with logarithmic coordinates, whereas, to obtain a similar correlation of ZN/p versus f, most investigators have employed rectangular coordinates. However, the authors state that they obtained a straight-line relationship on log-log paper by plotting f versus ZN/p at each value of unit pressure.

On the basis of dimensional analysis the expression $f = K\,\dfrac{ZN}{p}$ can be balanced as follows

$$f = \frac{F}{W} = \frac{MLT^{-2}}{MLT^{-2}} = 0$$

$$ZN = \frac{(ML^{-1}\,T)(T^{-1})}{ML^{-1}\,T^{-2}}$$

[15] Chief Engineer, Canadian Allis-Chalmers, Ltd., Toronto, Canada. Life Member A.S.M.E.

[16] Research Engineer, Kingsbury Machine Works, Inc., Philadelphia, Pa. Mem. A.S.M.E.

[17] Research Engineer, Standard Oil Company of California, Richmond, Calif.

M = mass

L = length

T = time

On the other hand, if Q is included, the dimensions of f would be $MLT^{-2}/T = ML/T^{-3}$. However, it is obvious from the charts presented that the data obtained correlate well with the ZNQ/p variable and this is considered of greater importance than any theory.

It is difficult to explain why Q, the quantity of oil supplied to the bearing, should have an independent effect on the friction or power loss. If we assume that, for all operating conditions used, or in other words for all values of Q, the bearing was completely filled with oil, then additional oil would influence only the end effects. If, however, the bearing were not completely filled, additional oil would tend to increase the amount of oil in the bearing and increase friction by increasing the amount of shearing of the oil film.

It can also be theorized that, under the very high operating speeds employed, the oil flow in the bearing does not follow the usual pattern of viscous or laminar flow. In this case, changes in quantity of oil supplied to the bearing could easily influence the amount of turbulence and therefore affect friction directly. Any explanation that the authors could supply regarding reasons why Q, the quantity of oil, should influence friction would be of interest. It might also be well to mention the limits of application of the formulas including Q, because they would not be suitable for a bearing that operated with an intermittent oil supply or for long periods with no additions of oil.

H. M. Otto.[18] The authors are to be complimented for their presentation of important original data and novel form of power-loss formula.

This formula (in terms of horsepower)

$$HP = 3.77 \times 10^{-3}\, d^{1.65}\, l^{0.55}\, (N/1000)^{1.43}\, Z^{0.43}\, Q^{0.43} \ldots [22]$$

is convenient where the flow Q has already been measured or fixed.

However, the following form may be derived which may be of greater convenience when the designer wishes to determine the loss for a given temperature rise.

$$HP = kQS\,\Delta t \ldots\ldots\ldots\ldots [23]$$

Where S is the specific heat of the oil in Btu per gallon per deg F, M.T.D. and Δt the temperature rise, deg F. Substituting Equation [23] in Equation [22], inserting the proper constants, and solving, we obtain

$$HP = 0.94 \times 10^{-3}\, d^{2.72}\, l^{0.966}\, (N/1000)^{2.51}\, Z^{0.755}\, (S\,\Delta t)^{-0.755}$$

Equation [22] may be occasionally difficult to manipulate due to the interdependence of the variables Z and Q. Further experimentation may divulge, however, a form of equation free from this difficulty, such as taking Z at the inlet or outlet conditions.

J. F. Spiegel.[19] The authors mention the theoretical formula $f = \phi_4\left(\frac{ZN}{p}, \frac{c}{d}, \frac{l}{d}\right)$ derived by Hersey. From this formula it follows that, for a bearing of given geometrical conditions, the coefficient of friction f is a direct function of the term ZN/p, which means that, for all operating conditions of a given bearing, the values of f can be represented by a single curve. Experimental investigations, as for instance those by the McKee's[20] have shown that under conditions of perfect lubrication this actually is the case, the deviations being within reasonable limits.

In the light of these facts, it seems somewhat surprising that the data for f, as found by the authors, vary widely for given values of ZN/p being evidently influenced by changes of the unit pressure p and the rate of oil supply Q.

For this apparent contradiction to the theoretical conclusions an explanation can be offered in the opinion of the writer. As stated by the authors, the values of the mean operating viscosity have been based on the average of the inlet and outlet temperatures of the oil. A closer examination of the operating conditions, however, reveals that the inlet and outlet temperatures alone do not give sufficient indication of the actual temperature of the film.

In all test cases, the oil is fed to the upcoming side of the journal. As soon as it enters the axial groove alongside the journal, a mixture takes place with the hot oil discharged from the loaded section of the bearing. This mixture of hot and cold oil travels through the top half, being further heated up due to the friction losses there, and then reaches the longitudinal groove on the downgoing side. Out of this groove the journal draws in a certain amount of oil, the remainder being discharged to the outlet pipe. During the travel through the loaded section a certain percentage of the oil is squeezed out through the ends of the lining due to side leakage.

From the foregoing, it follows, not only that the average quantity of oil flowing through the loaded section is as a rule much smaller than the gross quantity of oil fed to the bearing, but also that the effective supply and discharge temperatures are necessarily much higher than the temperatures measured. Consequently, the actual mean operating viscosity in the loaded section is considerably smaller than that derived by the authors.

The exact calculation of the mean operating temperatures encounters considerable difficulties due to the complex interrelation of all the variables in question. It is, however, possible to make some approximations which simplify the calculation without involving serious errors. Preliminary studies in this direction have been made by the writer and, if the viscosities derived hereby are introduced into the term $ZNQ/1000p$ then also the theoretical curves show variations of f due to changes of the unit pressures, and the values of f approach fairly closely the experimental results found by the authors for given operating conditions. Of course, when plotting f directly versus ZN/p a single curve is obtained irrespective of the variations of Q and p. Thus, it is proved that also in the case of high-speed bearings, the simple theoretical relations hold true.

It is readily admitted that the empirical formula developed by the authors is of great usefulness in most practical cases; however, it is felt that in extreme cases it will be advisable to resort to thorough theoretical studies in order to gain a clear picture of the actual working conditions. A correct determination of the actual minimum thickness of the oil film, which is a criterion of the safety of the bearing, is not possible without knowledge of the actual average viscosity of the oil.

In this connection it is suggested that, in future experimental investigations of high-speed bearings, measurements be taken also of the temperature of the bearing lining inside the loaded section by means of thermocouples or other suitable instruments.

[18] Turbine Engineering Department, Control Section, General Electric Company, Schenectady, N. Y. Jun. A.S.M.E.

[19] Design Engineer, Kingsbury Machine Works, Inc., Philadelphia, Pa.

[20] "The Effect of Running-In on Journal-Bearing Performance," by S. A. McKee, *Mechanical Engineering*, vol. 49, 1927, pp. 1335–1340.

"Friction of Journal Bearings as Influenced by Clearance and Length," by S. A. McKee and T. R. McKee, Trans. A.S.M.E., vol. 51, 1929, paper APM-51-15, pp. 161–166.

Such data would be of help in verifying the results of theoretical studies.

C. S. L. Robinson.[21] The authors have confirmed the fact that the quantity of oil supplied to a bearing affects the power losses. It might be pointed out, however, that an analysis of the results can be made in terms of dimensionless groups.

Retaining the original nomenclature, assume $L = \phi(Q, p, Z, N, d, l, c)$.

The specific weight of the oil is not included. This assumes that not much work is done in lifting the oil from a lower level to a higher one, and that the kinetic energy of the oil is not important. There may be energy expended in pumping oil against a pressure change, but this depends upon the volume, not the weight, of oil flow. This precludes the possibility of a Reynolds number criterion.

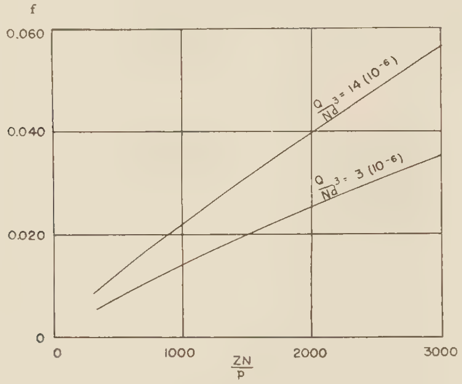

Fig. 15 Coefficient of Friction for Square Bearings (l/d = 1) for Largest and Smallest Values of Q/Nd^3 Tested

Applying the pi theorem of dimensional analysis

$$L = \text{constant} \times pNd^3\phi(Q/Nd^3,\ ZN/p,\ l/d,\ c/d) \text{ or}$$
$$f = \phi(Q/Nd^3,\ ZN/p,\ l/d,\ c/d)$$

The new dimensionless group[22] Q/Nd^3, might be called the "specific oil flow" or the "specific quantity of oil." It is interesting to compare it with the dimensionless source strength used by Muskat and Morgan[23]

$$q_0 = 3ZQr^2/\pi c^3 W \text{ which can be factored into}$$
$$q_0 = \text{constant} \times (ZN/p)(Q/Nd^3)(l/d)(d/c)^{24}$$

Furthermore, the test results obtained by the authors are consistent with those given by Muskat and Morgan, where $(r/c)f$ was plotted against $(r/c)^2\ ZN/p$ for various values of q_0.

The accompanying curve Fig. 15 shows the approximate magnitude of the effect of Q/Nd^3 on the coefficient of friction. This is based on some of the authors' original data for a 3 × 3-in. bearing and a 4 × 4-in. bearing.

The writer is indebted to the authors for permitting the use of their results in the foregoing discussion.

L. M. Tichvinsky.[25] The writer is greatly interested in this paper because of a similar experience of testing a high-speed bearing which was described in a previous paper.[26]

Under the heading, "Effect of Top-Half Grooving on Loss," there is a statement which, if substantiated, seems of importance. It is: "The effect on the power loss was not measurable in these tests, the reason for this being that a vacuum occurs over a large portion of the upper half." Did the authors measure this vacuum and if so what figures have they obtained? The losses in the upper half which is not carrying any load must not be overlooked especially if the width of the upper shell groove is $^2/_3$ that of the bearing length, as mentioned in the paper. In an endeavor to segregate the upper-half and relief losses from the lower-part losses, the following was done in the case of the 7 × $10^1/_2$-in. bearing under 202 psi pressure at 3600 rpm:

For various oil viscosities and a flow of oil of 7.5 gpm the losses were calculated separately for the two 90-deg reliefs, the upper half and the lower half.[27] The average relief clearance was taken from the bearing drawing. The clearance in the upper half was calculated by figuring the minimum oil-film thickness in the lower half and subtracting it from the total diametral clearance. On the curve, Fig. 16 of this discussion, the total calculated losses, composed of losses in reliefs, upper half and lower half are compared with losses measured during tests. It is seen that the difference between the measured and calculated losses is greater for higher values of ZN/P. The individual losses also increase with speed. In the case of the bearing tested, the calculated losses in the upper half and in the reliefs represent each about 20 per cent of the total losses so that the losses in the lower half amount to 60 per cent of the total bearing losses.

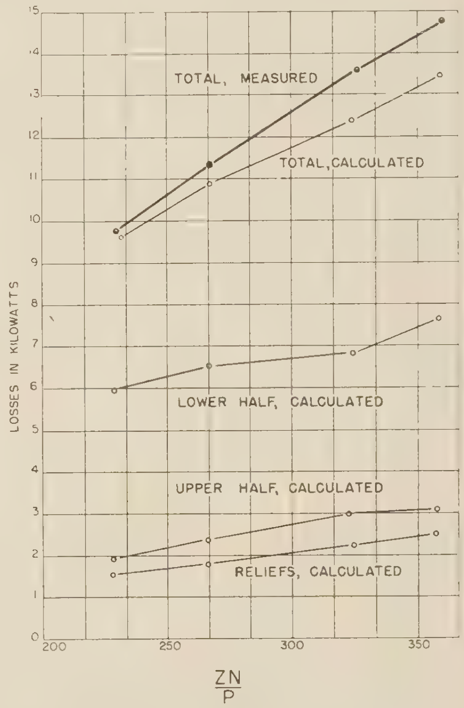

Fig. 16 Distribution of Losses in a 7 × $10^1/_2$-In. High-Speed Bearing

(Z = [15 — 22] centipoises; N = 3600 rpm; P = 202 psi; Q = 7.5 gpm.)

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[21] Gear Engineering Department, General Electric Company, River Works, West Lynn, Mass. Mem. A.S.M.E.

[22] Reference (4) of paper, p. 84.

[23] "The Thick-Film Lubrication of Full Journal Bearings of Finite Width," by M. Muskat and F. Morgan. Trans. A.S.M.E., vol. 61, 1939, p. A-117.

[24] The units of ZN/p are centipoises × rpm/psi; and those of Q/Nd^3 are gpm/rpm × (in.)^3.

[25] U. S. Naval Engineering Experiment Station, Annapolis, Md.

[26] "Tests of a 7 by $10^1/_2$-Inch Bearing at 3600 Rpm," by L. M. Tichvinsky, Trans. A.S.M.E., vol. 60, 1938, pp. 393–397.

[27] For the method of calculation refer to: "Journal Bearing Performance," by R. Baudry and L. M. Tichvinsky, Trans. A.S.M.E., vol. 57, 1935, p. A-121.

C. D. Wilson.[28] In their paper, the authors take into account the quantity of oil flowing through the bearing. This is an important factor which has been long neglected in calculating bearing power losses. Over 2 years ago, the writer conducted many power-loss tests on high-speed bearings. One of the first things noticed was the considerable effect that the oil flow had upon the power loss. When the test data were plotted in the conventional manner (i.e., coefficient of friction as a function of ZN/p), it was found that a separate and distinct curve was obtained for each rate of oil flow to the bearing.

In large high-speed turbine bearings, more oil is usually circulated than the minimum required for stable operation. This is done in order to provide a large factor of safety and to keep the operating temperatures within the limits of current practice. Much of the excess oil supplied to the bearing spills out the ends without passing through the load-carrying portion of the bearing. When the oil flow to the bearing is changed, a greater or less percentage of the oil is by-passed in this way. This undoubtedly

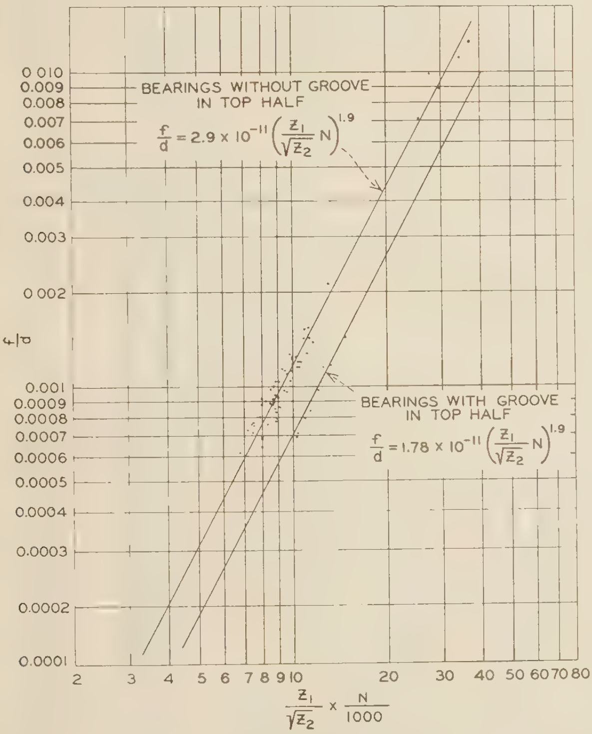

Fig. 17 Results of Allis-Chalmers Tests on 23 Different Bearings

results in a different temperature distribution inside the bearing for each rate of oil flow and makes the determination of the mean viscosity of the oil film extremely difficult. The authors in their paper have taken this into account by modifying the viscosity value of the oil at the average bearing temperature by the actual oil flow in gallons per minute. The writer, in correlating his own test data, found that the relation $f = \phi(ZN/p)$ agreed well with the test results when the viscosity Z was expressed as the ratio of the oil-outlet viscosity Z_1 divided by the square root of the oil-inlet viscosity Z_2. By expressing the viscosity term as a function of both the inlet and outlet viscosities, the temperature rise in the bearing is taken into account, and test results obtained with different oil flows were found to be consistent for the same bearing.

In order to check test data obtained with bearings of different diameters and loads, however, it was found necessary to modify further the ZN/p relation by multiplying it by the diameter d and by omitting the unit load p. This resulted in the empirical relation

$$f = \phi\left(\frac{Z_1 N d}{\sqrt{Z_2}}\right)$$

Test data showing this relation for two different types of bearings are shown in Fig. 17 of this discussion. One curve

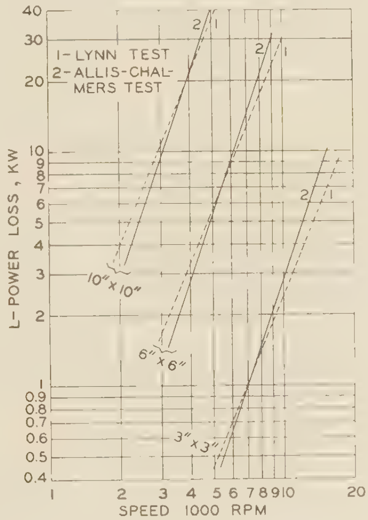

Fig. 18 Comparison of Lynn Bearing Test With Allis-Chalmers Bearing Test

(p = 150 psi; Z = 13 centipoises.)

shows the relation for bearings with a groove cut in the top half of the bearing and the other curve shows the relation for similar bearings without the groove. The test data represent tests on twenty-three different bearings, ranging in size from $2^1/_2$ in. diam × $3^1/_2$ in. long, to 17 in. diam × 18 in. long, running at speeds between 1500 rpm and 8000 rpm. Oil flows were varied from 1.2 gpm in the smaller bearings to over 100 gpm in the larger bearings. One series of tests was made on a 12-in-diam bearing running at 3600 rpm with various oil flows, ranging from 16 gpm up to 100 gpm, so as to study the effect of oil flow on the power losses. Four different oils having Saybolt Universal viscosities of 150, 210, 350, and 560 at 100 F were used.

All of the tests were made on commercial bearings operating in standard pedestals as set up for regular shop tests. Power losses were determined by the heat-balance method. Inlet and outlet temperatures were measured by test thermometers and oil flows were measured with positive-displacement oil meters. Bearing loads were limited by commercial practice to between 60 psi and 175 psi. Referring to Fig. 17 of this discussion, it is interesting to note the reduction in the coefficient of friction for a constant temperature rise and constant speed when a groove is cut in the top half of the bearing. For a constant oil flow, tests showed that the power loss in a 12 × 12 in. bearing running at 3600 rpm was reduced by more than 20 per cent when a groove was cut in the top half of the bearing and the bearing was retested under otherwise identical conditions.

The power-loss formula for bearings with a groove cut in the

[28] Steam Turbine Engineering Department, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

top half, as based on the writer's test results, is expressed (in the notation of the paper for comparison) as follows

$$L = 7.91 \times 10^{-6} \times d^3 \times l \times \left(\frac{N}{1000}\right)^{2.9} \times \frac{Z_1^{1.9}}{Z_2^{0.95}}$$

Fig. 18 of this discussion compares the calculated results obtained using this formula with the Lynn calculated results for a 30 F rise and a viscosity of 13 centipoises as taken from Fig. 14 in the paper.

The comparison shows exact agreement between the writer's results and the Lynn results at 3600 rpm for the 10-in-diam bearing, at 5500 rpm for the 6-in-diam bearing, and at 7500 rpm for the 3-in-diam bearing. Although the slopes of the two curves are different, they show closer agreement with each other than with the other calculated results in Fig. 14 of the paper.

The writer agrees with the authors in most of their conclusions regarding power losses in high-speed journal bearings. Over the limited range of unit loads tested, no variation in the value of the coefficient of friction with change in load could be detected. The power loss, however, appeared to vary directly with the total load on the bearing. The writer's tests also indicated that the width of the groove in the top half of the bearing greatly affected the power loss. This was especially true with the larger oil flows which tend to reduce the percentage of vacuum in the top half of the bearing.

It is interesting to note that two independent investigations, in which high-speed bearings were tested under a wide range of conditions, both indicate that there are additional factors which must be considered in order to achieve a closer agreement between existing theory and the actual power losses.

Authors' Closure

In Mr. Needs discussion, he calls attention to the comparisons, made in Fig. 14 of the paper, which comparisons are not accurate since some of the curves are for optimum conditions, whereas others are based upon 150 psi loading. Fig. 19 of this closure, makes a comparison of the various formulas based upon a load of 150 psi, and an absolute viscosity of 13 centipoises. Regarding curves 3 and 5 the following assumptions have been made:

Bearings, in.	Top half groove width, in.	Clearance ratio, mils per in.
10 × 10	4	1.3
6 × 6 below 6000 rpm	2	1.3
6 × 6 over 6000 rpm	2	2
3 × 3	1	2

NOTE: Arc of contact: 120 deg for top half, and 120 deg for bottom half.

It is interesting to note the close agreement of Petroff's fundamental equation with the results given in the paper at a viscosity of 13 centipoises. At other viscosities the agreement will not be quite as good.

In a number of the discussions, the advisability of the use of "flow of oil" through the bearing has been questioned or has been favorably commented upon. It is quite possible that the flow of oil through the bearing affects other variables so that, if the effect were known, it would be possible to disregard the factor "flow of oil." However, so far as the design of bearings for use in service is concerned, the question of flow of oil is extremely pertinent. Journal bearings are usually designed on the basis of supplying a quantity of oil which will allow from 30 to 35 F temperature rise as the oil passes through the bearing. A vacuum usually exists in the unloaded portion of high-speed bearings, i.e., there is insufficient oil supplied to fill the bearing completely. Under this condition, it can readily be seen that the quantity of oil will increase or decrease the arc in which the vacuum exists. A smaller quantity of oil will give vacuum over a greater arc. On this basis, we would expect that, as the quantity of oil increased, the bearing loss would increase due to the oil being sheared for a greater arc.

There is, in general, a misconception as to the quantity of oil which flows through a bearing that operates at high speed. If

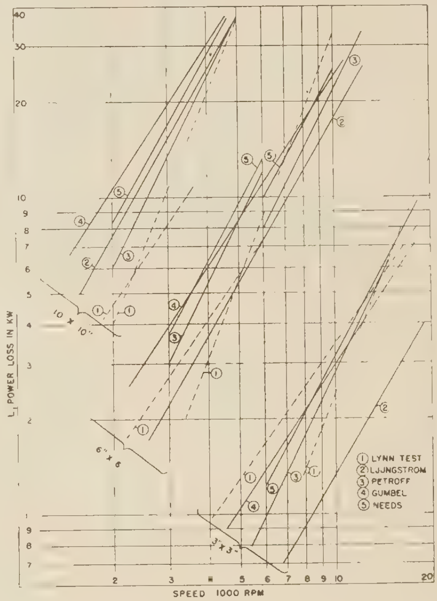

Fig. 19 Comparison of Lynn Bearing Test With Theoretical Calculations

(P = 150 psi; Z = 13 centipoises.)

TABLE 3 TEST DATA FOR FIG. 3 OF PAPER

(Bearing 6 in. × 8 in.; vertical clearance 0.008 in.; horizontal clearance, 0.015 in.; two 1/16-in. × 1-in. circumferential grooves in top half; feed at horizontal joint at upcoming side of journal)

Bearing-oil temp, F, In t_1	Bearing-oil temp, F, Out t_2	Flow, gpm Q	Load, psi P	Speed, rpm N
214.0	236.5	5.5	355	4800
219.0	234.0	6.0	100	4800
219.0	236.5	5.96	200	4830
216.0	232.0	7.0	200	4800
210.0	234.5	3.43	200	4800
212.0	233.0	3.13	100	4800
213.0	233.0	3.50	51	4800
212.5	236.0	5.0	355	4800
212.0	236.0	4.7	355	4820
214.0	255.5	0.97	355	3600
210.0	237.0	2.2	355	3600
214.0	236.5	2.73	355	3600
217.0	235.0	3.88	355	3600
213.0	247.0	1.50	355	3600
216.0	231.0	3.97	200	3600
216.5	234.0	2.69	200	3600
214.5	242.0	1.50	200	3600
211.5	233.0	1.50	100	3600
212.0	228.0	2.55	100	3600
211.0	229.7	1.45	51	3600
168.5	195.6	6.10	355	4800
167.0	192.2	7.03	355	4800
166.5	191.7	5.88	200	4800
162.5	185.0	7.22	100	4800
163.5	184.5	7.40	100	4840
161.5	183.8	7.15	51	4800
200.5	231.0	3.88	100	6000
202.8	236.5	4.97	355	6000
194.5	215.5	9.25	100	6000
197.5	220.0	9.10	200	6000
205.0	222.0	5.75	100	4800
207.5	225.8	5.96	200	4800
163.2	187.0	5.94	51	4800
161.9	186.5	5.86	100	4800
161.0	191.1	3.74	51	4800
161.0	191.5	3.75	100	4800
162.0	197.0	3.59	200	4800
156.0	195.7	4.75	200	6000
158.3	187.6	7.15	100	6000
158.8	190.2	6.97	200	6000
157.5	185.5	9.09	51	6000
157.0	184.5	9.00	100	6000
154.9	184.0	8.98	200	6000
155.0	186.8	8.71	355	6000

TABLE 4 TEST DATA FOR FIG. 4 OF PAPER

(Bearing 8 in. × 6¼ in.; vertical clearance 0.01 in.; horizontal clearance, 0.02 in.; $^1/_{32}$-in. × $1^1/_4$-in. diagonal groove in top half; feed at horizontal joint at upcoming side of journal)

Bearing-oil temp, F, In t_1	Bearing-oil temp, F, Out t_2	Flow, gpm Q	Load, psi P	Speed, rpm N
213.0	235.0	7.17	130	4800
221.5	239.0	11.90	130	4800
218.0	236.5	9.18	60	4800
213.0	234.0	7.02	60	4800
212.5	235.0	11.78	500	4800
212.5	235.0	11.70	500	4800
213.0	235.0	11.95	500	4800
213.5	239.0	9.46	500	4800
213.0	238.5	9.50	500	4820
216.7	241.5	9.36	500	4800
223.0	250.0	9.38	500	4850
214.5	229.5	6.93	130	3560
215.0	229.5	6.95	130	3600
222.2	239.5	4.88	130	3600
211.0	234.0	2.75	60	3600
213.5	229.5	4.90	60	3600
214.0	228.0	7.06	60	3600
168.5	202.0	9.27	500	4800
169.0	198.5	11.63	500	4800
167.0	205.0	7.34	500	4800
167.0	193.5	12.62	276	4800
166.0	197.5	9.20	276	4800
162.0	191.0	9.52	130	4800
208.0	218.0	15.20	60	3600
219.0	230.5	15.90	276	3600
180.2	195.2	9.90	60	3600
201.0	215.0	10.10	130	3600
210.2	225.0	10.20	276	3600
210.5	223.5	7.15	60	3600
212.5	228.0	6.94	130	3600
219.0	237.5	6.82	276	3600
206.0	226.0	9.98	60	4800
205.5	226.5	10.20	130	4800
208.0	231.1	9.89	276	4800
163.6	190.5	10.20	60	4800
162.0	189.5	10.20	130	4800
163.8	193.0	10.00	276	4800
166.8	199.3	9.83	500	4800
161.0	194.0	6.03	60	4800
161.0	197.0	5.90	130	4800
162.0	202.0	5.70	276	4800
160.0	204.0	5.83	500	4800
167.5	185.5	8.30	60	3600
161.0	181.5	8.14	130	3600
163.1	184.1	8.90	276	3600
161.0	185.5	8.65	500	3600
163.1	181.1	11.20	276	3600
162.0	182.3	10.88	500	3600
154.0	188.5	5.57	500	3600
158.5	187.0	5.64	276	3600
163.8	183.0	8.62	130	3600
158.5	199.5	7.79	60	6000
157.6	198.0	8.33	130	6000
152.0	204.0	7.52	500	6000
155.4	198.9	8.50	276	6000
154.8	191.0	11.00	60	6000
158.0	194.0	11.00	130	6000
158.6	197.0	10.75	276	6000
156.9	198.0	11.10	500	6000
157.5	190.0	15.70	60	6000
157.0	189.5	16.05	130	6000
155.0	189.0	16.05	276	6000
155.0	191.0	15.85	500	6000
165.2	187.0	15.23	60	4800
168.0	189.0	16.35	130	4800
162.0	187.0	14.15	276	4800
157.0	185.0	13.70	500	4800

we assume that the film thickness at the point of maximum loading for a 6 × 6-in. bearing when running at 3600 rpm is 3 mils, there will be 2.65 gpm passing between the journal and the liner in the loaded portion. This quantity of oil fed to the bearing will give a temperature rise of 21 F if the average viscosity of the oil is 13 centipoises. If the quantity of oil is reduced so as to allow for a 30-deg temperature rise, it can readily be seen that the amount of oil which will be fed to the lining will be such as to allow it to pass through the loaded portion of the bearing more than once before being discharged. In this respect, it is expected that the average between the inlet and outlet temperature will not be greatly different from the average film temperature.

Mr. Tichvinsky questions the magnitude of the vacuum which occurs in the top half of the bearing. Measurements of 10 to 20 in. Hg have been obtained on 8-in. and 10-in. bearings running at 3600 rpm.

Mr. Nagler asks: "At what point of length-to-diameter ratio does the self-aligning feature cease to become operative?" There is no specific answer that can be given to this question due to the fact that the self-aligning feature of the bearing is dependent upon the pinch fit between the lining and the bearing housing. We have had successful operation with the effective length of the bearing equal to 75 per cent of the diameter.

Mr. Nagler's question in regard to water cooling of bearings is one which undoubtedly raises points of controversy among engineers. In general, it is more difficult to babbitt a lining with water-cooling coils in it. There is greater likelihood of having the babbitt crack with cooling coils than without. A greater quantity of babbitt is required with cooling coils than without and, if cooling coils break, water usually gets into the lubricating system and may cause very serious results.

Most designers of high-speed apparatus have gotten away from the use of cooling coils and are using a sufficient flow of oil through the bearings to limit the temperature rise. The oil is passed through an oil cooler in order to maintain a constant inlet temperature to the bearings.

It is gratifying to note that the results of Allis-Chalmers tests conducted by Mr. Wilson have agreed with our results as closely as they have. Our data for a given bearing when plotted in the form illustrated in Fig. 17 are not properly represented by a single curve. Term f will vary 100 per cent for constant values of $(Z_1N)/(1000\sqrt{Z_2})$. A visual inspection of Fig. 17 shows a variation of 30 per cent between the various test points. This undoubtedly is not due to the running of tests but rather due to the method of presentation.

In accordance with Mr. Kerster's request, we are presenting a portion of the test data for the points plotted in Figs. 3, 4, and 5 of the paper in Tables 3, 4, and 5, respectively.

TABLE 5 TEST DATA FOR FIG. 5 OF PAPER

(Bearing 3 in. × 3 in.; vertical clearance 0.006 in.; horizontal clearance, 0.009 in.; $^1/_{32}$-in. × $1^3/_4$-in. circumferential groove over the top half; feed is on the horizontal joint on upcoming side of shaft)

Bearing-oil temp, F, In t_1	Bearing-oil temp, F, Out t_2	Flow, gpm Q	Load, psi P	Speed, rpm N
161.0	176.0	3.10	56.9	7900
165.0	179.0	3.20	90.1	8050
169.0	183.0	3.20	220.0	8050
152.5	171.0	2.92	349.0	8000
159.0	178.5	2.85	521.0	7900
151.1	180.5	1.65	521.0	7900
171.5	195.0	1.75	349.0	8150
165.5	186.5	1.75	220.0	7900
163.5	184.0	1.75	90.1	8000
161.5	183.0	1.68	56.9	8000
160.5	184.0	4.07	220.0	12000
164.0	188.0	4.15	349.0	12000
156.0	184.0	4.00	521.0	12100
160.0	194.0	2.75	521.0	12200
162.5	196.0	2.35	349.0	12000
156.5	191.0	2.15	220.0	12000
154.5	189.0	2.15	90.1	12000
155.0	191.0	2.07	56.9	12100
155.0	181.0	3.70	56.9	12000
158.5	182.5	3.68	90.1	11900
159.5	166.0	1.75	56.9	3840
157.0	170.0	0.56	90.1	3820
173.5	191.0	0.61	349.0	4080
145.0	174.5	0.52	521.0	4150
166.0	182.5	1.04	521.0	4190
162.5	177.0	1.00	349.0	4140
171.5	182.0	1.10	220.0	4100
169.0	178.0	1.06	90.1	4160
166.0	174.0	1.05	56.9	3970
(Bearing 3 in. × 3 in.; vertical clearance 0.006 in.; horizontal clearance, 0.009 in.; no groove in top half; oil feed to both sides at horizontal joint)				
152.0	197.5	0.60	56.9	8000
154.0	201.0	0.60	84.5	8000
158.0	205.0	0.72	220.0	8100
163.5	208.0	0.90	349.0	8000
168.5	209.0	1.07	521.0	8100
(Bearing 3 in. × 3 in.; vertical clearance 0.006 in.; horizontal clearance, 0.015 in.; $^1/_{32}$-in. × $1^3/_4$-in. circumferential groove over the top half; feed is on the horizontal joint on upcoming side of shaft)				
168.5	181.0	3.40	220.0	7900
176.0	189.5	3.65	349.0	8190
170.0	185.0	3.50	521.0	8000
157.0	172.5	3.35	56.7	9000
156.5	172.0	3.30	90.1	8600
148.0	168.5	1.90	56.7	8340
158.5	177.5	2.15	220.0	8450
158.0	178.0	1.95	349.0	7760
165.0	187.0	2.00	521.0	8130

Flow Properties of Lubricants Under High Pressure

By A. E. NORTON,[1] M. J. KNOTT,[2] and J. R. MUENGER[3]

In this paper, results are given of a preliminary study of the rate-of-shear versus shear-stress relationship for several oils known to undergo apparent solidification when subjected to high pressure. Lard, rapeseed, sperm, and one mineral oil were tested under a temperature range of —5 C to 20 C while subjected to pressures up to 50,000 psi. Experimental curves of flow versus pressure difference were obtained for capillary flow, and these curves were transformed mathematically to the desired curves of rate of shear versus shear stress. A brief discussion of some of the problems inherent in capillary testing of plastic materials is included in this report.

FOREWORD BY M. D. HERSEY[4]

The following contribution is one of a series of investigations on the properties of lubricants under high pressure, conducted by the Special Research Committee on Lubrication of the Society. These studies were begun at Harvard University in 1915, and reported in various publications dating from 1916. They were more completely outlined in a paper by Henry Shore and the writer at the Annual Meeting of the Society in 1927 (1),[5] and in a joint paper with R. F. Hopkins (2).

A phenomenon cautiously termed "apparent solidification," produced by increasing the pressure on a lubricating oil at constant temperature, was briefly described in the first of these two papers. Future experiments were recommended in order to determine the flow or shear characteristics of the lubricant—in a word, its consistency—while in that condition. Is it a hard solid like that formed by the freezing of water into ice, or a soft jelly more like an oil at its pour point? And how does its consistency vary when the pressure is increased well beyond the critical value for solidification?

This phenomenon was confirmed by Robert Kleinschmidt at Harvard (3) and by Yoshio Suge in Tokyo (4). Shore found an empirical relation connecting solidifying pressures with temperatures (5), while Cragoe discussed the question theoretically in the light of Clapeyron's equation (6). It remained for Professor Norton, assisted at the start by Knott and later by Muenger, to carry through the first quantitative measurements.

It appears that castor oil and naphthene-base mineral oils have not been solidified by pressure, and that the only lubricating oils for which pressure solidification has been reported are lard, horse, rapeseed, whale (including sperm), mineral oils containing sufficient paraffin wax, and compounded oils. We may add to this list crude petroleum, oleic acid, and any pure substance whose freezing points have been determined under pressure.

The experiments to be described constitute a preliminary phase of the project. They are reported at this time to provide a record of the work accomplished under Professor Norton's supervision. The report was compiled by Mr. Muenger in consultation with Mr. Knott and others concerned. Through the courtesy of Dean Westergaard and Professor Den Hartog of the Graduate School of Engineering, Harvard University, arrangements have been made for the continuation of the research for a limited period.

The principal difficulties outstanding are due to the relatively large pressure differences thus far employed in the observations of capillary flow, and to the further fact that the lubricant under test is neither unworked nor completely worked but is in some intermediate, undefined, partially worked condition. In spite of these uncertainties, the progress report at hand reveals the order of magnitude of the effects in question, thus providing a first approximation to the data required. These results are given in absolute units, with rate of shear plotted against shearing stress, in the last four diagrams of this paper.

Professor Norton believed that such investigations are of educational as well as scientific and industrial value. This will be evident from the closing paragraph in his discussion of a recent paper on "Teaching Lubrication" (7):

"Any graduate course should aim not only to prepare engineers for advancing the science and art of lubrication but also to give these men a unified knowledge of materials. If properly taught, the subject of lubrication can be allied with the study of elasticity and plasticity, especially with the latter, since the rate of shear is an important feature of both liquids and plastic materials."

INTRODUCTION

What is perhaps the most important characteristic of a lubricating material can be defined by the relationship between the shear stress S, applied to the material and the resulting rate of shear R. For example, a Newtonian liquid is a material whose shear behavior can be represented by a straight line passing through the origin. Its viscosity S/R and fluidity R/S are constant at all values of shear stress for any given pressure and temperature. Most lubricating oils at ordinary pressures and temperatures have this type of graph.

Other materials which may be represented by a nonlinear curve, passing through the origin, are known as non-Newtonian liquids. Rubber suspensions fall into this class. Yet other materials whose graphs have intercepts on the axis of shear stress are known as plastic solids. They require an initial value of shear stress known as the "yield" shear stress S_0 to start the flow. If the plastic solid has a straight-line relationship, it is known as a Bingham solid and can be represented by two parameters, its yield shear stress S_0 and "mobility" $R/(S - S_0)$, which is analogous to the fluidity of liquids.

Typical curves for these various types of materials are indicated in Fig. 1. There are other definitions of ideal materials and,

[1] Late Gordon McKay Professor of Applied Mechanics, Harvard University, Cambridge, Mass. Mem. A.S.M.E. Deceased, February 24, 1940.

[2] Brown & Sharpe Manufacturing Company, Providence, R. I. Jun. A.S.M.E.

[3] Assistant in Mechanical Engineering, Harvard University, Cambridge, Mass.

[4] Research Director, Morgan Construction Company, Worcester, Mass. Fellow A.S.M.E.

[5] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

generally speaking, the more parameters in the definition, the more general the definition is.

The influence of pressure on viscosity is well known and has been the subject of previous investigations. These investigations, however, were concerned mainly with oils as Newtonian liquids, and properties beyond the point of apparent solidification were not investigated. Other researches studied greases which, of course, are initially plastic solids. Thus there is a gap in our knowledge of lubricating materials which may hinder a better understanding of lubrication in cases where high local pressures materially alter the nature of the lubricant. The action of an oil

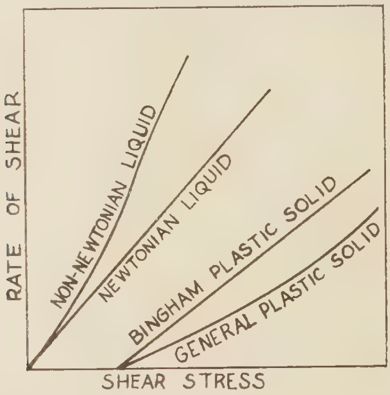

Fig. 1 Shear Characteristics of Various Types of Materials

film in gear teeth and the effect of surface irregularities upon thin oil films are two cases which at present are not well understood.

The purpose of this project was to investigate the properties of certain lubricants at or near the condition of apparent solidification, due to physical conditions. An attempt has been made to determine the characteristics of these lubricating oils which are normally liquid but which may become stiffer due to high pressure or low temperature singly or in combination. These characteristics of the oils may be indicated by curves similar to those of Fig. 1, by tabulation of yield shear stress and mobility in the case of a Bingham solid, or by a mathematical statement of the relationship of rate of shear to shear stress.

Data on Oils at Atmospheric Pressure

For the preliminary studies covered by the present report, four oils were chosen that were known from previous investigations to be subject to apparent solidification under high pressure, namely, lard, rapeseed, sperm, and Veedol medium (SAE 30). Specific gravities and viscosities are given in Table 1.

TABLE 1 PROPERTIES OF TEST OILS AT ATMOSPHERIC PRESSURE

Oil	Specific gravity, 60/60 F	S.U.V. 100 F	S.U.V. 210 F	Viscosity in centipoises 100 F	Viscosity in centipoises 210 F
Lard	0.920	207	54	40.8	7.5
Rapeseed	0.912	230	61	45.3	9.0
Sperm	0.886	108	48	19.2	5.8
Veedol medium	0.885	519	67	100	10.2

Viscosity in pound-seconds per square inch (reyns) may be found from the viscosity in centipoises upon dividing by 6.9×10^6.

The lard oil is Swift's No. 2, the rapeseed and sperm oils were purchased from the Mardin Wild Corporation, Somerville, Mass., in 1938, while the Veedol medium was purchased in the usual sealed can in 1939.

Two of these oils, rapeseed and lard, were tested at atmospheric pressure using a Bulkley and Bitner consistometer loaned by the National Bureau of Standards. These tests were conducted in order to learn whether such oils are noticeably thixotropic and to measure their viscosity, or other consistency constants at low temperatures in a thoroughly worked condition.

The consistometer is well described in a paper by Bulkley and Bitner (8), and it is sufficient to say here that it is an instrument which measures the rate of flow through a capillary produced by a small pressure difference. The material tested may be either liquid or plastic, and provision is made for working the material previous to the test. The apparatus is well adapted for repeating tests quickly, since the only observation necessary for the test is the timing of a standard travel of a mercury column which indicates the displacement of the material tested. The pressure difference and bath temperature are kept constant during repetitions, and the material tested remains in the apparatus.

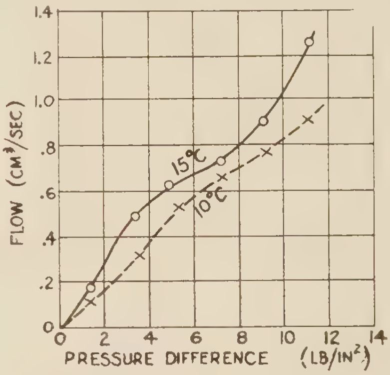

Fig. 2 Flow Curves for Lard Oil at Atmospheric Pressure

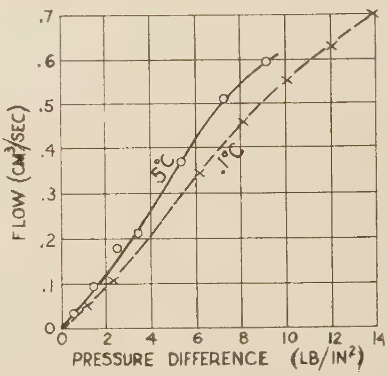

Fig. 3 Flow Curves for Rapeseed Oil at Atmospheric Pressure

Tests were made on rapeseed oil at 5 C and at 0.1 C and upon lard oil at 15 C and 10 C. An average of eight passages through the capillary were timed for each plotted point. This was done in view of the impossibility of obtaining close agreement on individual runs at the lowest rates of shear. The data of these

TABLE 2 DIMENSIONS OF TEST CAPILLARIES

Apparatus	Test material	Temp, C	Data plotted in Fig.	Length, cm	Internal diam, cm
Bulkley and Bitner consistometer (atmospheric pressure)	Lard oil	15	2	7.54	0.163
		10	2	7.54	0.163
	Rapeseed oil	5	3	7.54	0.163
		1	3	7.54	0.163
Long test capillary (high pressure)	Rapeseed oil	0	5	144.5	0.0456
	Rapeseed oil	−5	6	144.5	0.0456
	Sperm oil	0.2	7	144.5	0.0456
	Sperm oil	−5	8	144.5	0.0456
Two capillaries in series (high pressure)	Sperm oil	0	10	16.1	0.0456
	Sperm oil	20	11	88.3	0.0456
	Lard oil	20	12	25.0	0.0456
	Veedol medium	20	13	78.5	0 0456

tests are plotted in Figs. 2 and 3. The dimensions of the test capillaries which were used in this project are given in Table 2.

In the case of lard oil, it was found that when the temperature was low enough to cause apparent solidification, the time necessary for a certain amount of flow decreased with successive passages up to a certain point. That is, the lard oil was unquestionably thixotropic.

From Figs. 2 and 3 it appears that both rapeseed and lard oil are slightly non-Newtonian, but not plastic at the temperatures for which the curves are drawn. Knowing the dimensions of the capillary (length 7.54 cm, radius 0.0815 cm), it is possible to compute the viscosities at low rates of shear. Corresponding to pressure differences not exceeding 1 psi, the viscosities are for lard oil at 15 C, 120 centipoises, and at 10 C, 210 centipoises; for rapeseed oil 240 centipoises at 5 C, and 360 centipoises at 0.1 C.

High-Pressure Tests With a Single Capillary

In proceeding to work at high pressure, apparatus formerly used by Hersey and Snyder (9) was employed. This apparatus

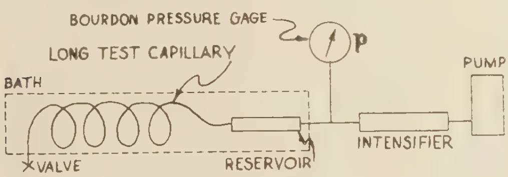

Fig. 4 Diagram of Apparatus for Long-Capillary Tests

forces oil through a long capillary. The pressure is high at the inlet, dropping gradually to atmospheric pressure at the outlet. A diagram of the apparatus is shown in Fig. 4. The procedure is to time the flow of a sample of oil while the inlet pressure is held steady by the pump. The sample is then weighed and the results reduced to curves of rate of flow versus inlet pressure. A series of tests has been run on castor oil with pressures up to 50,000 psi and with temperatures ranging from —10 C to 20 C. These tests check the previous findings that castor oil does not solidify within this range and indicate that the apparatus is capable of reproducing tests to a fair degree of accuracy.

The next tests were conducted upon sperm oil and rapeseed oil at temperatures in the neighborhood of 0 C. Results of these tests are shown in Figs. 5, 6, 7, and 8. These tests showed definite solidification and revealed that the time of application of pressure had a marked effect upon the consistency.

In this work with the long test capillary, the pressure was held for a given period of time before beginning a run. Immediately after a run, the pressure was stepped up 2500 psi and the procedure repeated. In the case of rapeseed oil, it was found that flow slowed down at 25,000 psi and stopped at 35,000 psi when a 10-min period was used. However, when the pressure was held constant for longer periods, the stoppage of flow, indicating solidification, occurred at lower pressure, e.g., when the pressure was held for a period of 2 hr, stoppage of flow occurred at 20,000 psi. This time effect is clearly shown in Fig. 5 where curve 1 represents a 2-hr application of pressure for each point; curve 2 a 30-min application; curve 3 a 10-min application; curve 4 a 10-min application, with the series of runs starting at 20,000 psi; and curve 5 a 10-min application, with the series of runs starting at 25,000 psi.

From the data of Fig. 5, it would seem that the oils are extremely sensitive to time of application of pressure only when they are beginning to solidify. For pressures up to 15,000 psi, the points fell on a smooth curve regardless of length of time the pressure was imposed. The solidification of rapeseed oil due to pressure was confirmed by opening the reservoir after there had been no flow at high pressures. The oil was removed in the form of a hard white cylinder, looking very much like a candle. That this was clearly a pressure effect was evident, for rapeseed oil at 0 C and atmospheric pressure is liquid.

Little has been said about the tests using the long capillary other than to point out time effects. Analysis of the curves from

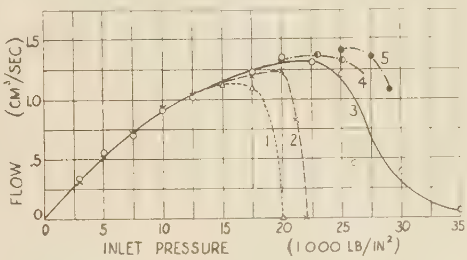

Fig. 5 Long-Capillary Flow Curves for Rapeseed Oil at 0 C

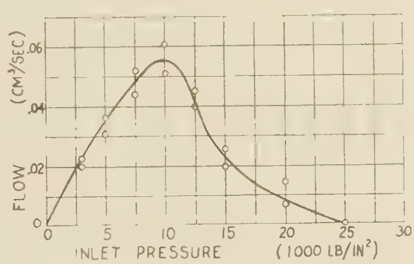

Fig. 6 Long-Capillary Flow Curves for Rapeseed Oil at —5 C

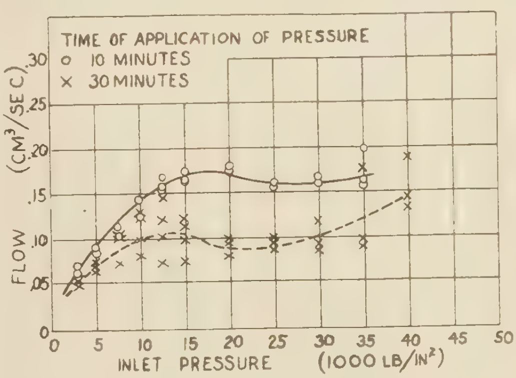

Fig. 7 Long-Capillary Flow Curves for Sperm Oil at 0.2 C

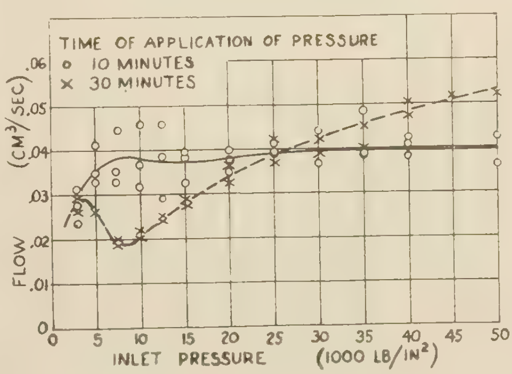

Fig. 8 Long-Capillary Flow Curves for Sperm Oil at —5 C

this work is difficult due to the great variation of consistency along the length of the capillary, and a painstaking study of these curves probably is not justifiable in view of the meager data.

High-Pressure Tests With Two Capillaries in Series

The large drop in pressure along the single test capillary was disadvantageous due to the difficulty of handling the results mathematically. The consistency of the material in the capillary

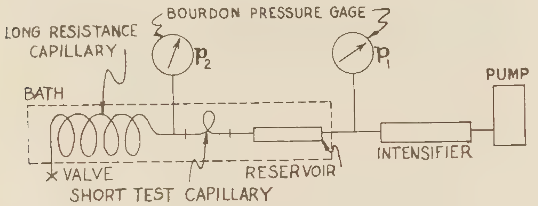

Fig. 9 Diagram of Apparatus for Tests With Two Capillaries in Series

ranged from a plastic solid at the high-pressure end to liquid at the low-pressure end. The first attempt at overcoming this difficulty was to use a needle valve as a pressure reducer at the outlet of the capillary. It was impossible to obtain a steady flow with this device, and the arrangement shown in Fig. 9 was adopted. This consisted of a short test capillary with two Bourdon gages to measure terminal pressures p_1 and p_2 and a long capillary in series with it to offer resistance to flow. The purpose of this was to render the pressure difference in the test capillary, $p_1 - p_2$, small enough so that the oil could be assumed at a constant pressure throughout the test capillary. By using different lengths of resistance capillary, various rates of flow were obtained. Curves were drawn showing rates of flow versus pressure difference for constant average pressures $(p_1 + p_2)/2$. The pressure was maintained for 10 min before taking a run, and it was kept off for 10 min before proceeding to a different pressure. In this way it was hoped to eliminate time of application of pressure as a variable affecting the curves. Sperm oil at 0 C, sperm oil at 20 C, lard oil at 20 C, and Veedol medium at 20 C were tested; the results are shown in Figs. 10, 11, 12, and 13. Each figure has several flow curves, each corresponding to a constant average pressure which is indicated in pounds per square inch along the curve.

Interpretation of Flow-Pressure Curves

Various schemes were tried for the interpretation of these curves. From the curves for sperm oil at 0 C, values of yield shear stress and mobility were found by the use of Buckingham's equation (10) based on Bingham's law and are shown in Figs. 14 and 15. The procedure was to draw a tangent to each curve at its lower left portion and find its intercept P' on the axis of pressure differences. This procedure assumes that such a tangent will be the nearest approximation to the asymptote of Buckingham's cubic equation and that Bingham's law is most nearly realized at the lower rates of shear. Observations at higher rates of shear were disregarded in this analysis. Then, according to Buckingham's equation, the initial pressure difference P_0, causing the flow to start, is equal to $(3/4)P'$. By statics, the yield shear stress S_0 is equal to $(r/2L)P_0$, where r and L are the radius and length, respectively, of the capillary. The mobility was computed by the formula $(8L/\pi r^4) \tan \gamma$, where $\tan \gamma$ is the slope

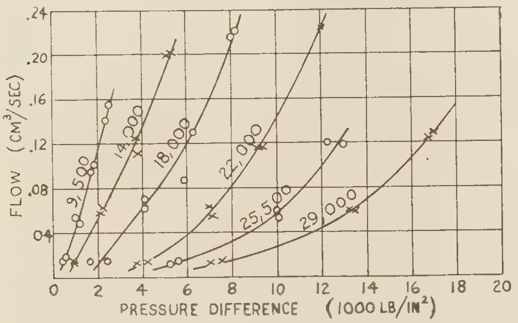

Fig. 10 Flow Curves for Sperm Oil at 0 C
(Average pressures indicated along curves in psi.)

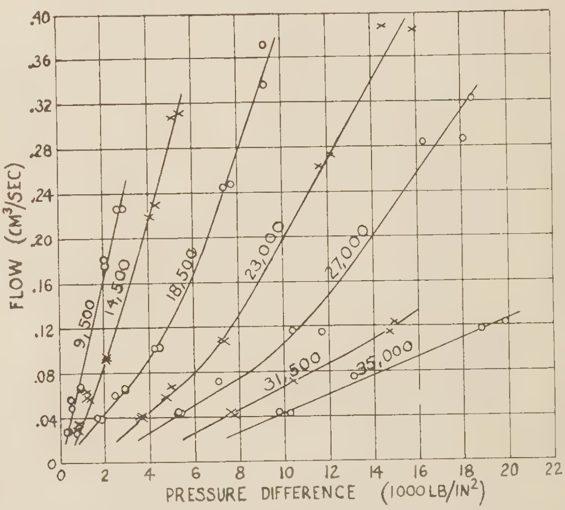

Fig. 11 Flow Curves for Sperm Oil at 20 C
(Average pressures indicated along curves in psi.)

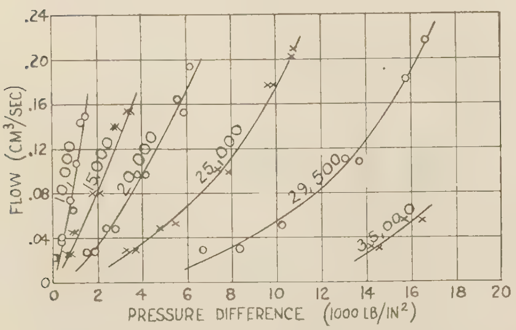

Fig. 12 Flow Curves for Lard Oil at 20 C
Average pressures indicated along curves in psi.)

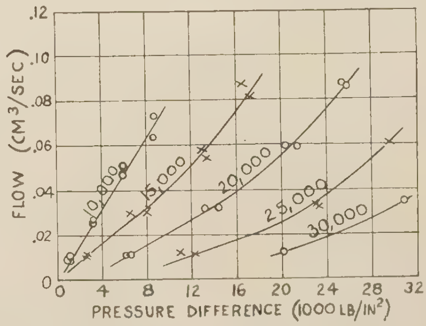

Fig. 13 Flow Curves for Veedol Medium at 20 C
(Average pressures indicated along curves in psi.)

of each curve taken at the lower left portion, where the tangent line has been drawn. This method of expressing the characteristics of the oil was not very satisfactory since the curves apparently do not follow Bingham's law at the higher rates of flow. The plots of yield shear stress and mobility, then, may be thought of as a first approximation in representing the characteristics of sperm oil at 0 C, useful in showing the general effect of pressure. Furthermore, the mobility curve should be used only in a restricted range of rate of shear as indicated on the plot.

Two principal methods have been proposed for the interpretation of flow-pressure curves, and these have been discussed (11) by Hersey. The methods are termed the integration method and the differentiation method. The use of Buckingham's equation

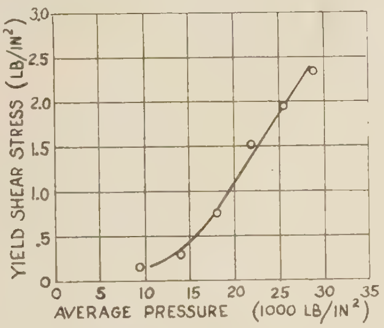

FIG. 14 VARIATION OF YIELD SHEAR STRESS WITH PRESSURE FOR SPERM OIL AT 0 C

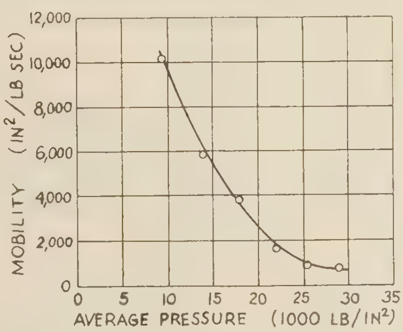

FIG. 15 VARIATION OF MOBILITY WITH PRESSURE FOR SPERM OIL AT 0 C FOR RATES OF SHEAR BELOW 5000 RECIPROCAL SEC

to evaluate the constants of the material was an example of the integration method.

The integration method consists of assuming a particular law governing the behavior of the material in shear and then integrating twice to find the equation of flow. As a second example, if it is assumed that the velocity gradient is proportional to the shear stress raised to the power n, one finds that the flow is proportional to the pressure difference raised to the same power. If the derived equation for flow can be made to fit the observed flow curve, the constants in the fundamental law can be evaluated. This type of approach did not lead to any conclusive results. It may be of interest to note that the curves for sperm oil at 0 C could be fitted fairly well by power curves of increasing order as the average pressure increased. For example, the curve of average pressure = 9500 psi could be fitted by a straight line, average pressure = 18,000 psi by a second-degree curve, and average pressure = 29,000 psi by a cubic curve, to a fair degree of approximation.

The procedure finally used for interpreting the results given by Figs. 10 to 13 was the differentiation method in the form of

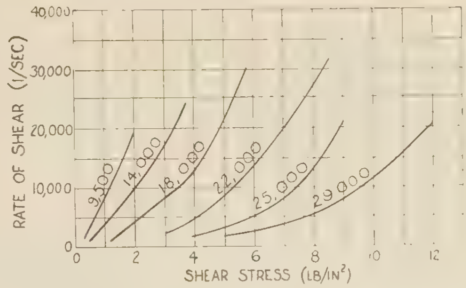

FIG. 16 SHEAR CHARACTERISTICS OF PARTIALLY WORKED SPERM OIL AT 0 C
(Average pressures indicated along curves in psi.)

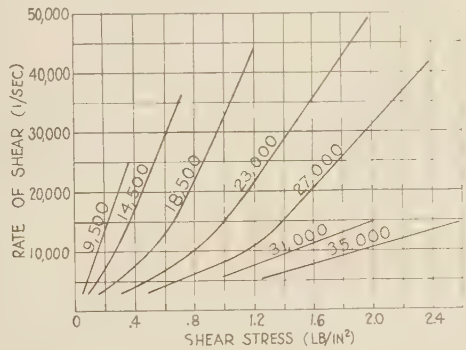

FIG. 17 SHEAR CHARACTERISTICS OF PARTIALLY WORKED SPERM OIL AT 20 C
(Average pressures indicated along curves in psi.)

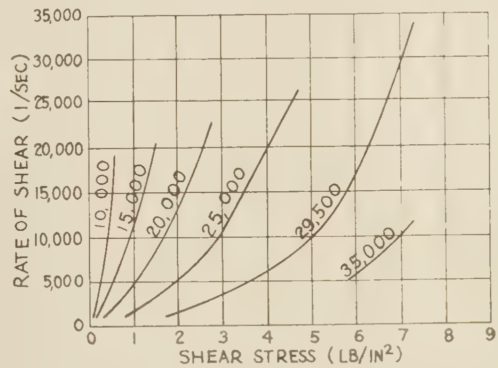

FIG. 18 SHEAR CHARACTERISTICS OF PARTIALLY WORKED LARD OIL AT 20 C
(Average pressures indicated along curves in psi.)

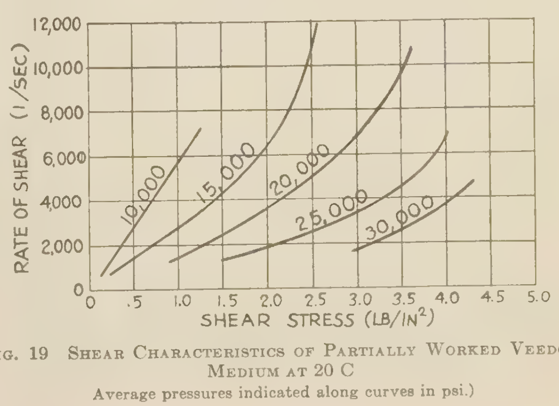

FIG. 19 SHEAR CHARACTERISTICS OF PARTIALLY WORKED VEEDOL MEDIUM AT 20 C
Average pressures indicated along curves in psi.)

the Weissenberg-Rabinowitsch transformation. This method was described by Rabinowitsch (12); a proof of it also appears in a paper by Mooney (13). It has been used by Blott and Samuel (14) and doubtless by others. The method merely assumes the existence of a functional relationship between rate of shear and shear stress. The final form of the transformation is

$$R = \frac{1}{\pi r^3} (SQ' + 3Q)$$

where R is the rate of shear, r is the capillary radius, S the shear stress at the wall, Q' is the first derivative of the rate of flow in respect to the shear stress, and Q is the rate of flow. The experimental plots can be converted into curves of rate of flow versus shear stress, and Q' can be found graphically. Curves for rate of shear against shear stress obtained by this method are shown in Figs. 16, 17, 18, and 19.

Difficulties of the Problem

Certain questions have arisen in the course of the work and might well be discussed at this point. A very important one is just what significance should be attached to pressure when one is dealing with a plastic material in flow. Treating the flow-producing forces as pure hydrostatic pressures is open to criticism, and yet an attempt to deal with other forces in the material itself is difficult, both experimentally and analytically. In short, there is the problem of dealing with a combination of forces of a hydrostatic nature and of a strain nature. The present research has sidestepped this problem by dealing with materials which are relatively soft under test conditions and then assuming that the forces are predominantly hydrostatic.

Another difficulty lies in the technique of measuring a difference in pressure across the test capillary. With the Bourdon gages used in this work, the difference in pressure had to be a high percentage of the absolute pressure. This was due to the loss in accuracy when subtracting p_2 from p_1. An attempt was made to calibrate the gages and apply corrections to the readings, but this left much to be desired, for it was found that the gages did not repeat very accurately. The main source of error in the gages is due to the link mechanism which multiplies the exceedingly short tip travel. Considerations of piston type or manganin-coil gages have not led to much encouragement. Piston gages are subject to leakage and friction; these would be important considerations when one tries to measure a small difference in pressure at a high absolute pressure. Furthermore, construction difficulties are severe. Manganin coils would need electrical readings to six significant figures to obtain the desired accuracy on the difference. The difficulty arises in the fact that with either of these systems one is actually measuring two pressures and subtracting, whether that is done mechanically or arithmetically. The most hopeful scheme is a differential-bellows gage, proposed by P. G. Exline of the Gulf Research and Development Company. The use of optical levers on Bourdon tubes, perhaps, offers some possibility of improving their accuracy so that smaller differences in pressure may be used.

There are several reasons for desiring to reduce the difference in pressure. One is that the mathematical analysis of the flow curves requires the assumption that the material does not vary in properties along the length of the capillary. This is obviously far from true if a large pressure drop exists.

Another disadvantage is that a large pressure difference means that a great deal of work has been put into the material. This work must take the form of heat as the oil shears. Although the capillary is in a bath, it is questionable whether a large amount of heat can be dissipated without a substantial temperature gradient. Further experimental work would be desirable to check the temperature inside the capillary.

Time lag in the effect of pressure on the test material is a problem which has been given scant attention in the two-capillary work, other than to eliminate it as a variable by standardizing the time of application of pressure. The shear-stress versus rate-of-shear relationship is dependent upon the time of application of pressure as has been shown, and further work upon this subject might be desirable.

A thixotropic material is one which changes consistency upon being deformed or worked. The material may regain its original consistency after a lapse of time. This subject has been studied by Reynolds, Freundlich (15), McMillen (16), and others. There is reason to suppose that thixotropy exists in solidified oils as well as in greases. One method of investigation of thixotropy in work of this type would be to use test capillaries of different dimensions for a given oil and temperature. If no discrepancy appeared in the rate-of-shear versus shear-stress curves derived from such tests, it would be an indication of freedom from thixotropic behavior. With sufficient data it might be possible to formulate consistency as a function of amount of shear as well as rate of shear. It should be kept in mind that, while there is no provision for working the test samples in this high-pressure apparatus, the tests cannot be regarded as representing unworked oils. There is an undetermined amount of working in the end effects of the test capillary plus the working occurring inside the capillary.

Provision for working the test material thoroughly could be provided most easily in a rotation type of consistometer. This type of instrument has the added advantage of having the test material under a uniform pressure. It is to be hoped that future work on oils under pressures causing apparent solidification will be done on a rotation consistometer. In the meantime, capillary tests will provide a useful preliminary survey.

Conclusions

The problem under investigation is a difficult one, and it has by no means been solved. The difficulties of the work have just been mentioned, and the results must be considered as first approximations. However, interesting results were obtained on the increased effect of pressure due to time of application in regard to solidification of rapeseed and sperm oils, as can be seen by Figs. 5, 6, 7, and 8. The shear characteristics of sperm, lard, and Veedol medium oils at 20 C, as affected by pressure, have been shown by curves of rate of shear versus shear stress in Figs. 17, 18, and 19. The behavior of sperm oil at 0 C has been described by curves of rate of shear versus shear stress and by plots of yield shear stress and mobility against average pressure, Figs. 14, 15, and 16. Non-Newtonian behavior of rapeseed and lard oil at low temperatures has been indicated in Figs. 2 and 3.

To summarize, the consistency of an oil near apparent solidification is dependent upon time of application of pressure as well as upon the amount of pressure and the temperature. The consistency is very likely dependent upon the amount of shear as well.

Acknowledgments

The work was done in the Lubrication Laboratory of the Graduate School of Engineering of Harvard University with funds furnished by the Milton Fund of the University and by the Special Research Committee on Lubrication of this Society. The apparatus was furnished by the A.S.M.E. Mr. Mayo D. Hersey gave generously of his time in acting as consultant, and the authors are especially grateful to him. Mr. A. L. Labastie, a student in Harvard College, helped with the laboratory work. Professors P. W. Bridgman and F. Birch and Mr. L. H. Abbot of the University aided by sharing experience gained in high-pressure research. The Spencer Kellogg Company, the Summerill Tubing

Company, and the National Bureau of Standards donated material or loaned equipment.

BIBLIOGRAPHY

1 "Viscosity of Lubricants Under Pressure," by M. D. Hersey and Henry Shore, *Mechanical Engineering*, vol. 50, 1928, pp. 221–232.

2 "Collected Results on Viscosity of Lubricants Under Pressure, I—Fatty Oils," by M. D. Hersey and R. F. Hopkins, *Journal of Applied Physics*, vol. 8, 1937, pp. 560–566.

3 "Experiments by Robert Kleinschmidt on the Viscosity of Lubricating Oils Under High Hydrostatic Pressure," Trans. A.S.M.E., vol. 50, paper APM-50-4, 1928, pp. 2–5; *Mechanical Engineering*, vol. 50, 1928, pp. 682–683.

4 "Viscosity of Oil Under Pressure," by Yoshio Suge (in Japanese), Bulletin of the Institute for Physical and Chemical Research, Tokyo, vol. 11, 1932, pp. 877–894; vol. 12, 1933, pp. 643–662.

5 Discussion on "Viscosity of Lubricating Oils," by Henry Shore, *Mechanical Engineering*, vol. 50, 1928, pp. 682–683.

6 "Changes in the Viscosity of Liquids With Temperature, Pressure, and Composition," by C. S. Cragoe, Proceedings, World Petroleum Congress, London, vol. II, 1934, pp. 529–541.

7 Discussion on "Teaching Lubrication," by A. E. Norton, *Journal of Engineering Education*, vol. 30 (N.S.), June, 1940, pp. 880–882.

8 "A New Consistometer and Its Application to Greases and to Oils at Low Temperatures," by R. Bulkley and F. G. Bitner, *Journal of Rheology*, vol. 1, no. 3, 1930, pp. 269–282.

9 "High-Pressure Capillary Flow," by M. D. Hersey and G. H. S. Snyder, *Journal of Rheology*, vol. 3, no. 3, 1932, pp. 298–317.

10 "On Plastic Flow Through Capillary Tubes," by E. Buckingham, Proceedings American Society for Testing Materials, vol. 21, 1921, part 1, pp. 1154–1156.

11 "Future Problems of Theoretical Rheology," by M. D. Hersey, *Journal of Rheology*, vol. 3, no. 2, 1932, pp. 196–204.

12 "Über die Viskosität und Elastizität von Solen," by B. Rabinowitsch, *Zeitschrift für physikalische Chemie*, Bd. 145, Abt. A-1, 1929, pp. 1–26.

13 "Explicit Formulas for Slip and Fluidity," by M. Mooney, *Journal of Rheology*, vol. 2, no. 2, 1931, pp. 210–222.

14 "Flow Characteristics of Lime-Base Greases," by J. F. T. Blott and D. L. Samuel, *Industrial and Engineering Chemistry*, vol. 32, no. 1, January, 1940, pp. 68–72.

15 "Thixotropy," by H. Freundlich, Hermann, Paris, 1935.

16 "Thixotropy and Plasticity," by E. L. McMillen, *Journal of Rheology*, vol. 3, nos. 1 and 2, 1932, pp. 75–94, 163–195.

Discussion

L. J. Bradford.[6] Advances in science are made in three stages: (*a*) New phenomena are observed, (*b*) these phenomena are studied to discover and interpret their meaning, and (*c*) the phenomena are usefully applied. The late Prof. Norton and his associates have, in the work described in this paper, accomplished the first of these. The interpretation and application of these data will follow. In the development of these phases, all those interested in the work should participate.

Examination of Figs. 16 to 19, inclusive, indicates that in all cases the curve of rate of shear plotted against shearing stress is substantially a straight line passing through the origin for a pressure of 10,000 psi. It may be concluded that this is also true for all lower pressures. At 14,000 psi this condition ceases. The rate of shear rises more rapidly than does the shearing stress, and the curve is concave. Extrapolation of the curves for this and greater pressures yields an intercept on the shear-stress axis. They are clearly the curves of plastic substances.

Curves for 18,000 psi in Fig. 16, and for 18,000, 23,000, and 27,000 psi in Fig. 17, show another peculiarity. It will be seen that each is composed of two substantially straight lines joined by a curve. It is quite possible that the other curves would show the same characteristic had they covered wider ranges of shear stress. This suggests that the oils investigated pass from Newtonian liquids to plastic solids at some pressure between 10,000 and 14,000 psi.

These plastics are of the Bingham type and have a dual consistency, depending upon the rate of shear to which they are subjected, the two types being connected by a transition region, lying roughly between rates of shear of 10,000 to 15,000 reciprocal sec.

Another fact of considerable interest and importance which has been noted is the relationship of time to the transformation of the oils from Newtonian liquids to Bingham solids. This is of importance because any attempt to make use in bearing design of the elevation of viscosity caused by pressure must be limited to the change possible while the oil is in the load-carrying region. This is usually only a fraction of a second. Quite possibly the pressure effects will not appear at all.

The work described by the authors is obviously incomplete and should certainly be continued. The range of the investigation into the rate of shear versus shear stress should be considerably extended. The effect of time and work should be thoroughly investigated, and it might be found worth while to look into the effect caused by repeated and rapid application of pressure.

The Special Research Committee can perform a valuable service to the science of lubrication by using its influence to further the investigation of the phenomena described by the authors.

R. B. Dow.[7] The authors are to be congratulated as the first to offer quantitative data on flow properties under high-pressure differences in the congealed state. It has been recognized for some time in lubrication practice that "pumpability" at low temperature is a property not described adequately in terms of viscosity of the lubricant alone. This paper indicates a start in the right direction and it is to be hoped that further work will eliminate some of the errors and difficulties which were experienced by the authors.

It is to be pointed out, however, that these experiments give no information about solidification in the thermodynamic sense, and the nature of freezing as understood in the sense of Clapeyron's equation must still remain an open question. It would be desirable to determine freezing of a lubricant by compression by the free-piston method, a method which enables the volume changes to be followed. The writer has plans projected for an experiment of this kind. It is hoped that the sharpness and extent of freezing of a variety of lubricating oils can be studied and the results correlated with their various chemical and physical characteristics.

Regarding the data of the present paper, it would appear that few generalizations can be made since the results show clearly that the history of the pressure treatment is a vital factor, which, from the nature of the conditions, is to be expected, for example, Figs. 7 and 10. The data of Fig. 5 show that a 2-hr application gives uniform results and reproducibility; evidently equilibrium conditions are being approached in this case. However, it is to be noted that the procedure followed does not distinguish between the effect of magnitude of pressure and the effect of time of application of pressure. A pressure of 500 psi, let us say, applied for 10 min, on a sample initially at atmospheric pressure at 0 C would produce quite a different effect from that produced by the same pressure added to an already existing pressure which may have produced partial solidification. If successive increments of pressure are increased according to the methods of the authors, for the same time intervals, it is clear that the state of solidification will be more complete at the higher pressures and this in turn will affect the flow characteristics. It is suggested that sudden pressure relaxations (10 min) during a test be avoided, and

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that it might help if the state of the substance were brought back to initial conditions before applying a new pressure. Since fairly large flow rates have been used in some cases, it would seem as if the present procedure brings in objectionable inertia effects due to variable accelerations of undetermined masses of liquid and partly solidified matter.

P. G. Exline.[8] The high pressures existing between lubricated metal surfaces in many industrial applications fully warrant an investigation of the nature of this paper. The pioneer work reported by the authors must eventually be supplemented by additional work on many materials under a wide variety of conditions before its maximum usefulness can be realized, but the groundwork has been well done.

The difficulties of measuring the pressures accurately and of maintaining steady pressures for a long enough period to secure a good measurement of the flow will undoubtedly be overcome by improvements in apparatus. The analytical difficulties inherent in the capillary method may not be so easily solved. Have the authors considered the use of low temperatures to determine if the behavior obtained at low temperature and atmospheric pressure is the same as at high pressures and higher temperatures?

Figs. 5 and 6 show a complete cessation of flow for rapeseed oil at high pressures. Was any attempt made to determine how long the valve at the end of the capillary would have to be left open before the oil at that end would return to the liquid state and start flowing out?

M. D. Hersey.[9] Some idea of the heat effects possible may be obtained from the mean temperature rise calculated[10] for radial conduction under the limiting condition of thermal equilibrium

$$T_m = \frac{5}{384}\left(\frac{G^2r^4}{\mu k}\right)$$

where G denotes the gradient $(p_1 - p_2)/L$ while μ is the viscosity of the oil, assumed uniform, and k its thermal conductivity, r and L being the radius and length of the capillary.

For the mineral oil of Fig. 19 of the paper, the viscosity at 20,000 psi and 20 C under a shear stress of 2.5 psi is equal to 2.5/5000 or $5(10)^{-4}$ lb sec per sq in. The conductivity[11] at this temperature, disregarding any slight increase due to pressure, is about 0.029 lb per sec deg C. Substituting these values, together with the capillary dimensions from Table 2 of the paper, gives for a mean pressure difference of 16,000 psi (Fig. 13) T_m = 1.6 C.

Would it be possible to summarize the experimental results obtained for the committee during the summer of 1940, including check observations with a smaller-diameter capillary?

R. V. Kleinschmidt.[12] It is unfortunate that the excellent work reported in this paper should be interrupted by the untimely death of Professor Norton. The importance of such research is perhaps most greatly appreciated by those of us who have had an opportunity to work in this field.

Some 15 or 20 years ago, the work of Mr. Hersey and others indicated that the peculiar lubricating properties of oils were in some way related to their tendency to increase markedly in viscosity or even to solidify under pressure. At the same time, it became obvious that if they do solidify, their lubricating properties must be largely dependent upon the properties of the solids formed. Certainly, a material which solidified into hard angular crystals would be a poor lubricant, whereas, one which formed a more or less plastic solid might be better. Finally the solid might take the form of smooth plates like graphite or mica which would conceivably make an excellent lubricant. It is thus obvious that the properties of these pressure-solidified oils are of fundamental importance. To determine such properties is the object of the present research.

Without wishing in any way to detract from the value of the work performed to the present time, the writer would like to suggest a direction in which future effort should proceed. While it is natural that preliminary work should be done on commerical lubricants, it must be remembered that such materials are not only extremely complex mixtures but that they are continually varying in actual composition, as various petroleum pools are tapped. Therefore, it would seem to be essential that a fundamental study should include work on some pure substances, and on relatively simple mixtures of substances normally found in lubricants.

Furthermore, it is important to consider not merely "plasticity" as such, but the possibility of surface slippage and planes of slip within the body of the solidified lubricant.

Finally, it is important that the experimental methods be simplified in so far as possible by determining at the outset any general relations between, for example, solidification due to pressure and ordinary freezing due to low temperature. It is, of course, by no means certain that any such relationship exists since, in a bearing, the pressure conditions in the oil are probably far from isotropic. The simplification of laboratory work which would result makes a search for such a relationship worth undertaking.

The writer feels that the work reported in this paper should be continued and extended into a far-reaching basic study of lubricants and their behavior. While the Society cannot sponsor the study of any considerable number of commercial lubricants, it can and should develop the fundamental laws and the techniques required. These will then be quickly taken up by the commercial laboratories. In view of the enormous value of machinery which must be protected by lubricants and the vast amount of power which is wasted in friction, any slight improvement in lubrication would pay high dividends to industry on money invested in such research.

C. M. Larson.[13] Referring to the Veedol medium which is a Pennsylvania 100 V.I. SAE 30 motor oil such an oil would compress 17 per cent of its volume under 100,000 psi, whereas, a Gulf Coast oil, zero V.I. SAE 30 would lose 15 per cent under the same pressure. Yet the 100 V.I. SAE 30 oil under 12,000 psi has not as high Saybolt Universal viscosity at 100 degrees F or 210 degrees F as the zero V.I. SAE 30 oil under 6000-psi pressure. Thus, the 100 V.I. oil compresses more readily, yet its increase in viscosity under pressure is less than the zero V.I. oil. The heavier the oil based on atmospheric-pressure viscosities, the higher the rate of compressibility.

When it is considered that the pressure per square inch of aircraft-engine bearings at take-off varies for different engines from 2500 to 3500 psi, it is possible to have viscosity-pressure-effect increases from 6 to 12 per cent at the operating temperature but, when a plane is in a power dive and bearing pressures of 8000 psi are encountered, viscosity-pressure build-up of 25 per cent or higher is possible in the oil film. With hypoid-gear-tooth pressures of 100,000 lb, the viscosity-pressure build-up can be easily

[8] Engineer, Lubricating Research and Instrument Development, Gulf Research & Development Company, Pittsburgh, Pa. Mem. A.S.M.E.

[9] Research Director, Morgan Construction Company, Worcester, Mass. Fellow A.S.M.E.

[10] "Note on Heat Effects in Capillary Flow," by M. D. Hersey, *Physics*, vol. 7, 1936, pp. 403–407.

[11] "Thermal Properties of Petroleum Products," by C. S. Cragoe, U. S. Bureau of Standards, M97, 1929, pp. 24–25.

[12] Stoneham, Mass. Mem. A.S.M.E.

[13] Chief Consulting Engineer, Sinclair Refining Company, New York, N. Y. Mem. A.S.M.E.

in the order of 10 times the original atmospheric viscosity at the temperature of operation. Many substances which are plastic and are considered lubricants at atmospheric pressures are abrasives, harder than steel. Roller bearings build up pressure of higher than 100,000 lb ahead of the rollers.

Robert Matteson.[14] In the introduction to the paper, the authors attribute Newtonian behavior to "most lubricating oils at ordinary pressures and temperatures," implying that for these materials the viscosity is independent of shear rate. But Elwell[15] has shown theoretically that all liquids should exhibit a decrease in viscosity with shear rate, and Kyropulous[16] has proved experimentally that "natural oils," including lubricating oils, begin to exhibit a measurable decrease in viscosity at shear gradients as low as 3000 s^{-1}. The fact that viscosity of oils does decrease at such low shear rates undoubtedly introduces a complicating variable into the work of the present authors who report measurements well above 35,000 s^{-1} and are primarily engaged in studying the effect of pressure on viscosity.

A question arises concerning the method used in calculating flow and shear rates. If compressibility of the oils is neglected, the flow rates will be high by approximately 5 per cent at 15,000-psi and 10 per cent at 30,000-psi average pressure for lubricating oils[17] and somewhat less for the fatty oils. Shear rates will also be affected to the same extent. Where large pressure drops occurred between inlet and discharge sections of the capillary tube, the effect of compressibility varies all along the tube and further complicates the problem.

With regard to the difficulties encountered in measuring pressure differences, these could be overcome by using electrical pickups now available which involve no moving parts nor offer geometrical obstacles to alter the flow pattern in the tube, as is the case with the connections required for pressure gages of the Bourdon type.

The wisdom of concentrating attention upon oils such as lard, rapeseed, and whale oil is open to question in view of the relative industrial importance as lubricants of these oils, as compared with the products obtained from petroleum. It is perhaps felt that, as explained in the paper, these fatty oils were known to undergo apparent solidification under pressure and, hence, offered the best working materials. This may be true but the Veedol medium also exhibited the property of solidification in the range of pressure studied by the authors. Futhermore, Suge's data[18] indicate that other petroleum oils will behave similarly at high pressure. It is for oils of this class that data are most needed.

It is doubtful whether the results obtained with fatty oils can be used with confidence in predicting the behavior of petroleum fractions. One serious difference in the two types of oils is that the fatty oils, being polar in composition, exhibit orientation phenomena which are less pronounced in the case of the mineral oils. A second difference is in the degree to which the "free volume" between the molecules must be changed in going from atmospheric pressure to 30,000 psi, where apparent solidification sets in with fatty oils, and the contraction in "free volume" of a petroleum fraction where the pressure may be of the order of 40,000 or 50,000 psi.

These remarks are not in the nature of negative criticism or refutation of the general conclusions reached by the authors. We realize that the experiments described are of a preliminary nature and it is our hope that the research may be continued with greater emphasis placed upon the investigation of the behavior of petroleum fractions under high pressure.

[14] Research Engineer—Physicist, Standard Oil Company of California, San Francisco, Calif.

[15] "The Reaction-Rate Theory of Viscosity and Some of Its Applications," by R. H. Elwell, *Journal of Applied Physics*, vol. 9, 1938, pp. 252–269.

[16] "Die Zähigkeit von Schmierölen bei hohen Geschwindigkeitsgefällen in der Schmierschicht," by S. Kyropoulos, *Forschung auf dem Gebiete des Ingenieurwesens*, vol. 3, 1932, pp. 287–296.

[17] "Compressibility and Velocity of Pressure Waves in Petroleum Oils," by R. Matteson, *Journal of Applied Physics*, vol. 9, 1938, pp. 44–49.

[18] "Viscosity of Oil Under Pressure," by Y. Suge, Bulletin, Institute of Physical and Chemical Research, vol. 11, Tokyo, 1932, pp. 877–894; "Influence of Temperature and Pressure on the Viscosity of Oils," by Y. Suge, vol. 12, 1933, pp. 643–662.

M. Mooney.[19] This paper on the rheology of lubricants at high pressure represents a preliminary survey in an important and new field of investigation. The work has been well done and well presented. It is particularly encouraging to one interested in the science of rheology as such to see the differential method of analyzing capillary-flow data coming into use. This method is definitely more powerful than the older method of analysis by integration, as is demonstrated in the present paper.

The data in Fig. 7, and also the data for 30-min application of pressure in Fig. 8, suggest that slippage of the solid material against the surface of the capillary may be taking place at the higher pressures. It is to be hoped that, when these investigations are carried further, measurements of slippage will be included in the program. Such measurements could be obtained with the apparatus employing two capillaries, provided that two or more short capillaries of different radii are used. A method of analyzing such data for slippage has been described in a paper referred to by the authors. In view of Figs. 7 and 8, the functional relationship of rate of shear and shearing stress, as plotted in Figs. 16 to 19, will require verification.

The authors appear to doubt the validity of the customary analysis of stresses in a capillary tube when the material is plastic in its behavior. On this point, it is possible to reassure the authors and state that, so long as the flow is lamellar and parallel to the axis of the capillary, the usual calculation of shearing stress from pressure gradient is valid. In detail, the situation is as follows: Let r, θ, and z be a set of coordinates, and represent by a subscript a plane normal to the corresponding direction. The pressures normal to the coordinate lines are equal, or

$$P_r = P_\theta = P_z$$

At any point, P_θ is one of the principal stresses. The other two principal stresses lie in the r, z plane through the point considered and are oriented at 45 deg with respect to the r or z axis. The maximum and minimum pressures differ from the mean pressure by an amount equal to the shearing stress, which is $\frac{r}{2}\frac{dP_z}{dz}$. Thus

$$P_{\max} = P_z + \frac{r}{2}\left|\frac{dP_z}{dz}\right|$$

$$P_{\min} = P_z - \frac{r}{2}\left|\frac{dP_z}{dz}\right|$$

M. Muskat[20] and F. Morgan.[20] The authors have already given considerable attention to possible criticisms of their paper. Furthermore they have carefully outlined the experimental and analytical difficulties which they have encountered.

A particularly troublesome element in the experiments, pointed out by the authors, is that relating to the determination of the pressures in the system with such accuracy that flow experiments

[19] General Development Division, United States Rubber Company, Passaic, N. J.

[20] Gulf Research & Development Company, Pittsburgh, Pa.

could be carried out, with small differential pressures which are known, with reasonably high precision. The variation in the nature of the fluid along the length of the capillary, when it is subjected to a large pressure differential, undoubtedly, greatly complicates the interpretation of the results. However, it may be pointed out that, if a method should be found for determining the high pressures with good accuracy, such as the differential bellows gage proposed by Exline, much useful information may still be derived by repeating the original experiments under high pressure drops, provided the pressure distribution were measured along the length of the capillary. Then, the effect of the amount of working on the fluid, as well as the variation of the viscosity with the pressure, could be followed in a continuous manner by observing the sequence of pressure-drop increments along the length of the capillary. Such a procedure would be equivalent to series of measurements over short capillaries with different average absolute pressures. It would have the advantage over the latter, however, in that a close control over the previous state and history of the fluid would be automatically provided by the pressure drop in the segment of the capillary immediately preceding the particular segment being studied.

The obvious difficulty of using a long capillary tube in high-precision experiments, which arises from the variability of and uncertainty in the magnitude of the cross section, can be readily avoided by calibration runs at low pressures with a liquid known to be Newtonian. The pressure distribution along the capillary in such experiments will give a direct measure of the local average tube radius. In fact, it will give at once the variation of the fourth power of this radius and thus avoid magnification of the errors when the capillary radius itself is raised to the fourth power. Moreover, if this idea is generalized, one is led to the proposal that the capillary be deliberately made of several sections of different radii, the effective values of which could then also be determined by calibration tests with a Newtonian liquid. In this way the effects of different capillary dimensions, as well as of various amounts of working, could be investigated in a single experiment.

It is realized, of course, that these comments do not provide a solution to the basic problem of the accurate determinations of the pressures. Rather they relate only to the further development of the experimental program, once the difficulties of technique have been satisfactorily solved.

C. H. Schlesman[21] and R. Bulkley.[21] In general, we accept the performance of lubricants on bearing surfaces so casually, because of their exceedingly high percentage of satisfactory performance, that we are inclined to overlook the importance of lubrication to industry and our own lack of knowledge on the subject.

It is pointed out in the paper that there are three types of lubricants, i.e., those which may be considered as Newtonian liquids, those which are non-Newtonian liquids, and those known as plastic solids. It is the purpose of this discussion to call attention to an equally important type of lubricant, the classification of which, in the absence of experimental data, must remain unknown for the present.

The behavior of lubricants which are Newtonian liquids appears to be well understood and the design of bearings operating in this region appears to rest upon adequate experimental foundations, thanks to the careful researches fostered by this Society and the work of some of our leading rheologists.

In the field of lubricants of the plastic class, the subject is much more controversial and it is suspected that rule of thumb and practical experience are necessary in designing in this field. It does not appear to be good practice to operate loaded bearings for prolonged periods entirely within this region of lubrication. On the other hand, bronze worm gears operating against steel worms in which the load passes from tooth to tooth can be operated successfully in this region of lubrication.

The authors present experimental evidence to show that plastic phenomena occur with certain types of lubricants under high pressures. Recent work in the field of X-ray analysis indicates that, in addition to this plasticity, orientation of the molecules also occurs which, perhaps, in part accounts for the flow properties under high pressure, but which, in addition, imparts certain lubricating properties to a fluid. X-ray work with crystal diffraction equipment supports the fact that long hydrocarbon chains can form parallel bundles and that polar materials, of which sperm and lard oil are examples, show definite orientation under suitable conditions.

The foregoing groups of lubricants are exceedingly important ones. Another group has become of outstanding importance in recent years. Viewed from the standpoint of physical mechanics, the groups of lubricants mentioned in the paper are representative of materials in which the mechanical or electrical bonds exert large forces within the molecule and weak forces between lubricant molecules and the material of bearings and journals. In lubricants of the type considered here, the presence of powerful chemical bonds or the development of such bonds in service leads to the formation of rather stable molecular compounds at the interface between the lubricant and the bearing or journal.

Successful utilization of the heavily loaded small-size hypoid gear in the rear axles of modern passenger cars capable of developing as much as 150 hp depends upon this principle of lubrication. In the presence of a Newtonian liquid, under the extremely high pressure and temperature load imposed upon individual teeth in such a gear, small portions of the pinion steel actually weld into the face of the wheel with exceedingly rapid destruction of both gear members. When an active lubricant is substituted for the inert fluids so commonly employed in other forms of lubrication, a thin film is formed upon the gear surfaces by interaction of the lubricant and the steel of the gears. This film, being held to the steel with powerful forces and yet showing a far lower shear value than steel itself, serves to act as a cushion between the gear teeth, reducing the friction and preventing the actual welding which otherwise occurs. This is, perhaps, the most extreme example of the active type of lubricant.

An equally important application is one requiring a milder acting lubricant. In modern aircraft engines, for example, firing pressures often exceed 1000 psi. Through rocking of the pistons or the use of tapered piston rings, only the sharp edge of the piston ring may rest against the cylinder at a given instant. Actual embedding of the cast iron of the ring into the steel of the cylinder wall occurs even with mirror-finished cylinders. Under these circumstances, certain lubricants have been found capable of forming on the metal surfaces weakly attached layers, held there by physical bonds or by chemical bonds, which themselves serve as lubricants or which act as a cushion to improve the action of fluid lubricants.

It should be pointed out that the strides of industry in this country are so rapid that new types of lubrication are taking their place in industry while we are still trying to explain the behavior of those which have been in use over a century. The only hope, then, of keeping abreast of industry lies in basing future research upon the new discoveries as they emerge from the laboratory, and in bridging the gap between the past and present as rapidly as funds will permit.

An incidental item of great interest is the observation, reported by the authors, that rapeseed oil, solidified by high pressures, remained solid when the pressure was removed by opening the reservoir. It would normally be expected that the solid form would revert to liquid at once when the pressure was lowered.

[21] Socony-Vacuum Oil Company, Inc., Paulsboro, N. J.

If this observation can be confirmed it will constitute an extreme case of hysteresis, and it may have an important bearing on present theories of the persistence of strain in solid materials. An alternative explanation might be a very rapid polymerization.

P. R. Vogt.[22] The presentation of this excellent paper almost a year after Prof. Norton's death is a tribute both to the thoroughness with which he prepared the foundation for this work and to his wise choice of assistants, which left men who are able to continue without the benefit of his direct guidance.

A more complete understanding of the change of oil viscosity with pressure will be of importance to everyone concerned with lubrication problems. In the automotive industry connecting-rod-bearing loads as high as 2000 psi of projected area are common practice, and airplane engines sometimes use as high as 10,000 psi. Assuming roughly that the maximum pressure in the oil wedge is about 4 times the load per square inch of projected area,[23] the actual oil pressures in the foregoing applications are 8000 and 40,000 psi, well within the range of greatly increased viscosity or "apparent solidification" reported in this paper. Gear drives, especially rear axles, sometimes operate at tooth-contact pressures of 300,000 psi or more; and although part of the load is sustained by direct metal-to-metal contact, considerable "apparent solidification" of the lubricant is bound to occur. This will certainly have an appreciable effect on the efficiency and life of the drive.

As yet the effects are unknown; and up to now, engineers do not generally realize that the phenomenon even exists; but eventually it is entirely possible that the change in viscosity due to pressure may assume as prominent a place as that now given to the change due to temperature.

To this end it is desirable that the high-pressure investigations be continued along the lines suggested in the paper. Further tests should be made on a wide variety of commercial lubricants such as engine oils, transmission oils, and extreme-pressure lubricants in an attempt to find, if possible, a correlation between high-pressure viscosity behavior and known results obtained with the particular lubricant in actual service. In other words, does a universally satisfactory oil have different high-pressure viscosity characteristics from a poor oil? Of course, this will depend upon the type of service for which the oil is good or poor.

Tests should be conducted with the variables of temperature, time of pressure application, and amount of previous working held as nearly as possible to actual service conditions. In particular, the work done on SAE 30 engine oil should be repeated at normal engine-oil temperature of about 150 C. Therefore it is sincerely to be hoped that Knott and Muenger will continue this valuable work.

Authors' Closure

Messrs. Larson and Vogt give examples showing that a better knowledge of pressure effects on lubricants is of importance in practical problems of machine operation. However, Prof. Bradford correctly emphasizes the need for interpretation of the data given in this paper before they can be applied to such problems. The first step might be the separation of extraneous effects from the flow-pressure curves, such as are given in Figs. 10 to 13 of the paper. Temperature rise in the test capillary; compressibility of the oil, as mentioned by Messrs. Larson and Matteson; the possibility of slippage, as mentioned by Messrs. Kleinschmidt and Mooney, lead one to suspect that the curves of Figs. 16 to 19 of the paper have too great a curvature. It is stated in the paper that these curves should be regarded as first approximations, and they should be verified as Mr. Mooney suggests. There is need for further distinction between the effects of pressure due to magnitude and those due to continued application as cited by Prof. Dow. The study of hysteresis deserves attention, and it would be desirable to shorten the time of application of pressure so as to approach service conditions. In reply to Mr. Exline's question, we regret that we have no data concerning the time required for a solidified oil to return to its liquid state.

It is to be hoped that refinements in technique will eliminate some of the uncertainties mentioned in the paper. Suggestions of Messrs. Matteson, Mooney, Morgan, and Muskat give some indication of possible steps. The determination of several pressures along the length of the test capillary, as suggested by Messrs. Muskat and Morgan, would indeed give useful information on the effect of working the oil but, with the present methods of connecting lengths of capillaries and tapping for pressure determinations, the flow could not be considered the same as that for a single capillary tube.

The choice of fluids for further experiments would seem to depend upon whether the emphasis is to be placed upon explanation of the phenomena being studied or upon the accumulation of data for actual practical applications. The use of pure substances and simple mixtures suggested by Mr. Kleinschmidt rather than complex oils should be of great interest. The solidification of pure substances should be more sharply defined, and their shear behavior might be much simpler. However, it is possible that the selective solidification of components of an oil give it shearing properties which must be studied by use of the oil itself, rather than one or two of its components.

The establishment of an analogy between the effect of low temperatures and high pressures would simplify the experimental work to a marked extent. As was mentioned in the foreword, Shore has called attention to an empirical linear relationship between solidifying pressures and temperatures, the slope of the lines being constant for several oils. That such a relationship can be established for studying shear behavior over wide ranges of shear is not clear. Further data seem to be necessary before this analogy, mentioned by Messrs. Exline and Kleinschmidt, can be established.

Mr. Matteson questions attributing Newtonian behavior to lubricating oils under normal conditions, and he suggests a complicating variable is thereby introduced in the study of shear behavior under pressure. Since this investigation specifically presupposed non-Newtonian behavior in the regions to be studied and merely set forth rate-of-shear versus shear-stress curves at various average pressures, it is hard to see how non-Newtonian behavior at atmospheric pressure would complicate the work. Furthermore Figs. 16, 17, and 19 show that sperm oil at 0 and 20 C under 9500 psi pressure and Veedol medium at 20 C and 10,000 psi are Newtonian. It is safe to assume that this also indicates Newtonian behavior for these oils at lower pressures. It would not, of course, be safe to predict such behavior at higher rates of shear than were studied. There are many confirmations that lubricating oils behave as Newtonian liquids under normal conditions. Bradford and Villforth[24] have just recently presented such data on five oils for rates of shear up to 320,000 s^{-1}. Any experimental verification of the hydrodynamic theory of lubrication may be considered a verification of Newtonian behavior of the lubricant under the conditions of the test, for the hydrodynamic theory is based upon Newton's law of viscosity. Such verifications are for the so-called "thick

[22] Detroit, Mich. Mem. A.S.M.E.

[23] "Pressure Distribution in Oil Films of Journal Bearings," by S. A. McKee and T. R. McKee, Trans. A.S.M.E., vol. 54, 1932, paper RP-54-8, pp. 149–161.

[24] "Relationship of Viscosity to Rate of Shear," by L. J. Bradford and F. J. Villforth, Jr., Trans. A.S.M.E., vol. 63, 1941, pp. 359–362.

films," and the dimensions of the test capillaries used in this work would correspond to thick-film conditions in bearings. The effect of orientation of molecules is thought to be pronounced only in the case of thin films, but it is possible that orientation as well as plasticity affects the flow curves as Messrs. Schlesman and Bulkley point out.

Further work was carried on during the summer of 1940 using two capillaries in series. The principal results are given in the form of rate-of-shear versus shear-stress curves for Veedol medium at 0 C in Fig. 20 of this closure, and for castor oil at

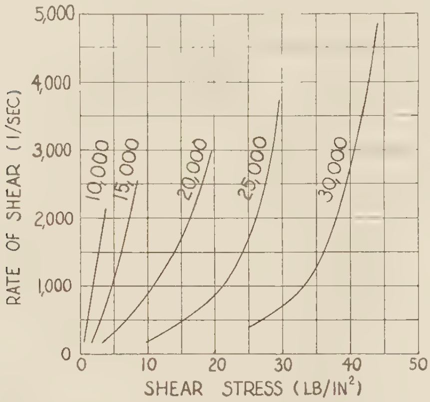

Fig. 20 Shear Characteristics of Partially Worked Veedol Medium at 0 C

(Average pressures indicated along curves, in psi.)

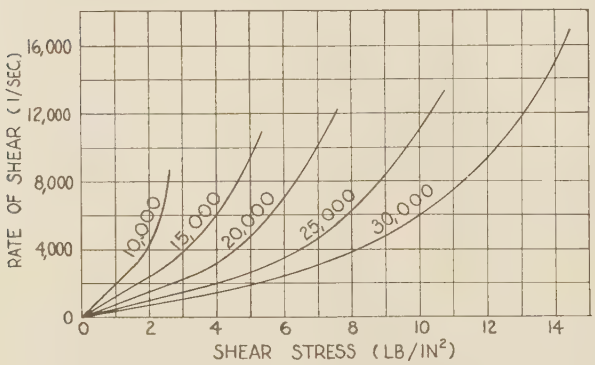

Fig. 21 Shear Characteristics of Partially Worked Castor Oil at 20 C

(Average pressures indicated along curves, in psi.)

20 C in Fig. 21. The test capillary used for the Veedol medium had an internal diameter of 0.0456 cm and a length of 6.08 cm; the test capillary used for the castor oil had an internal diameter of 0.0456 cm and a length of 16.1 cm. These curves merely extend the data given in the paper and are subject to the limitations hitherto discussed.

Tests were made upon Veedol medium at 40 C, using pressure differences appreciably smaller than in the previous work. The observations were erratic due to insufficient accuracy in determining the pressure difference. For example, on the 10,000-psi average-pressure curve, several negative pressure differences were recorded even after using a calibration curve for the Bourdon tubes. The curves, therefore, were ill defined, and the data are not given.[25] These observations emphasized the limitations of the present apparatus.

[25] A complete record of the data obtained, during the summer of 1940, has been filed with the Special Research Committee on Lubrication of The American Society of Mechanical Engineers.

Mr. Hersey asks for a statement concerning check observations, using a smaller-diameter test capillary. An attempt was made toward the end of the summer to check the flow curves for sperm oil at 20 C, Fig. 11, since it was felt that these curves were the best defined. For this work a test capillary was chosen with an internal diameter approximately 0.6 that previously used, the length being such as to give approximately the same flow for a given pressure difference as occurred in the previous work. Such a capillary made the taking of observations easier, since it was possible to predict the value of p_1 necessary to give the desired average pressure for a given resistance capillary. The lowest measurable shear stresses were much higher than those of the previous work, and the observations were, therefore, not conclusive. Mr. Mooney mentions a method of determining the existence of slip described in bibliography reference (13), based upon flow observations from differently dimensioned test capillaries when equal shear stresses are used. The method assumes that the material studied does not have thixotropic behavior, but it is interesting to note that, if data from several capillaries coincide when plotted as $Q/\pi r^3$ against S, freedom from both slip and thixotropy is indicated.

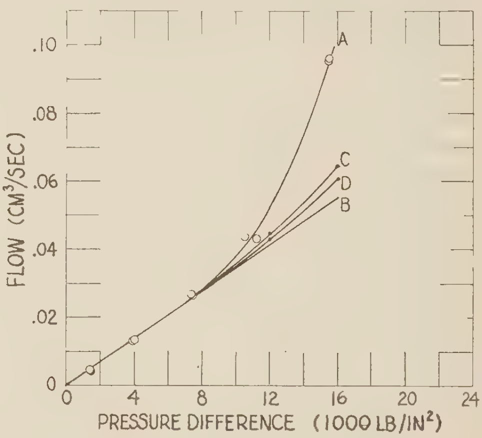

Fig. 22 Flow Curves for Castor Oil at 20 C Under 25,000 Psi Average Pressure

An investigation of the temperature rise in the oil was made, using an iron-alumel thermocouple in place of p_2 of Fig. 9 of the paper. Readings so obtained were ambiguous in view of the very small diameters of the test capillaries, but they indicated that the temperature rises were not excessive. The thermocouple junction was placed in the bore of a connecting block, having a cross-sectional area approximately 12 times that of the test capillary. The bore had a volume of approximately 0.05 cm³ while an average-flow sample might contain 0.5 cm³. In other words, approximately 10 times the volume of the bore was swept out during the taking of a test. Neglecting the conduction of heat by the thermocouple, some sort of mean exit temperature was measured. The limit of sensitivity of the thermocouple was $^1/_3$ C, and the highest observed temperature rise was $3^1/_3$ C, while the majority of the tests showed no perceptible temperature rise.

It is instructive to recall the test procedure when discussing temperature variations. A given pressure p_0 was applied for 10 min while the outlet valve remained closed. Upon opening the valve, p_1 was kept at p_0 while p_2 adjusted itself to flow conditions. When p_2 became steady, a flow sample was taken, the whole procedure being limited to a relatively short time by the capacity of the intensifier. When the thermocouple was used, the opening of the valve was followed by a sudden tem-

perature drop which slowly diminished, followed by a slowly rising positive temperature increment (referred to bath temperature). This clearly indicated release of energy of compression and the behavior also indicated that thermal equilibrium was not reached in the flow tests. The maximum observed value of this temperature drop was 7 C. Therefore it is felt that the mean-temperature rise, as calculated from thermal equilibrium, is substantially higher than that which existed in the flow tests. The amount of heat carried away in the oil stream is neglected in the equation mentioned by Mr. Hersey, but this is somewhat compensated by the assumption that the internal walls of the test capillary are at bath temperature.

For a numerical example of temperature effects, the 25,000-psi average-pressure flow curve for castor oil at 20 C was chosen. The experimental curve is marked A in Fig. 22. If castor oil had Newtonian behavior under the test conditions, and assuming that the viscosity in the capillary was everywhere equal to the viscosity at the mean pressure of 25,000 psi, the flow curve would be linear, as suggested by curve B, when temperature effects are negligible. Then, using thermal conductivity k equal to 0.039 lb per sec deg C, and viscosity μ equal to 2×10^{-8} lb sec per sq in. as obtained from Fig. 21, the mean-temperature rises, as calculated from Mr. Hersey's equation, become 1.7, 3.9, and 6.9 C for pressure differences of 8000, 12,000, and 16,000 psi, respectively. The change of viscosity due to these temperature rises was calculated from data given in chart J, Fig. 1 of bibliography reference (2), and a correction was applied to Poiseuille's law so as to give curve C of Fig. 22. The curve might be considered to show the maximum deviation from a linear graph which could be attributed to temperature rise. The authors are well aware that this procedure is open to severe criticism because it tacitly assumes superposition for many effects, but it seems justified for a first analysis.

It is also possible to investigate the assumption that the viscosity of the oil may be specified by the viscosity at the mean pressure. Writing Poiseuille's law

$$Q = \frac{\pi}{8} \frac{r^4}{\mu} \frac{dp}{dL}$$

and expressing the viscosity as

$$\mu = \mu_0 \, 10^{cp}$$

where μ_0 is the viscosity at atmospheric pressure and test temperature, and c is a constant, an expression may be obtained for Q in terms of p_1 and p_2. The form of the equation for μ is justified by experimental results given in reference (2) and μ_0 and c were computed from that source. The resulting curve for castor oil at 20 C and 25,000 psi is shown as graph D in Fig. 22. This graph, then, represents a flow curve for an idealized case when the oil is Newtonian and has no temperature rise, but where it does have a viscosity variation with pressure such as has been actually observed. This curve may give some indication of the errors arising from the variation of properties along the length of the test capillary. The errors arising from the variation of properties of the material along the length of the test capillary for oils which have undergone solidification may reasonably be expected to be more pronounced.

The authors are grateful to Dr. C. H. Schlesman of the Socony-Vacuum Oil Company for the loan of equipment which was used during the summer of 1940, and to others who cooperated with the project. Prof. J. P. Den Hartog of the Graduate School of Engineering, Harvard University, served as adviser to the project, and Mr. G. A. Sullivan assisted with the laboratory work during this period.

A New Degasifying Steam Condenser for Use in Conductivity Determinations

By F. G. STRAUB[1] and E. E. NELSON,[2] URBANA, ILL.

This paper describes a degasifying steam condenser which will furnish a continuous sample of either steam or condensate which is free from dissolved gases but which contains the dissolved solids which were present in the original sample. Seven of these condensers have been built and installed in different power plants varying in pressure from 150 to 1250 psi. They have reduced the carbon dioxide from 20 ppm and the ammonia from 3.5 ppm to as low as 0.01 ppm for both gases. The unit is automatic requiring practically no attention once it is installed. The sample of degasified steam or condensate may be passed through a conventional conductivity cell and a continuous record kept if desirable. Data collected from the various plants are given as well as data collected in laboratory tests.

VARIOUS methods have been used in the steam power plant for determining the amount of total dissolved solids in the steam or in the condensate from the turbines. If an accurate method is available it furnishes the operator with a yardstick with which he may measure the amount of carry-over from the boiler, as well as the amount of condenser leakage. Since such a method involves the determination of total solids as low as 1 ppm, the calorimeter method does not have sufficient accuracy and it cannot be applied to the condensate. Weighing the solids after evaporation of the water gives sufficient accuracy under proper control but it is limited to special tests and cannot be used as a routine procedure to be run by the operators.

The so-called conductivity method has received much consideration and is being used in many power plants. In this method, a sample of the steam is condensed and passed through a cell fitted with proper electrodes and the resistance of the water to the flow of an electrical current is measured. This resistance varies with the amount of the dissolved solids in the sample. Recording instruments are available which record either the resistance or its reciprocal, the conductance. Thus, a continuous record is available. If two cells are used, one on the steam, and the other on the condensate from the turbine, and the resistance or conductance recorded, the difference indicates condenser leakage. Thus a record is available as to any change in solids in the steam, caused by variation in boiler performance, as well as any condenser leakage. Since the time lag is of very short duration, boiler tests may be conducted to determine the conditions most favorable for low carry-over or low total dissolved solids in the steam.

Unfortunately, some dissolved gases have a marked effect on the conductance of water, consequently, the conductance fails to give a record of the dissolved solids. Gases such as ammonia, carbon dioxide, and hydrogen sulphide, interfere with the use of this method of determining total solids. With a high-quality steam, having less than 1 ppm of dissolved solids, the specific conductance might be in the range of 0.5 to 1.5 micromhos or the specific resistance between 2,000,000 and 670,000 ohms. It has been found that, in many steam samples, 1 ppm of dissolved solids corresponds to a specific conductance of 1.5 to 2.0 micromhos. Ammonia when present in the water will vary in its effect on the conductance, depending upon the form in which it occurs. However, it has been assumed that 1 ppm of ammonia nitrogen is equivalent to 9 micromhos. Thus the effect of ammonia nitrogen is about 4.5 to 6 times that of the average solids occurring in the steam. Free carbon dioxide has a value of 0.6 micromho for 1 ppm or one half that of the average of the solids in steam.

Utilizing the Conductivity Method

One way of making use of the conductivity method has been to apply corrections to the observed values for the dissolved gases (1).[3] However, this necessitates analysis of the steam or condensate samples for these gases. When ammonia and carbon dioxide occur together, it is difficult to determine the true carbon-dioxide value. When the amount of the gases varies, it is necessary to analyze quite frequently. The great objection to the application of these corrections is that often the corrections are many times the amount due to the dissolved solids. Thus one sample of steam tested had a specific conductance of 14.6 micromhos with 2.3 ppm nitrogen ammonia. If the value already noted was used in correcting for the ammonia, the corrected value would be 14.6 — 20.5 or —5.9 micromhos, an impossible value. Here also, due to the ammonia present, the water was alkaline to phenolphthalein making the regular test for carbon dioxide of no value.

Much work has been done toward removing the gases from the steam sample. The first degasifying apparatus was developed by J. K. Rummel (2). This made use of the principle of reboiling the condensed steam. This method proved effective for reducing the free carbon dioxide to a low amount. However, the apparatus did not remove an appreciable amount of the ammonia. The equipment was of a nature to be used for special tests and could not be used for continuous operation along with conductivity-recording equipment. This was due to the attention necessary for controlling rates of flow of steam and condensing waters.

Powell, Bacon, McChesney, and Henry (3) developed an apparatus using the principle of condensing the steam in a vacuum to remove the soluble gases. This method was suitable for use with recording equipment and required very little attention. The carbon dioxide was removed, but the ammonia had to be determined and corrected for. This limited the use of their apparatus to steam having low ammonia.

Developing a Degasifying Condenser

Since neither of these degasifying units could be used on condensate samples, the present study was undertaken to devise a degasifying condenser which would reduce the soluble gases, including ammonia, to negligible amounts. This condenser should

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

be able to remove the gases from condensate as well as from steam and it should be practically automatic so that it could be used in connection with recording equipment.

The degasifying apparatus which was developed is shown in Fig. 1. It makes use of the principle of boiling a sample of the condensed steam from which the gases have been removed to furnish a gas-free steam which in turn removes the gases from the condensed steam. In order to obtain efficient removal of the gases, a scrubbing tower or stripping column is used in connection with a vent condenser, so as to allow venting of the gases after being removed from the condensed steam.

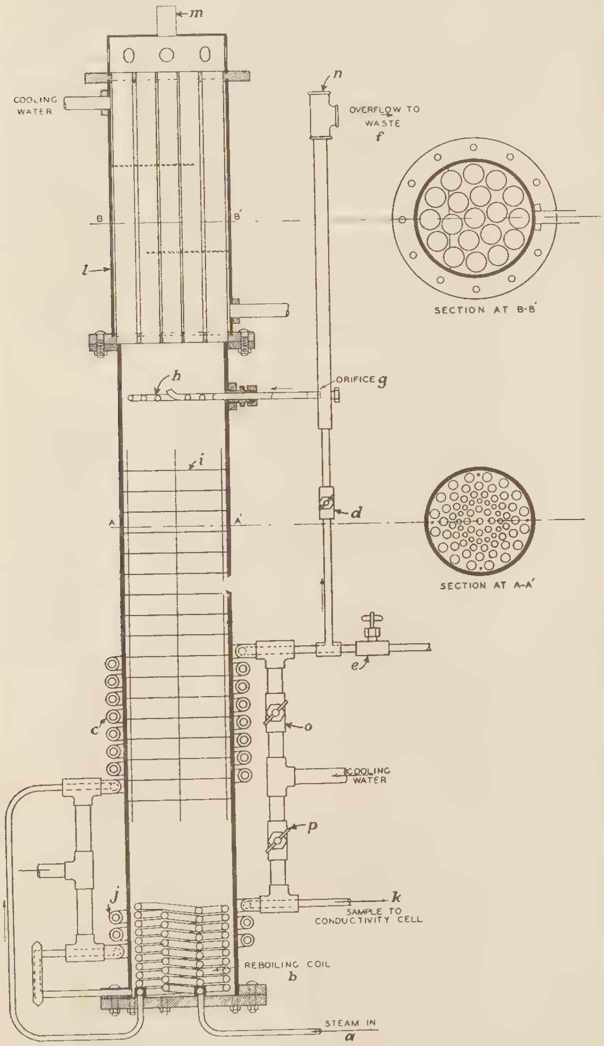

Fig. 1 Degasifying Steam Condenser Developed at University of Illinois

The degasifying condenser may be used for sampling steam or condensate. When used for steam sampling, it works as follows: The steam to be sampled is throttled to allow about 60 to 70 lb of steam per hr to flow to the unit. It is preferable to use an orifice or a length of small-bore tubing, however, a small valve might be used. The steam enters the unit through metal tubing *a* to the heating or reboiling coil *b* where the major portion of the available heat is removed. The partially condensed steam then passes to a condensing coil *c* where it is completely condensed and cooled. This coil is made up in the conventional manner using one tube within another. The condensed and cooled steam sample now passes through valve *d* which is opened wide. Valve *e* is kept closed. The sample passes up and part of it overflows to waste *f* or a conductivity cell, if the conductivity of the undegasified steam sample is desired. A portion of the condensed sample flows through an orifice *g*, through a preheating coil *h* where it discharges onto the top of the plates *i*. The flow of this sample is constant due to the constant head of water above the orifice. This sample is representative of the condensed steam since it has been condensed and cooled prior to passing through the orifice. It contains all the dissolved solids present in the steam along with the dissolved gases.

The sample passes down through the scrubbing or stripping column which is made up of a series of plates. The column could be packed with various types of packing, or other types of plates might be used. It has been found that the plates described give efficient operation with very low pressure drop. By the time the sample reaches the bottom plates, all of the dissolved gases have been removed from the water. This gas-free water then falls into the bottom reservoir or reboiling chamber. The heat from the steam passing through coil *b* boils the gas-free water and thus furnishes gas-free steam which passes up through the column and removes the gas from the sample flowing down.

The gas-free sample in the reservoir flows out through a cooling coil *j* and then at the proper temperature is available to flow to a conductivity cell *k*. The conductivity of the sample gives a measure of the dissolved solids directly without correcting for dissolved gases since it is gas-free.

The steam, after passing up through the column, is condensed in the top vent-type condenser *l* where the gases pass out the top *m* and the condensed steam drops back on the top plate. The condenser is so operated that no appreciable amount of steam is allowed to be lost through the vent.

In order to obtain efficient operation (remove all the gases such as CO_2, NH_3, H_2S, H_2, etc.), it is essential that the ratio of the amount of steam passing up through the column to the amount of the condensed-steam sample, being removed to the conductivity cell, be well above 1. We have found that, when this ratio is between 1.5 and 2, the gas removal is complete. In order to obtain this ratio, it is necessary to pass more steam through coil *b* than will be used at the conductivity cell *k*, so this excess is passed through the overflow *f*. In cases where the NH_3 is very low, this ratio may be reduced and the apparatus simplified somewhat. However, it appears better to build one unit which will remove all the gases which might be present than several units, which might be limited in application. It has been found that, by passing 60 to 70 lb per hr through the unit and using an orifice (about 0.0625 in. diam at *g*), the sample rate to the conductivity cell is 30 lb per hr. This gives the desired ratio of steam to sample in the column.

Apparatus Used for Sampling Condensate

When the unit is to be used for sampling condensate, the condensate is added at *n*. Valve *d* is closed and valve *e* is opened. Sufficient condensate is added so as to overflow constantly at *f*, thus assuring a constant rate of flow through the orifice *g* to the column. Any steam available (above 100 psi) may be then put in at *a* (again controlling flow to 60 to 70 lb per hr) and the condensed steam allowed to flow through valve *e* to waste. The gas-free-condensate sample will flow out through *k* to the conductivity cell. The dissolved solids will be present, but the gases will be removed.

This unit differs from those previously used in several ways. Thus J. K. Rummel used the reboiling principle, but he condensed the steam along with some of the gases and reboiled a solution containing gases. We reboil a gas-free sample. M.

Hecht and D. S. McKinney condensed a sample of steam and applied corrections for the dissolved gases. S. T. Powell and I. G. McChesney condensed in a vacuum but still had to correct for dissolved ammonia. No one has, to our knowledge, used the principle of the scrubbing tower or stripping column to remove these gases from a sample of condensed steam to be used for conductivity determinations. This principle has been used for other purposes such as purifying alcohols, etc.

The apparatus is compact, being about 3 ft high and 6 in. outside diam. It is only necessary to connect the steam-sample line, a cooling-water supply, and an overflow outlet. Fig. 2 is a view of the condenser, while Fig. 3 shows the construction of the plates in the column.

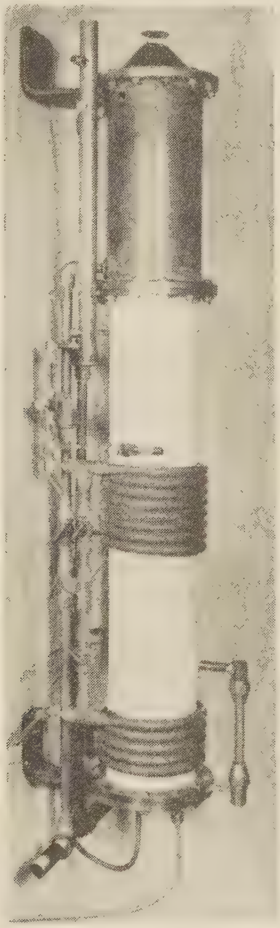

Fig. 2 View of the Condenser

The operation is practically automatic, once the unit has been installed and properly adjusted. The flow of the cooling water to the cooling coils is adjusted (valves *o* and *p*) as well as that to the top condenser, in order to limit the amount of cooling water. Further adjustment of these valves is necessary only at infrequent intervals. The use of an orifice to control the steam flow to the unit makes it unnecessary to adjust this flow. It is only necessary to check the overflow and sample rate from time to time to determine that the proper amount of steam is flowing.

TABLE 1 RESULTS OF TESTS IN UNIVERSITY OF ILLINOIS POWER PLANT

(150-Psi-gage saturated steam)

Method of condensing	NH_3, ppm	pH value	Specific conductance, micromhos
In coil under pressure..........	1.5	6.5	11.6
Modified Rummel............	0.9	9.7	6.2
New degasifying unit..........	0.0	7.4	1.4

Tests on Apparatus in Power Plants

One of the degasifying steam condensers was tested in the University of Illinois power plant. The steam available was 140 to 150 psi gage saturated steam. An orifice 0.125 in. diam × $^1/_4$ in. long was used and this allowed 71 lb of steam per hr to flow to the unit. The sample of degasified water flowed at a rate of 31 lb per hr. Table 1 gives the results of some tests conducted on this steam. The degasified steam had a specific conductance of 1.4 micromhos while the undegasified steam had a specific conductance of 11.6 micromhos. Thus, the gases had caused a change of 11.2 micromhos in a steam having a true value of only 1.4 micromhos.

Fig. 3 Construction of Plates in Stripping Column

A degasifying unit was installed in a large central station, using steam from the main steam line, having 1250 psi pressure and 900 F. The steam was passed through a steel tube, 0.06 in. ID by 0.25 in. OD, 11 ft long. This allowed 72 lb of steam per hr to flow through the unit and the sample rate to the conductivity cell was 32 lb per hr. The steam at the beginning had 0.67 ppm NH_3 and later this was reduced to 0.24 ppm. The degasify-

TABLE 2 RESULTS OF TESTS RUN ON 1250-PSI 900-F STEAM

Method of condensing	NH_3, ppm	Specific conductance, micromhos
In coil under pressure	0.67	5.93
In coil under pressure	0.61	5.07
New degasifying unit	0.00	1.40
New degasifying unit	0.00	1.52
In coil under pressure	0.24	2.90
New degasifying unit	0.00	1.25
New degasifying unit	0.00	1.63

TABLE 3 RESULTS OF TESTS RUN ON CONDENSATE FROM SAME PLANT AS TABLE 2

	NH_3, ppm	pH value	Specific conductance, micromhos
No degasifying	0.73	8.4	6.73
New degasifying unit	0.00	6.7 to 7.2	1.20

TABLE 4 RESULTS OF TESTS RUN ON CONDENSATE FROM EVAPORATOR AND BOILER STEAM

Source of steam	Method of condensing	CO_2, ppm	NH_3, ppm	Specific conductance, micromhos
Evaporator	In coil under pressure	18.2	0.03	7.5
Evaporator	Through degasifying unit	0.0	0.02	2.9
Evaporator	In coil under pressure	18.7	0.02	...
Evaporator	Through degasifying unit	0.0	0.01	3.0
Boiler	In coil under pressure	0.5	0.05	...
Boiler	Through degasifying unit	0.0	0.00	1.6

ing unit gave a sample free from ammonia. This unit has now been running more than 6 months in connection with a conductivity recorder. During this time, tests on the degasified sample showed them to be free from ammonia.

A unit was also tested in this same station using the condensate from the turbine. The steam used for the reboiling was about 225 psi gage, superheated about 200 F, and contained 1285 Btu per lb. An orifice 0.086 in. diam × $^1/_4$ in. long was used and gave a steam flow of 66 lb per hr through the unit. The sample flow of degasified water was 30 lb per hr. Here again the ammonia was reduced to 0.0 ppm. The pH value was reduced from 8.4 when the ammonia was present to 6.7 to 7.2 in the degasified sample. During the time of this test, the condenser leakage was low and the specific conductance of the gas-free sample was about the same as that of the gas-free steam sample. A unit has been installed which operates using the condensate from a central point which gives condensate from all the condensers. It is arranged so that, when the conductivity of the degasified sample varies and indicates condenser leakage, samples may then be taken from each condenser in turn, in order to determine which one is leaking. The results of the tests conducted in this station are given in Tables 2 and 3.

A unit was installed in another power plant in order to obtain a continuous record of the quality of the vapor from the evaporators. Since the pressure in the evaporators varied between 10 and 60 psi gage, it was not possible to use this vapor directly in the degasifying unit. It was first condensed and then run through the unit as condensate. The source of heating used was steam at 650 psi and 725 F. This was throttled through an orifice 0.05 in. diam by $^1/_4$ in. long. This gave a flow of 68 lb per hr through the unit. The evaporator distillate was passed through at a rate of 32 lb per hr. The ammonia content of the evaporator distillate at the time of the tests reported was very low, about 0.02 to 0.03 ppm, but the carbon dioxide was around 19 ppm. When this was passed through the degasifying unit, Table 4, the ammonia was reduced to about 0.01 ppm and the carbon dioxide to 0.0 ppm. The unit was then used to degasify the steam and it reduced the ammonia from 0.05 to 0.0 ppm and the carbon dioxide from 0.5 to 0.0 ppm.

In all determinations of pH value the glass electrode was used. In the University of Illinois tests the sample was kept free from contact with air until it had passed through the glass electrode container. In the central-station tests (Table 3) the sample was exposed to air while measuring the pH value. All conductance tests were run on the sample protected from contact with the air.

Ammonia Determination

The ammonia was determined by the Nessler method (4). In some tests the comparison was made in Nessler tubes and in others, the Hellige nitrogen-ammonia color disks were used for comparison. In all the tests reported as 0.00 ppm NH_3, the color after adding the Nessler reagent corresponded to the 0.00 standard color. In the test on the University steam when the Nessler test showed 0.00 ppm NH_3, 500-ml samples of the gas-free water were tested for NH_3 by evaporation (4). This method gave the NH_3 as equal to 0.02 ppm. This value may be high since the samples were not protected from contact with air while sampling. However, it shows that the ammonia had been reduced from 1.5 ppm to a maximum value of 0.02 ppm, or a removal of about 99 per cent of the ammonia. The residual ammonia would not affect the specific conductance by more than 0.2 micromho or be equivalent to more than 0.1 ppm of dissolved solids in the steam. Tests on the degasified samples for carbon dioxide showed it to be negligible in all tests.

The carbon dioxide was only determined in the steam or condensate before degasifying, in the tests reported in Table 4. This was possible due to the low ammonia present.

Conclusions

As a result of the tests conducted, it is possible to conclude that the degasifying unit will give a sample of water for conductivity determinations having the dissolved solids present in the absence of dissolved gases. It is possible to use this unit directly on all steams above 150 psi as well as on condensate.

BIBLIOGRAPHY

1 "Electrical-Conductance Measurements of Water and Steam, and Applications in Steam Plants," by Max Hecht and D. S. McKinney, Trans. A.S.M.E., vol. 53, 1931, paper FSP-53-11.

2 "Estimation of Solids in Steam by Conductivity," by J. K. Rummel, *Industrial and Engineering Chemistry*, Analytical edition, vol. 3, 1931, pp. 317–320.

3 "Design and Development of Apparatus for Measurement of Steam Quality by Electrical Conductivity Methods," by S. T. Powell, H. E. Bacon, Jr., I. G. McChesney, and F. Henry, Trans. American Institute of Chemical Engineers, vol. 33, 1937, pp. 116–138.

4 "Standard Methods for the Examination of Water and Sewage," Eighth edition, New York American Public Health Association, 1936, p. 41.

Discussion

R. E. Hall[4] and E. P. Partridge.[5] The marked advantages inherent in the conductivity method of determining purity of steam have enlisted a number of investigators over a period of several years in the drive to eradicate troublesome factors which, necessarily, have led to indeterminate error in the final results. The task has not been simple; various separate steps have had to be combined to provide the facility of measurement which is routine today.

One of these steps has been the development of suitable electrical-measuring equipment. The instrument makers have made a good job of this phase of the work. As one today uses a convenient and accurate conductivity meter, designed for 60- or 25-cycle current, and for 110 or 220 v, and provided with an attachment to compensate for the temperature of the sample to be measured, he can scarcely realize the difficulties which Fitze

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[5] Hall Laboratories, Pittsburgh, Pa.

(5),[6] Hecht and McKinney (1) or Rummel (2) had to overcome in their adaptation of equipment then available to the specific problems of steam. Perhaps it was their work which defined for the instrument makers the needs which they have filled.

Conductivity cells vary considerably in their construction, form, and dimensions, according to the individual preferences of their users. However, if the cell constant is appropriate and carefully determined, the form of the cell within reasonable limits will not affect the final accuracy of the determination.

Sampling of the steam has presented its own problems. We believe no exception has been taken to the recommendations for gathering the sample as set forth in the Power Test Codes (6). The trend is toward providing enough sampling connections to permit measurement of steam quality at all pertinent points. The value of this is well illustrated by Baker in his paper (7) and in particular by the problems in connection with the induction of new installations into service.

From the point at which condensation of the steam sample begins, on through the conductivity cell where the final determination is made, what shall be the composition of the pipes or vessels contacting the water, in order that they shall contribute the least contamination by dissolving therein? Some data are available on this question (8). Perhaps stainless steel as used by Powell (9) will finally prove best. A report of an investigation on this point and discussion thereof are being planned by Committee D-19 for the A.S.T.M. annual meeting in June, 1941; several other factors relating to determination of steam purity by conductivity will also be given consideration.

As the authors point out, methods of correction for, or elimination of, dissolved gases have necessarily been evolved, in order that the conductivity contributed by dissolved solids carried over from the boiler may be known. The values of Rummel (2) for CO_2 and NH_3 have been most extensively used for correction purposes. At times these have led to impossible results, as noted by the authors, and as pointed out also by Watson (10). The latter showed that it is incorrect to apply separate corrections for NH_3 and CO_2, because they are interreactive and, therefore, contribute to the conductivity in a measure which is not the sum of their separate conductivities, but which varies with the pH value which they jointly establish in the sample. When these parts are taken into consideration, the anomalies in the correction method are no greater than the percentage of error involved in the determinations.

The method of eliminating dissolved gases, as exemplified by the equipment developed by the authors, is a neat application to this problem of the principles of the stripping column. With equipment such as this for providing continuously a gas-free sample, and with sensitive conductivity meters to record the conductivity of the sample, certainly the record on quality of steam will be more simply obtained, and more accurate than heretofore. The writers would raise only one question, a point on which the authors have not touched, i.e., what is the composition of material utilized in the condenser and reboiler in order that they shall contribute the minimum dissolved metal ion to the sample?

BIBLIOGRAPHY

5 "Determination of Moisture in Steam by Electrical Conductivity," by M. E. Fitze, *Power*, vol. 68, 1928, pp. 484–485.

6 Power Test Codes, A.S.M.E. Series 1929, Instruments and Apparatus, Part 11, "Determination of Quality of Steam," pars. 69–74.

7 "Mechanical Purification of Steam Within the Boiler Drum," by M. D. Baker, Trans. A.S.M.E., vol. 61, 1939, pp. 711–720.

[6] Numbers in parentheses occurring in this and subsequent discussions from (1) to (4), refer to the Bibliography at the end of the paper; those from (5) to (10), inclusive, refer to the Bibliography at the end of this discussion.

8 "Determination of Purity of Steam by Gravimetric and Spectrographic Methods," by M. C. Schwartz, W. B. Gurney, and T. E. Crossan, Trans. A.S.M.E., vol. 62, Nov., 1940, pp. 719–722.

9 "Determination of Steam Quality," by S. T. Powell, *Combustion*, vol. 9, no. 5, 1937, pp. 25–31.

10 Discussion, by Alfred Watson, Trans. A.S.M.E., vol. 62, Nov., 1940, p. 732.

Max Hecht.[7] In order to make this paper more readily understandable, the authors are requested to amplify the following items in their closure:

The addition of the stripping column between the reboiler and the vent condenser represents a considerable improvement over the apparatus suggested by Messrs. Schwartz, Gurney, and Crossan,[8] which produced a water containing about 0.14 ppm residual ammonia nitrogen.

Operating Practice. The writer described in his paper (1) the utilization of a conductivity recorder for determining steam-condensate quality. This recorder was equipped with temperature compensation and manual compensators for cell constant and carbon dioxide. Similar recorders are being employed for steam quality from boilers as well as vapor purity from evaporators. It is noted in the Hecht and McKinney patent,[9] in the J. K. Rummel patent,[10] and in the Powell and McChesney patent,[11] that the apparatus and methods described in each are adaptable to continuous recording of specific conductance. The apparatus described by Schwartz and his associates, as well as commercially available degassifying units, are also adaptable for use with a recorder.

Corrections to Observed Conductance. Inasmuch as the authors do not describe the conductivity apparatus used in their experiments, and since no standard temperature is indicated for the specific conductance values reported in the paper, it is pertinent to ask if the room-temperature observations were compensated to some standard temperature. This temperature should be specified.

In view of the relatively large area of metals exposed in the authors' apparatus to both steam and condensate and the relatively slow rate of flow of the fluids through the apparatus, have the authors verified that no heavy metal salts are introduced into the sample, which would require a "water correction?" In the writer's experience the "water correction" may amount to as much as 0.8 or 0.9 micromho at 25 C.[12] Rummel (2) suggests 0.1 micromho. It is assumed by the writer, from an inspection of the paper,[13] that Schwartz deducts 0.05 at 25 C from his observed conductance. Straub[14] reports values of 0.50 and 0.33 micromho determined in steam condensate produced from a specially treated water. He used this supply for boiler-feed purposes in an experimental boiler and reports: "The specific conductance of the condensed steam was continually better than 1 micromho." Inasmuch as the authors describe special precautions to render this supply free of all gases, the writer concludes that the value of 1 or less micromho, represents contamination of the steam condensate by heavy metal salts. (It should be noted that this value of 1 or less is considerably higher than the suggested water corrections reported by Rummel and Schwartz. The specific-conductance values reported by the

[7] Pittsburgh, Pa.

[8] Bibliography (8), p. 720.

[9] U. S. Patent, No. 1,971,816.

[10] U. S. Patent, No. 2,046,583.

[11] U. S. Patent, No. 2,146,312.

[12] Bibliography (1), pp. 146 and 148.

[13] "Determination of Purity of Steam by the Electrolytic-Conductivity Method," by W. B. Gurney, M. C. Schwartz, and T. E. Crossan, Trans. A.S.M.E., vol. 62, Nov., 1940, Table 1, p. 730.

[14] "The Cause and Prevention of Steam Turbine Blade Deposits," by F. G. Straub, Bulletin No. 282, Engineering Experiment Station, University of Illinois, May 5, 1936, pp. 27 and 28.

authors in this investigation were not referred to a standard temperature.)

Although the authors do not use the ammonia corrections mentioned in the introduction to their paper, it should be pointed out that the value assumed by them is higher than experimentally determined values found by Rummel and by Schwartz. Rummel[15] found 7.3 micromhos per ppm nitrogen ammonia for ammonia alone in water, and 8 micromhos per ppm nitrogen ammonia for carbonated-ammonia solution. Schwartz[16] reported 8 micromhos per ppm nitrogen ammonia.

Nomenclature. It is observed that the terms "total dissolved solids," "total solids," and "dissolved solids" appear to be used interchangeably throughout the paper. The following definitions[17] on water for industrial use are given by the American Society for Testing Materials:

2 (*e*) *Dissolved Solids.*[18] "Dissolved solids" comprise the dried residue from evaporation of the filtrate, after separation of suspended solids.

2 (*f*) *Dissolved Salts.* "Dissolved salts" are the sum of the individually determined ions in a complete analysis.

The terms "dissolved solids" and "dissolved salts" are appropriate for use and applicable to the subject matter discussed by the authors.

A. E. Kittredge.[19] Preceding a broad discussion of steam-sampling equipment the writer wishes to commend the authors on the compact and effective mechanical design of the equipment they had described, in the light of the purpose for which this equipment was developed. Reference to the field of application for which this equipment is designed is purposely made because there is a fair distinction to be made between equipment designed to serve the single function of degasifying the steam sample for conductivity test and that for the dual function of both degasifying the steam sample for conductivity tests while yet permitting the collection of the separated gas for analysis. An appreciable demand for equipment of the latter type seems to be indicated by the need for detecting quickly the generation of hydrogen in high-pressure boilers and superheaters, resulting from the dissociation of steam; appearance of hydrogen in the sample, of course, indicating a dissociation and active corrosion by the free oxygen so liberated.

It seems impossible to discuss a paper of this kind technically without first establishing a few points of fundamental fact. In any physical process of gas removal there is no possible design which can produce an absolute zero in fact. Different designs emphasize different advantages but, in such a process, depending upon a driving force between the solvent and the solute, the end point must, from the nature of the process, still have an actual if not measurable difference between the actual value, whether measurable or not, and absolute zero. Appreciation of this fact is necessary to give proper evaluation to the different methods of design of degasifying equipment. This paper emphasizes the use of clean steam for flushing the fractionating tower and a counterflow arrangement of the condensed sample and the flushing steam. Both of these elements are in themselves desirable features if they can be utilized without sacrifice of other desirable features. The point we wish to make is that proper evaluation of all the elements entering into the degasifying process are necessary to determine the best cycle of operation for any particular equipment.

There are three basic factors to be considered in the design of degasifying equipment. These are:

1 The creation of a satisfactory equilibrium condition.
2 The selection of an advantageous operating temperature.
3 The provision of an effective degasifying means.

Equipments, designed to operate at relatively high vacuums and temperatures below 100 F, very easily produce satisfactory equilibrium conditions but are greatly handicapped by the higher viscosities of water at these temperatures. The higher viscosity of the water places a greater burden on the deaerating means in spite of the favorable equilibrium conditions. Degasification at low temperatures can be accomplished but operates under a definite handicap.

Operation of degasifying equipment at around atmospheric pressure with counterflow of steam and water provides a suitable equilibrium condition and utilizes the advantageously low viscosity of water at this temperature. In spite of the favorable equilibrium condition and operating temperature, the controlling limitation on the design of degasifying equipment for atmospheric operation will be the actual degasifying means. The latter is very apt to be handicapped and compromised in the design of small compact test equipment such as that under discussion.

Because the limiting factor in the design of degasifying equipment is the third element of the three tabulated, the design of equipment of the writer's company to be later described, utilizes the most effective degasifying means known, i.e., the atomizing method, at a very slight sacrifice to the most favorable equilibrium condition for the purpose of obtaining the greatest net effective result.

If a condensed-steam sample, containing as much as 1 cc per l of oxygen is flushed with an equal quantity of steam at atmospheric pressure in an open chamber without a counterflow arrangement, all but 1 part in 100,000 of the dissolved gas in the liquid would be transferred to the steam, if equilibrium were reached. That is to say, when the quantity of flushing steam equals the quantity of condensate to be deaerated and the steam itself contains 1 cc per l of oxygen, the presence of that oxygen would support in solution in the liquid only 0.00001 cc per l. This value ranges somewhere between 0.2 and 1 per cent of the smallest quantity of oxygen that can be determined by any known test method. It emphasizes the fallacy of limiting equipment design to conditions which are theoretically advantageous but practically worthless. For the same reason, we choose to place emphasis, in the design of our equipment, on effective means of degasification. Similar values apply to other gases, proportionate to their solubility and inversely proportionate to their specific volume at the operating conditions.

In contrast to the equipment presented by the authors, we wish to refer to equipment designed by the writer's company to serve all the purposes of the former equipment and, in addition, to make possible selection of gases removed from the steam sample for analysis. In the foregoing, we have briefly outlined reasons for placing emphasis on the effectiveness of the deaerating means as opposed to the obvious need of giving attention to satisfactory equilibrium conditions. In the degasification of a steam sample, involving the removal of carbon dioxide and ammonia, there is additional reason to use the most effective means of degasification possible.

Solutions of both carbon dioxide in water and ammonia in water form loose chemical combinations of carbonic acid and ammonium hydroxide respectively. Each also ionizes the solu-

[15] Data from curve prepared by J. K. Rummel and available through the courtesy of The Babcock & Wilcox Company, New York, N. Y.

[16] Footnote 13 of this discussion, refer to p. 729.

[17] "Tentative Methods of Reporting Results of Analysis of Industrial Waters," D596–40T A.S.T.M. Book of Standards, Supplement 1940, part 2, p. 541.

[18] The term "total dissolved solids" is not defined on pp. 56 and 92 of Bibliography (4), but what appears to be an ambiguous definition appears on page 151.

[19] Chief Engineer, Cochrane Corporation, Philadelphia, Pa.

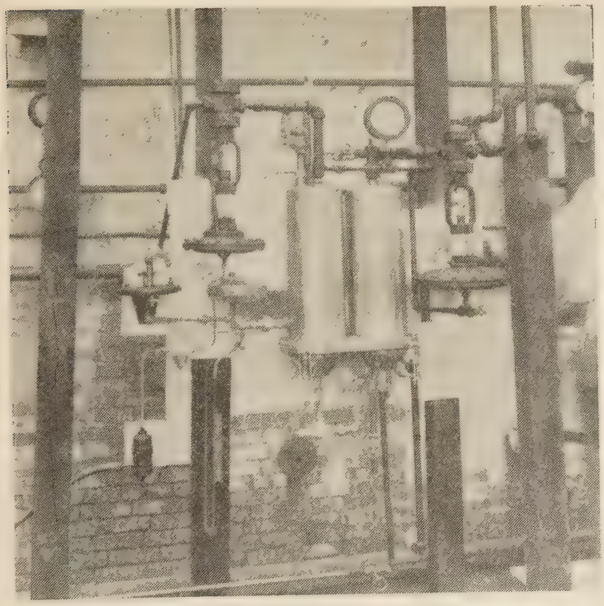

Fig. 4 View of Degasifier on Test

tion and the percentage of ionization of each dissolved gas increases as the total amount of gas in solution decreases. That part of either gas in solution as carbonic acid or ammonium hydroxide unionized exerts a gas pressure and is available for removal. It is from this fraction of the total gas in solution that diffusion of the gas particles from the liquid to the flushing steam occurs. It is apparent then that, as complete removal of the particular gas is approached, complete ionization of all the gas in solution is approached and the difficulty of removal of the remaining gas increases tremendously.

When dealing with a distilled-water sample, otherwise neutral, the presence of carbon dioxide will lower the pH value. As the carbon dioxide is removed, the pH value will rise toward the neutral point and the difficulty of removing the carbon dioxide will increase. On the other hand, the presence of ammonia in an otherwise neutral water sample will raise the pH value above the neutral point and the removal of the ammonia will lower the pH value and increase the difficulty of removal as the neutral point is approached.

The illustrations accompanying this discussion show details of the steam-sample degasifier mentioned.

The condensing and atomizing chambers, shown in Fig. 5, are 8-in-diam cylindrical vessels made of stainless steel. All parts of the equipment which contact the sample are stainless steel. The condensing and atomizing chambers are complete with relief valve, gage glasses, manometer connection, overflow connection, sampling connection, cooling coil, etc.

The pressure-control equipment consists of an 18-in-diam × 12-in-high constant-head tank with a $^3/_4$-in. float-operated regulating valve, two $^1/_2$-in. diaphragm-operated control valves, pressure regulator, air filter, pressure-reducing valve, constant-head chamber, and interconnecting piping and tubing for the operation of the controls.

A separate vent cooler is provided for cooling dissolved gases and condensing any steam passing beyond the main condenser. Necessary water and vent piping is supplied between vent cooler and condensing chamber and vent cooler and control equipment. A cooled-gas outlet is supplied on the vent cooler. A fixed orifice is provided for reduction of steam pressure in the steam-sampling line ahead of the degasifier.

A $^3/_4$-in. swing check valve and a $^1/_4$-in. pet cock are provided in the overflow line to regulate flow of condensate from the degasifier and prevent inflow of air through the overflow connection.

Method of Operation. The rate of steam sample supplied to the degasifier is controlled by a fixed orifice in the sampling line from the point at which the sample is taken, the orifice being designed to maintain a flow of 250 lb per hr.

Referring to Fig. 5, the sampled steam first enters the atomizing nozzle through the $^3/_4$-in. steam-inlet connection. The nozzle is designed to give the steam an appreciable pressure drop, approximately 50 psi. The energy thus made available serves to induce previously condensed steam to the nozzle, atomizing it thoroughly and removing the noncondensable gases from solution. Steam and noncondensable gases travel upward in the atomizing chamber and over to the condensing chamber. A separator is built into the top of the atomizing chamber to prevent excessive carry-over of the condensed sample from the atomizing compartment to the condensing chamber. In the condensing chamber, the steam is condensed and the condensate is with-

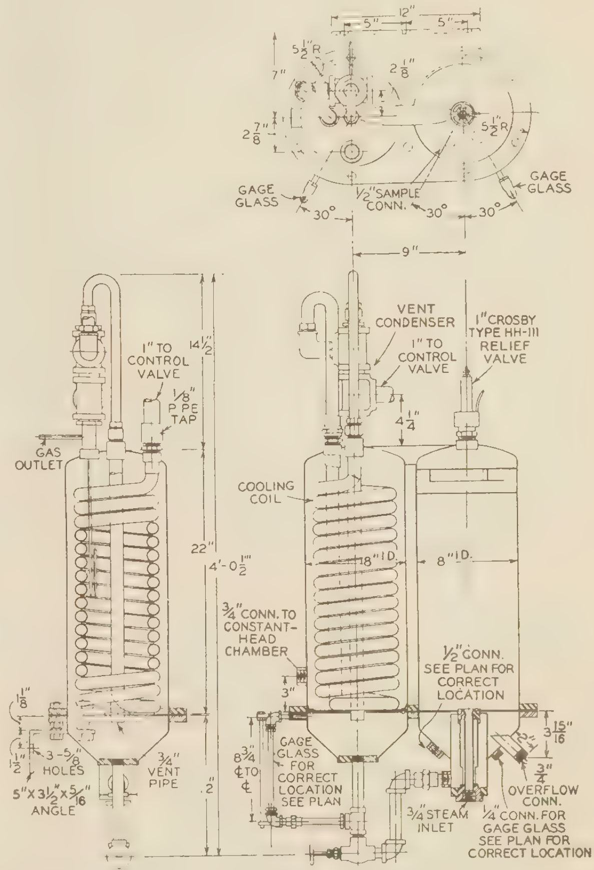

Fig. 5 Sectional Assembly of Degasifier

drawn to the atomizing chamber through the $^1/_2$-in. piping and needle valve, connecting the bottom of the two compartments. The needle valve is adjusted to maintain a level of condensate in the condensing chamber, as indicated in the gage glass.

The vent mixture is withdrawn, through the pipe extending to the bottom of the condensing chamber, to a small vent cooler where the remaining water vapor is condensed from the mixture, and the noncondensable gases cooled to approximately room temperature. The vent condenser and vent cooler are designed

to prevent appreciable accumulation of gases, so that the samples withdrawn can be analyzed without lag.

A continuous overflow is maintained from the sampling device for removing condensate from the degasifier. The rate of water discharged through the overflow connection is controlled by an adjustable orifice in the form of a $^1/_4$-in. pet cock at the base of a 2-ft leg. The variation in head on the orifice by the change in water level in the leg makes the orifice self-regulating. A check valve placed in the overflow line ahead of the orifice prevents air entering the atomizing chamber through the overflow connection, in case the pressure within the degasifier falls below atmospheric.

A $^1/_2$-in. sampling connection is placed at the bottom of the atomizing chamber for passing a sample of the condensed steam through a conductivity cell for determination of carry-over.

Steam pressure in the degasifier is maintained at 2 in. of water by controlling the amount of cooling water flowing through the condensing coil and vent cooler. Cooling water first enters the constant-head tank through the $^3/_4$-in. regulating valve which maintains a constant water level in the tank. There are two $^1/_2$-in. diaphragm-operated control valves for controlling the water flow. The first of these valves ahead of the degasifier has the diaphragm connected to the water line between the degasifier and the second valve, thus maintaining a constant water pressure at the inlet to the second valve. The second diaphragm-operated control valve is air-operated, being actuated by a pressure regulator briefly as follows:

The pressure regulator consists of a diaphragm, the bottom of which is connected through a constant-head chamber to the condensing chamber of the degasifier, and the top of which is connected to a leak-off valve. Air is supplied through an air filter and pressure-reducing valve, which maintains a constant pressure, ahead of the regulator, to the regulator. In passing through the regulator, air passes through a fixed orifice and then to the diaphragm on the second diaphragm-operated control valve. A leak-off valve is placed between the orifice and the diaphragm of the diaphragm-operated control valve. The pressure in the degasifier controls the position of the diaphragm of the pressure regulator which in turn controls the position of the leak-off valve, regulating the pressure under the diaphragm of the second diaphragm-operated control valve. The leak-off valve on the pressure regulator is of the compensating type to prevent overtravel of the diaphragm-operated control valve.

F. W. Quarles.[20] While others have used the principle of counterflowing condensate against the vapor, together with that of reboiling, in an effort to degasify and obtain the minimum amount of gases in solution in the condensed-steam samples, the writer believes that they have erred mainly in being too timid in the application of enough counterflowing and reboiling action.

The diagram shown in Rummel's paper (2) indicates that the degasifying action possible in the vent condenser was ignored and its effect on the results neglected also.

It occurs to the writer that, regardless of the scheme used, the gas vents will be accompanied by vapor which in all probability will contain a negligible amount of soluble solid matter. This will cause the condensed sample reaching the conductivity cell to have a greater soluble-solids content than in the steam. Has it been determined that the correction for this in the authors' apparatus is negligible?

The main principle involved in almost all of the schemes of degasification is that recognized by the Henry-Dalton gas laws and concerns the equilibrium relationship between the amount of gas associated with the vapor and liquid phases of the water.

Assuming equilibrium as calculated by these laws for two cases, one with 3 ppm of O_2 and the other with 3 ppm of NH_3 in the steam (with correction for dissociation in the case of ammonia and no undercooling of the condensate in either case), the amount of ammonia in the water solution would be of the order of 130 times that of oxygen.

In actual practice, equilibrium will not be reached because of the need of an infinite amount of surface, and the removal will be less complete than indicated. However, even in the case of ammonia the amount of removal by this method would seem to be worth-while and, on first thought, it would appear peculiar that the authors allow first that condensation take place in a small-bore tube where proper advantage of this action cannot be taken so that a greater duty is thrown on the scrubbing column. However, when consideration is given to the need for an extra amount of heat for reboiling and the difficulty of obtaining it from the sample-steam line without disturbing the concentration of soluble solids in the sample reaching the conductivity cell, the value of this expedient can be appreciated.

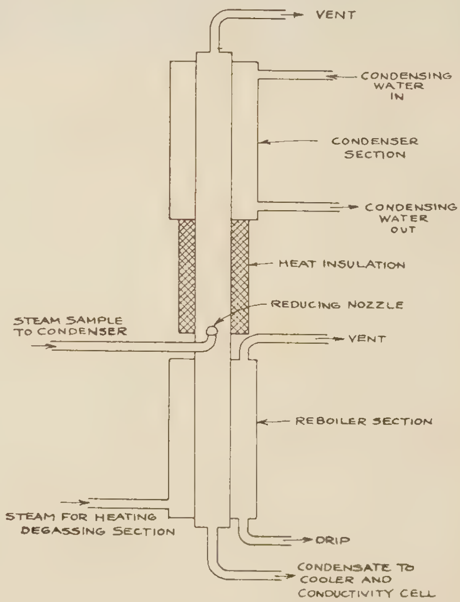

Fig. 6 Diagram of Simple Degasifying Apparatus

It may be of interest for the writer to present his idea of a simple apparatus shown diagrammatically by Fig. 6 of this discussion, which was suggested to S. T. Powell in a private discussion of the paper (3).

The apparatus suggested can be assembled by plant mechanics from materials readily obtainable; use of $1^1/_4$-in. IPS inner tube and 2-in. outer pipe with $1^1/_4$-in. $\times$ 2-in. fittings being suggested, it being preferable to use a full 20-ft length of inner tube. The tube diameters, however, are dependent upon the amount of steam sample to be degassed. By employing this arrangement, the water can be kept in a thin film to allow quick diffusion of gas to the interface where maximum allowable steam-scrubbing velocity is maintained, allowing a small margin of safety against holdup of the condensate. Not only the condensing, but also the reboiling, is done counterflow in film form.

It is considered desirable to heat-insulate the steam-sample line in order to have the steam enter the apparatus in a superheated state and, thereby, boil off some of the condensate in the

[20] Assistant to Superintendent of Power Production Stations, Consolidated Gas, Electric, Light and Power Company, Baltimore, Md. Mem. A.S.M.E.

middle section which to some extent corresponds to the middle section of the authors' equipment.

The reducing nozzle, it is believed, cannot very well be used for total pressure reduction, it being used mainly for the purpose of causing a swirling action of the steam, thereby increasing turbulence and quickly removing the gas which has diffused into the gas-vapor film at the interface.

It is obvious that other forms of heat may be used for the reboiler section equally as well as steam heat. The vertical height necessary for this design and the necessity for a separate heat supply for reboiling, however, are points against it when compared to that of the authors, assuming a 20-ft-length tube to be necessary.

To the writer, it seems remarkable that the authors' apparatus can produce adequate results in so short a vertical distance, since he had visualized a 20-ft vertical distance as about right.

The writer wishes also to point out that such equipment is well suited for analyzing the gaseous content of the steam, since good removal is obtained.

J. B. Romer.[21] Ever since J. K. Rummel developed the Babcock & Wilcox degasifying condenser, which he reported in his paper (2), there has been a great deal of comment regarding the corrections necessary. This comment has, in numerous cases, taken the form of questioning the ability to correct when both ammonia and carbon dioxide are present. The writer would like to make it quite clear at this point that, when this piece of apparatus is properly operated, the carbon dioxide is completely eliminated and proper correction can be applied for the residual ammonia.

By applying the principles of perforated-plate rectifying columns to the condenser described, the authors have made a contribution to the art which is well worth-while and gives us a compact piece of equipment which does not require correction factors and, hence, permits the attachment of a direct-reading recorder.

One of the serious problems formerly encountered was that of convincing the boiler owner or engineer that the conductivity method was reliable, his objection being that the correction amounted to as high, in some cases, as 90 per cent of the total reading. This objection has been overcome by making several comparative studies of the quality of steam condensate. The conductivity of several samples of steam condensate was first determined and then large volumes of the same condensate were carefully evaporated and the residue carefully analyzed by exact analytical methods. We found that the results checked within satisfactory limits and thereby overcame the objection to correction factors. As a result, conductivity is now a recognized method for determining the quality of steam condensate.

[21] Chief Chemist, The Babcock & Wilcox Company, Barberton, Ohio.

Authors' Closure

The stainless steel (18-8) is utilized in the condenser wherever the metal is in contact with steam or the condensate. In tests which have been run in the laboratory we have been able to obtain a product from the condenser having a specific conductance of 0.18 micromho at 25 C. This would indicate that there is a minimum amount of dissolved metal ion in the sample. The results which have been reported in this paper have all been corrected to 25 C.

A High-Temperature Bolting Material

By A. W. Wheeler,[1] Schenectady, N. Y.

In the process of providing new materials or old with improved heat-treatments to withstand the increasing temperatures employed in present-day steam turbines, many studies are being made on alloy steels and their heat-treatment which are most suitable for use as bolting material. This paper reviews a series of tests on heat-treatment, creep, rupture, and hardness, together with the application of the results to bolting-material practice.

With the increasing temperatures for which steam turbines are being designed, it becomes necessary to provide new materials or old materials with improved heat-treatments to insure equally successful operation under the more severe conditions of service.

The problem of heat-treatment must be given most careful consideration. However, heat-treatment is only one of the essential factors in the production of steel for high-temperature use. Perhaps the most important factor, and one not so uniformly controlled, is the melting practice. With the different types of furnaces now in common use, the methods of deoxidizing and adding alloying elements have a direct bearing not only upon heat-treating characteristics, but upon the physical and creep properties and the structural stability under high temperature.

In the development of steel by composition and heat-treatment for high-temperature operation, account must be taken of all the service requirements. The final acceptable result for any type of steel will probably be something of a compromise between the various properties, as it is not possible to have all properties meet the maximum values.

The tests covered in this paper are as follows:

1 Effect of heat-treatment on room-temperature physical properties.

2 Long-time high-temperature creep tests of the relaxation type.

3 Long-time rupture tests at high temperature.

4 Effect of time at high temperature on room-temperature hardness.

Heat-Treating Tests

The series of heat-treating tests covered in this paper was made on a material which has been on the market for a number of years, and the manufacturing processes are well established.

Tests were made on a 4-in-diam bar stock, electric-arc furnace heat, of the following composition: Carbon 0.45, chromium 0.99, molybdenum 0.35, vanadium 0.26, manganese 0.61, and silicon 0.32.

The quenching part of the heat-treatment was done on the full-size stock, after which each piece was split into quarter segments for the different draw temperatures. The test coupons were taken out about half way from the center to the outside of the bar.

These tests were undertaken primarily to find the required heat-treating cycle to improve the notched impact strength, since this is a bolting material which must have high sharp-notch impact resistance. Complete physical-test results on the heat-treating series are shown in Table 1.

Figs. 1 to 5, inclusive, present graphically the effect of heat-treatment on the physical properties at room temperature. Of these, Figs. 1, 4, and 5 show the effect of draw temperature on elastic limit, impact strength, and elongation, respectively.

While all of these charts indicate definite trends, perhaps the most striking is Fig. 2, which shows a definite optimum quenching temperature of 1650 F for the highest Charpy strength, regardless of the rate of cooling. Transposing these same data to show relationship between elastic limit and Charpy impact strength, as in Fig. 3, it will be observed that impact strength increases with increase of cooling rate in the quench and that there is a parallelism between the results obtained with the Charpy specimen and with the 60-deg V-notch specimen. The standard keyhole specimen is 10 mm square and 50 mm between bearing points. The notch exactly cuts the specimen in two, leaving a net section which is 5 mm by 10 mm. The V-notch impact-test piece is the same size as the standard keyhole Charpy specimen, but with a 60-deg sharp notch and a net area of cross section which is the same as that of the standard keyhole Charpy. This type of specimen was used for testing bolting material for two reasons, i.e., the type of notch closely approximates in shape the American National Standard thread, and the net section of the specimen, being the same as that of the keyhole specimen, facilitates comparison.

Standard keyhole Charpy and 60-deg V-notch impact tests were also made at various temperatures up to 1000 F. Results are shown in Fig. 6. It will be noted that this material is not sensitive to notches at high temperatures.

In order to try the effectiveness of heat-treatment in the larger sizes of stock, physical tests were made on specimens taken from points at different distances from the center of a $6^3/_{16}$-in-diam bar of composition and heat-treatment as shown in Fig. 7. The elastic limit of these specimens varied from 105,000 psi on the center specimen to 110,000 psi on the outer specimen, as shown graphically in Fig. 7. The dilatation curve Fig. 8 shows linear change under heating and cooling.

Creep Tests

A program of creep testing was started prior to the heat-treating investigation. Table 2 shows the chemical composition and heat-treatment of the creep specimens, and Table 3 contains the "before-creep" and "after-creep" physical properties. A summary of results of creep tests at various test temperatures is given in Table 4. All creep tests were relaxation tests made by the step-down or "flow-rate" method,[2] the total elastic plus plastic extension being limited to 2 mils per in.

Log-log stress-time plots were made for each item, also log-log stress-creep rate. Figs. 9 to 14, inclusive, show these results.

Micrographs at $\times 1000$ were made on items Nos. 863 to 870, inclusive, showing the structures before and after creep tests at 950 and 1000 F, Figs. 15 to 18, inclusive. There is no appreciable change in any of the specimens except the air-cooled item No. 864, Fig. 15, which had 2545 hr under stress at 1000 F. The after-

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[1] Turbine Engineering Department, General Electric Company.

Contributed by the Joint Research Committee on Effect of Temperature on Properties of Metals, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[2] For a more complete description of this method, see Progress Report by Subcommittee for Project 16 of the A.S.M.E.-A.S.T.M. Joint Research Committee on the Effect of Temperature on the Properties of Metals entitled, "The Resistance to Relaxation of Materials at High Temperature," by Ernest L. Robinson, Trans. A.S.M.E., vol. 61, 1939, pp. 543–554.

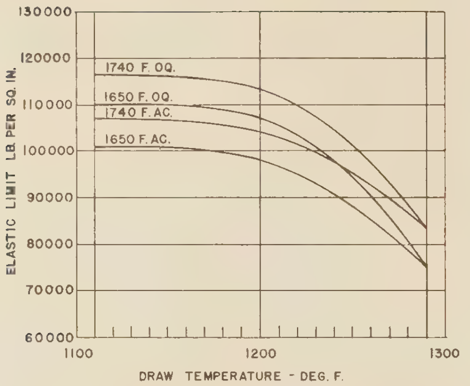


FIG. 1 CURVES SHOWING EFFECT OF DRAWING TEMPERATURE ON ELASTIC LIMIT FOR VARIOUS INITIAL QUENCHES AS RECORDED ON CURVES
(Refer to Table 1.)

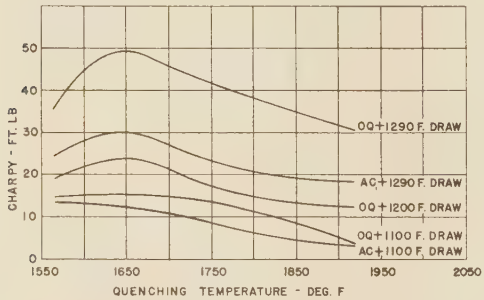


FIG. 2 CURVES SHOWING EFFECT OF QUENCHING TEMPERATURE, TYPE OF QUENCH, AND DRAWING TEMPERATURE ON CHARPY IMPACT STRENGTH
(Refer to Table 1.)

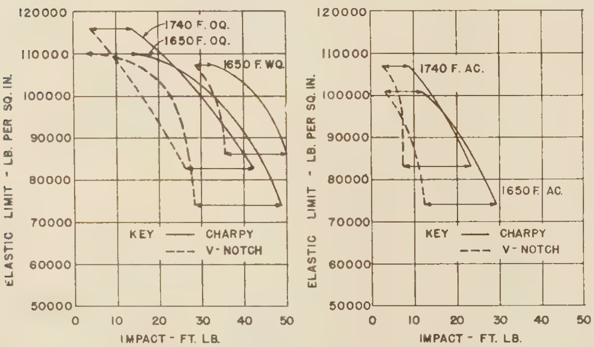


FIG. 3 CURVES SHOWING RELATIONSHIP BETWEEN KEYHOLE CHARPY AND 60-DEG V-NOTCH IMPACT STRENGTH AND ELASTIC LIMIT FOR VARIOUS TYPES OF QUENCH
(The curve is established by three different drawing temperatures on each curve; refer to Table 1.)

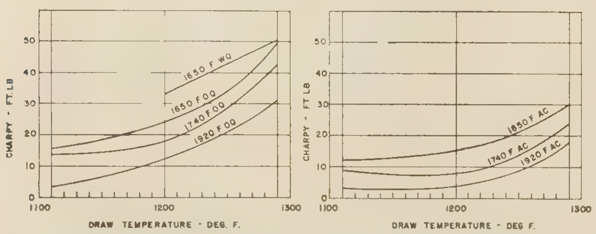


FIG. 4 CURVES SHOWING EFFECT OF DRAWING TEMPERATURES ON CHARPY IMPACT STRENGTH FOR VARIOUS INITIAL QUENCHES
(Refer to Table 1.)

TABLE 1 PHYSICAL PROPERTIES AT 70 F FOR HEAT-TREATING TESTS

(All stresses in pounds per square inch)

ITEM NO.	HEAT TREATMENT	TENSILE STRENGTH	ELASTIC LIMIT	ELONG. IN 2 IN. %	RED. A. %	IMPACT-FT. LB.	
						CHARPY	V-NOTCH
1	1560 F 8 HR. AC., 1110 F. 4 HR. FC.	140600	101000	19.5	51.9	13.4	
2	1560 F. 8 HR. AC., 1200 F. 4 HR. FC.	129400	77000	21.5	54.6	15.1	
3	1560 F 8 HR. AC., 1290 F. 4 HR. FC.	100900	65000	26.0	60.6	23.6	
4	1560 F. 8 HR. OQ., 1110 F. 4 HR. FC.	150900	104000	18.5	51.6	14.5	
5	1560 F. 8 HR OQ., 1200 F. 4 HR. FC.	134600	104000	21.0	57.4	18.5	
6	1560 F. 8 HR OQ., 1290 F 4 HR. FC.	105100	74000	25.5	61.6	34.8	
7	1650 F. 8 HR. AC., 1110 F. 4 HR. FC.	146600	101000	18.5	50.5	12.2	3.4
8	1650 F. 8 HR. AC., 1200 F. 4 HR. FC.	133400	98000	20.0	52.6	15.1	5.6
9	1650 F. 8 HR. AC., 1290 F 4 HR. FC.	105100	74000	27.0	62.5	29.9	12.8
10	1650 F. 8 HR. OQ., 1110 F. 4 HR FC.	159100	110000	17.0	51.1	15.1	3.4
11	1650 F. 8 HR. OQ., 1200 F. 4 HR. FC.	141600	107000	19.5	55.9	23.8	16.3
12	1650 F. 8 HR. OQ., 1290 F. 4 HR. FC.	106100	74000	24.5	64.2	49.4	28.8
13	1650 F. 8 HR WQ., 1110 F. 4 HR FC.	143700	107000	18.5	56.0	33.0	28.5
14	1650 F. 8 HR. WQ., 1200 F 4 HR. FC.	135200	104000	19.5	59.4	39.3	33.7
15	1650 F. 8 HR. WQ., 1245 F. 4 HR. FC.	126900	95000	21.5	61.8	41.8	34.5
16	1650 F. 8 HR. WQ., 1290 F. 4 HR. FC.	114400	86000	24.5	64.2	50.3	35.3
17	1740 F. 8 HR. AC., 1110 F. 4 HR. FC.	156100	107000	18.5	50.5	9.0	2.6
18	1740 F. 8 HR. AC., 1200 F. 4 HR. FC.	144100	104000	19.0	50.8	7.5	5.2
19	1740 F. 8 HR. AC., 1290 F. 4 HR. FC.	113900	83000	23.5	59.1	23.8	7.5
20	1740 F. 8 HR. OQ., 1110 F. 4 HR. FC.	174400	116000	16.5	48.6	13.9	3.8
21	1740 F. 8 HR. OQ., 1200 F. 4 HR. FC.	154600	113000	18.5	53.3	17.6	6.5
22	1740 F. 8 HR. OQ., 1290 F. 4 HR. FC.	114100	83000	23.5	64.4	42.6	25.6
23	1920 F. 8 HR. AC., 1110 F. 4 HR. FC.	168600	107000	15.0	33.1	3.0	
24	1920 F. 8 HR. AC., 1200 F. 4 HR. FC.	154900	104000	15.5	38.2	3.4	
25	1920 F. 8 HR. AC., 1290 F 4 HR. FC.	120400	83000	19.0	55.6	18.2	
26	1920 F. 8 HR. OQ., 1110 F. 4 HR. FC.	184800	113000	14.5	35.4	3.4	
27	1920 F. 8 HR. OQ., 1200 F. 4 HR. FC.	170600	116000	15.0	36.7	12.2	
28	1920 F. 8 HR. OQ., 1290 F. 4 HR. FC.	118600	83000	22.0	53.9	30.8	

HEAT TREATMENTS MADE ON 4 IN. DIAM BAR
CHEMICAL COMPOSITION - 0.45 C., 0.99 CR., 0.35 MO., 0.26 V., 0.61 MN., 0.32 SI.

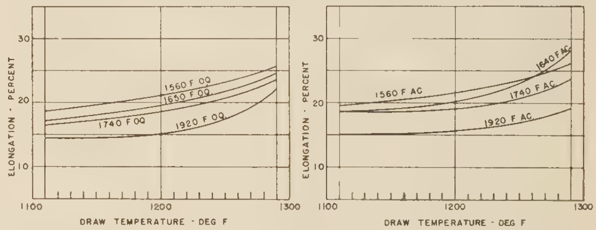


FIG. 5 EFFECT OF DRAWING TEMPERATURE ON PERCENTAGE OF ELONGATION FOR VARIOUS INITIAL QUENCHES
(Refer to Table 1.)

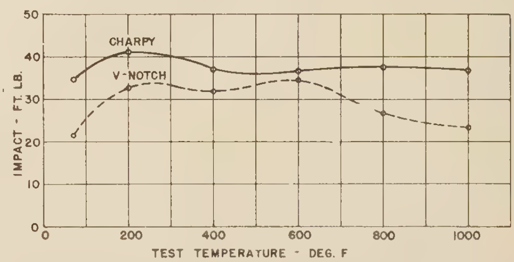


FIG. 6 CURVES SHOWING RESULTS OF KEYHOLE CHARPY AND 60-DEG V-NOTCH IMPACT TESTS MADE AT TEMPERATURES UP TO 1000 F

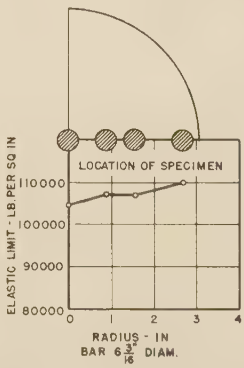


FIG. 7 DIAGRAM SHOWING VARIATION IN ELASTIC LIMIT IN SPECIMENS TAKEN FROM CENTER TO OUTSIDE OF 6 3/16-IN-DIAM BAR
(Chemical composition: 0.44 C, 0.53 Mn, 0.22 Si, 0.97 Cr, 0.31 Mo, 0.25 V, 0.012 P, 0.018 S. Heat-treatment: 1650 F for 8 hr, oil-quenched; 1250 F for 4 hr, air-cooled.)

TABLE 2 COMPOSITION AND HEAT-TREATMENT OF CREEP SPECIMENS

ITEM NO	DIAM OF STOCK-IN	CHEMICAL COMPOSITION C	CR	MO	V	MN	SI	HEAT TREATMENT
810	1	0.46	0.95	0.44	0.22	0.56	0.22	1740F 8HR. OQ, 1200F. -2 HR. FC.
837	1	0.46	0.95	0.44	0.22	0.56	0.22	1740F. 8HR. AC., 1200F. -2 HR. FC.
838	1	0.46	0.95	0.44	0.22	0.56	0.22	1740F. 8HR. AC, 1200F. -2HR FC.
863	4 *	0.45	0.99	0.35	0.26	0.61	0.32	1700F. 2HR. AC., 1180F. -2HR. FC.
865	4 *	0.45	0.99	0.35	0.26	0.61	0.32	1650F. 8HR. OQ., 1200F. -8HR. FC.
867	4	0.45	0.99	0.35	0.26	0.61	0.32	1650F. 8HR. OQ., 1200F. -8HR. FC
869	4	0.45	0.99	0.35	0.26	0.61	0.32	1650F. 8HR. WQ., 1250F. -4HR. FC.
864	4 *	0.45	0.99	0.35	0.26	0.61	0.32	1700F 2HR. AC., 1180F. -2HR FC.
866	4 *	0.45	0.99	0.35	0.26	0.61	0.32	1650F. 8HR OQ, 1200F -8HR FC
868	4	0.45	0.99	0.35	0.26	0.61	0.32	1650F. 8HR. OQ., 1200F. -8HR FC.
870	4	0.45	0.99	0.35	0.26	0.61	0.32	1650F. 8HR. WQ., 1250F. -4HR. FC

* HEAT TREATED HALF SEGMENT OF 4 IN. DIAMETER BAR.

TABLE 3 PHYSICAL PROPERTIES AT 70 F OF SPECIMENS BEFORE AND AFTER CREEP TEST

ITEM NO	TEST TEMP DEG. F	DURATION HOURS	CONDITION	TENSILE STRENGTH	ELASTIC LIMIT	ELONG IN 2 IN %	RED A %	CHARPY FT. LB.	MOD. OF ELAS. AT TEST TEMP.
810	932	3375	BEFORE CREEP	192800	164000	15.5	55.0	14.5	27000000
			AFTER CREEP	183000	146000	16.4	46.1	14.5	
837	932	3870	BEFORE CREEP	159000	113000	18.0	48.9	12.8	26400000
			AFTER CREEP	163000	116000	15.5	47.9	3.79	
838	932	3870	BEFORE CREEP	147400	116000	18.5	53.4	13.8	26000000
			AFTER CREEP	150000	116000	16.0	50.7	15.1	
863	950	2503	BEFORE CREEP	149200	107000	18.5	51.4	19.5	22700000
			AFTER CREEP	146900	116000	19.3	53.3	11.7	
865	950	2503	BEFORE CREEP	152700	110000	16.5	54.4	33.7-36.1	22200000
			AFTER CREEP	145100	116000	17.9	60.0	34.5	
867	950	2503	BEFORE CREEP	131900	95000	21.0	56.0	29.9	27800000
			AFTER CREEP	134200	100000	20.0	60.5	25.6	
869	950	2503	BEFORE CREEP	130700	95000	21.5	54.6	41.0	27200000
			AFTER CREEP	131000	97000	22.8	62.2	35.3	
864	1000	2545	BEFORE CREEP	149200	107000	18.5	51.4	19.5	21400000
			AFTER CREEP	140500	102000	20.0	53.4	13.9	
866	1000	2545	BEFORE CREEP	152700	110000	16.5	54.4	33.7-36.1	21500000
			AFTER CREEP	137000	108000	20.0	60.0	31.7	
868	1000	2545	BEFORE CREEP	131900	-95000	21.0	56.0	29.9	22500000
			AFTER CREEP	127000	93600	20.7	60.8	25.6	
870	1000	2545	BEFORE CREEP	130700	95000	21.5	54.6	41.0	20700000
			AFTER CREEP	130000	100000	23.6	61.5	36.1	

TABLE 4 CREEP-TEST RESULTS

(All stresses in pounds per square inch)

ITEM NO	TEST TEMP DEG F	RESIDUAL STRESS AFTER 10000 HR	CREEP STRENGTH 0.1% PER 100000 HR	CREEP STRENGTH 1% PER 100000 HR	n SLOPE ON RATE-STRESS LOG-LOG PLOT	n-1 SLOPE ON TIME-STRESS LOG-LOG PLOT
810	932	20400	20300	27070	8.0	7.0
837	932	21800	21500	28700	8.2	7.2
838	932	23200	23200	29700	10.45	9.45
863	950	16500	16400	21960	8.0	7.0
865	950	14000	13700	18300	7.75	6.75
867	950	17000	17000	22480	7.75	6.75
869	950	13400	13900	18000	9.0	8.0
864	1000	8600	8700	13800	5.0	4.0
866	1000	6200	6100	11400	3.7	2.7
868	1000	7800	7900	13480	4.6	3.6
870	1000	5100	5600	9200	4.7	3.7

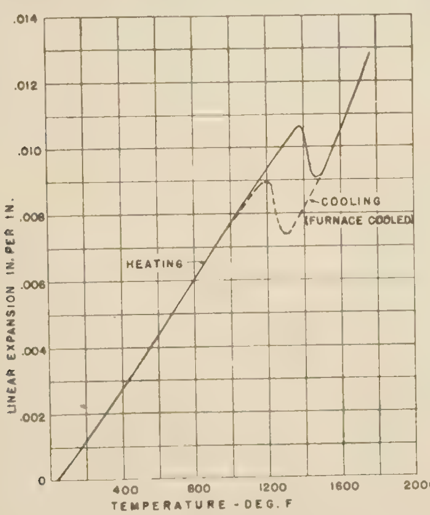


FIG. 8 DILATATION CURVE SHOWING LINEAR CHANGE UNDER HEATING AND COOLING

(Specimen heated in 1 hr to maximum temperature, held 1 hr, then furnace-cooled at 240 F per hr. Chemical composition: 0.45 C, 0.99 Cr, 0.45 Mo, 0.26 V, 0.61 Mn, 0.32 Si.)

creep specimen shows an apparent increase in ferritic areas due to carbide spheroidization and migration of carbon to the grain boundaries, resulting in considerable loss in the initially low Charpy strength.

It is recognized that creep strength falls off as the quenching rate increases, but the Charpy impact strength increased with the higher quenching rate. The oil-quenched treatment finally selected for commercial bolting is a compromise between creep strength and Charpy impact strength, sacrificing slightly in creep strength to provide much greater impact strength and, in addition, greater structural stability.

Rupture Tests

Long-time rupture tests[3] were made at 900 and 1000 F on material like creep-test item No. 866, which had been oil-quenched and drawn. In running a long-time rupture test, a series of bars is pulled at successively lower stress, and periods of sojourn at high temperature, required to cause failure, are plotted on log-log paper to enable prediction of a long-time strength. At 900 F, the fractures were always transcrystalline, the longest time for fracture being about 5000 hr under 60,000-psi stress. At 1000 F, the fractures were transcrystalline up to 1200 hr, with the first intercrystalline failure occurring at 3400 hr.

Comparative tests made on normalized material, like creep-test item No. 864, showed transcrystalline failure up to 140 hr and intercrystalline failure at 310 hr.

These rupture tests are conducted like regular constant-stress creep tests so that elongation-time plots, as well as stress-time plots, can be made.

Plotted results of rupture tests at 900 F on oil-quenched and drawn material are shown in Fig. 19 and at 1000 F in Fig. 20. Results of rupture tests on normalized material at 1000 F are shown in Fig. 21.

Correlation of Creep and Rupture Tests

Fig. 22 shows the results of creep and rupture tests in relation to each other, comparing the creep rate of 1 per cent per 100,000 hr to the 100,000-hr rupture strength, as determined by extrapolation of the observed data. Structural changes in the material beyond the time of longest test may change the results but that is a matter of conjecture. It will be noted that the ratio between creep strength and rupture strength is greater in the case of the normalized material than for the oil-quenched material, but this is quite possible because of structural difference and is a metallurgical phenomenon which is hard to explain at the present time.

After a larger number of comparative creep and rupture tests have been made, perhaps something more definite can be determined in this creep-rupture relationship, but it is the belief of the author that changes of heat-treatment, differences in melting practice, and even slight changes in some alloying elements in the composition, will greatly affect the ratio of creep strength to rupture strength.

Hardness Tests

Tests were made to determine the effect of time and temperature on the hardness of chromium-molybdenum-vanadium bolt material. The composition of the bar tested was carbon 0.45, chromium 0.98, molybdenum 0.35, vanadium 0.27, manganese 0.57, and silicon 0.28.

[3] For a more complete description of methods of running long-time rupture tests refer to "The Fracture of Carbon Steels at Elevated Temperatures," by A. E. White, C. L. Clark, and R. L. Wilson, Trans. American Society for Metals, vol. 25, September, 1937, pp. 863–888; also "Fracture of Steels at Elevated Temperatures After Prolonged Loading," by R. H. Thielemann and E. R. Parker, *Metals Technology*, April, 1939, Technical Publication No. 1034.

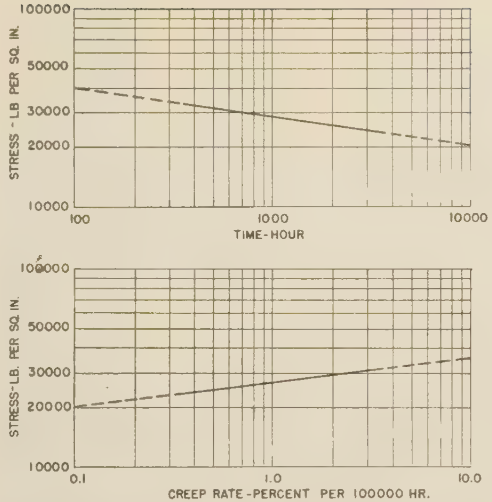


FIG. 9 RELAXATION CREEP TEST ON ITEM NO. 810; LOG-LOG PLOTS SHOWING RELATION BETWEEN STRESS VERSUS TIME AND STRESS VERSUS CREEP RATE

(Test temperature 932 F, oil-quenched material. Refer to Tables 2, 3, and 4.)

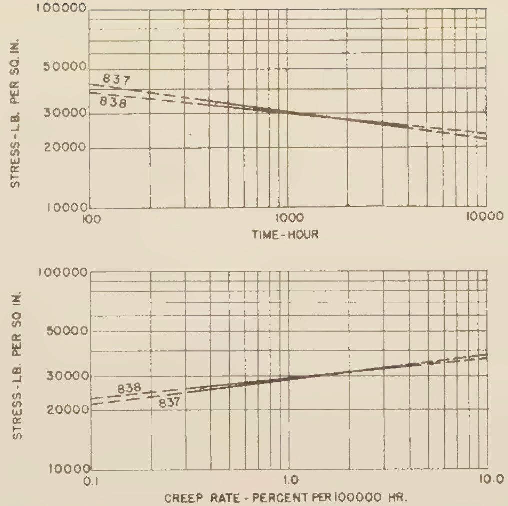


FIG. 10 RELAXATION CREEP TEST ON ITEMS NOS. 837 AND 838; LOG-LOG PLOTS SHOWING RELATION BETWEEN STRESS VERSUS TIME AND STRESS VERSUS CREEP RATE

(Test temperature 932 F, air-cooled material. Refer to Tables 2, 3, and 4.)

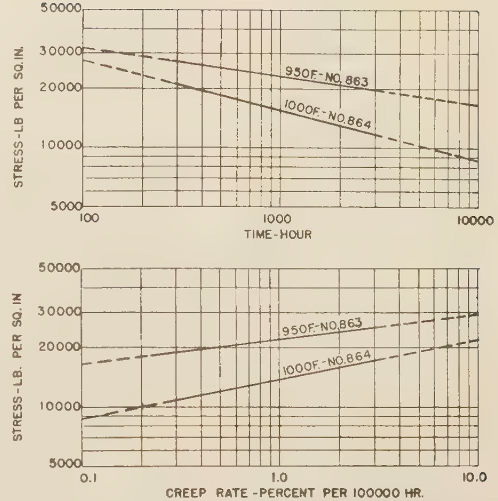


FIG. 11 RELAXATION CREEP TEST ON ITEMS NOS. 863 AND 864; LOG-LOG PLOTS SHOWING RELATION BETWEEN STRESS VERSUS TIME AND STRESS VERSUS CREEP RATE

(Test temperatures 950 and 1000 F, air-cooled material. Refer to Tables 2, 3, and 4.)

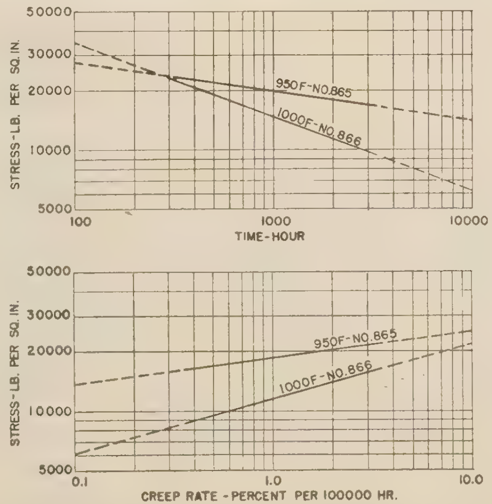


FIG. 12 RELAXATION CREEP TEST ON ITEMS NOS. 865 AND 866; LOG-LOG PLOTS SHOWING RELATION BETWEEN STRESS VERSUS TIME AND STRESS VERSUS CREEP RATE

(Test temperatures 950 and 1000 F, oil-quenched material. Refer to Tables 2, 3, and 4.)

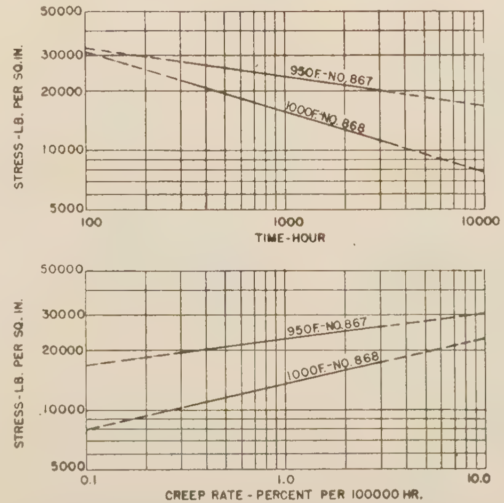


FIG. 13 RELAXATION CREEP TEST ON ITEMS NOS. 867 AND 868; LOG-LOG PLOTS SHOWING RELATION BETWEEN STRESS VERSUS TIME AND STRESS VERSUS CREEP RATE

(Test temperatures 950 and 1000 F, oil-quenched material. Refer to Tables 2, 3, and 4.)

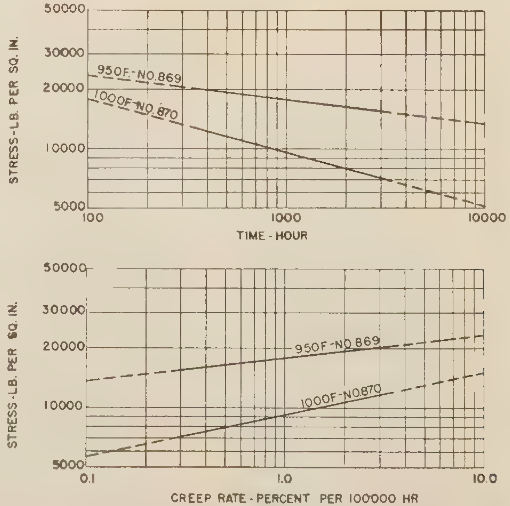


FIG. 14 RELAXATION CREEP TEST ON ITEMS NOS. 869 AND 870; LOG-LOG PLOTS SHOWING RELATION BETWEEN STRESS VERSUS TIME AND STRESS VERSUS CREEP RATE

(Test temperatures 950 and 1000 F, water-quenched material. Refer to Tables 2, 3, and 4.)

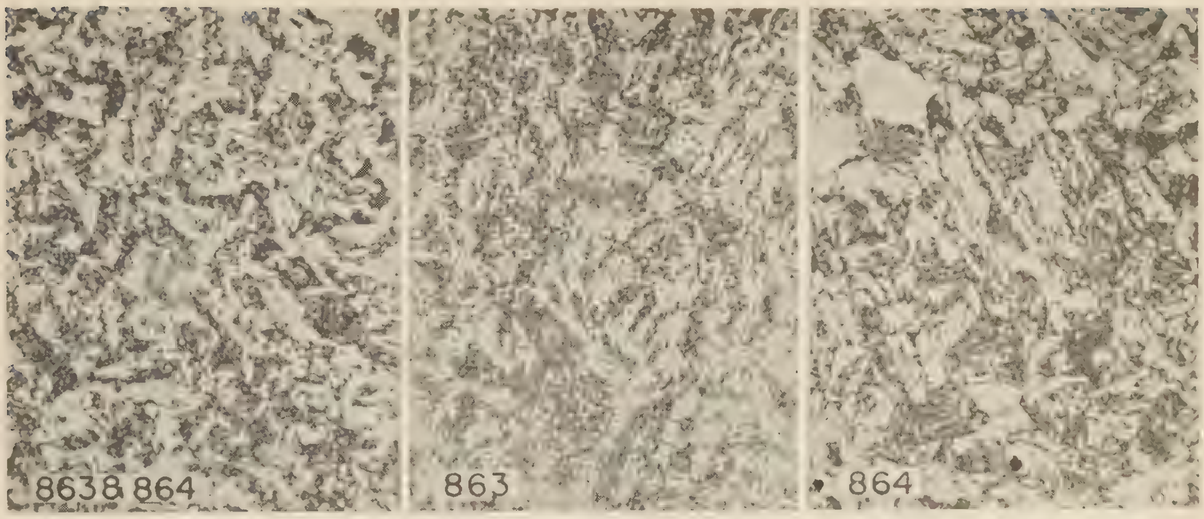


FIG. 15 MICROGRAPHS SHOWING CREEP-TEST ITEMS NOS. 863 AND 864 BEFORE CREEP, NO. 863 AFTER CREEP AT 950 F, AND NO. 864 AFTER CREEP AT 1000 F

(Heat-treatment before creep test, air-cooled from 1700 F and drawn at 1180 F; etched with 5 per cent nital; ×1000.)

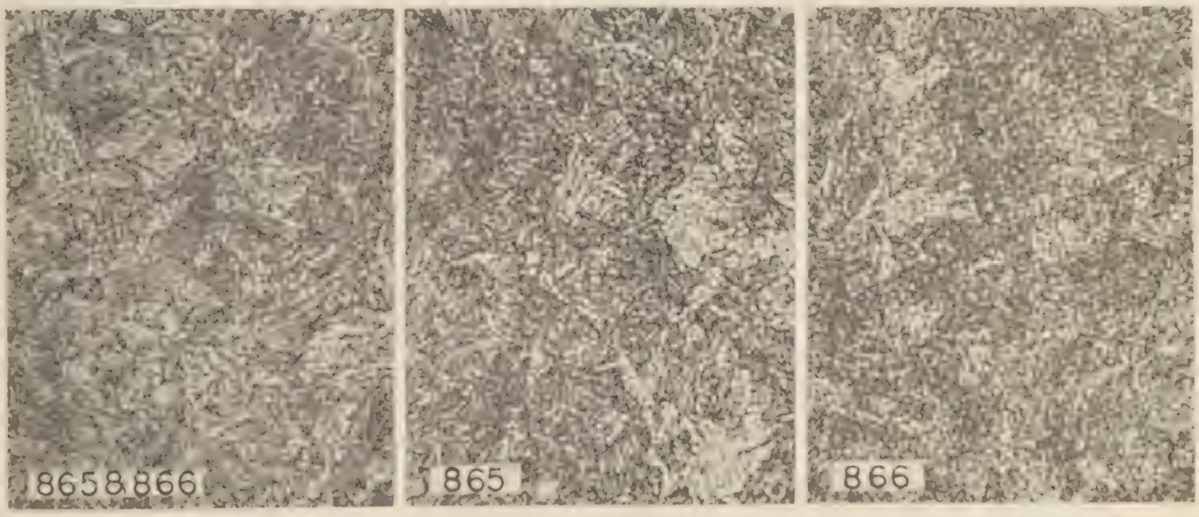


FIG. 16 MICROGRAPHS SHOWING CREEP-TEST ITEMS NOS. 865 AND 866 BEFORE CREEP, NO. 865 AFTER CREEP AT 950 F, AND NO. 866 AFTER CREEP AT 1000 F

(Heat-treatment before creep test, oil-quenched from 1650 F and drawn at 1200 F; etched with 5 per cent nital; ×1000.)

Three bars, $3^1/_{16}$ in. diam, were oil-quenched after 8 hr at 1640 F. Rockwell B hardness tests were then made. The bars which were identified as bars *A*, *B*, and *C* were then drawn at 1110, 1200, and 1290 F, respectively. After a 4-hr draw, a quarter segment was cut from each for complete physical tests. The draw was then resumed. At the end of 129 hr, a second quarter segment was taken from bar *C*. At 291 hr, a second quarter was taken from both bars *A* and *B*, and a third quarter from bar *C*. At 1000 hr, another segment was taken from each bar, completely using up bar *C*, and leaving a final quarter of bars *A* and *B*, which were re-treated with the initial quench, followed by the 4-hr draw. Results of physical tests made throughout this investigation are shown in Table 5. It will be noted that the elastic limit on the re-treated specimens is considerably higher than that obtained in the initial heat-treatment. This is due mostly to size effect. The initial heat-treatment was made on a 3-in-diam bar and the final treatment on a quarter segment of the 3-in. bar. Some improvement might also be caused by diffusion which broke up the

TABLE 5 PHYSICAL PROPERTIES AT ROOM TEMPERATURE FOR HARDNESS-TEST SERIES

(All stresses in pounds per square inch)

ITEM NO.	HEAT TREATMENT	TENSILE STRENGTH	ELASTIC LIMIT	ELONG IN 2 IN %	RED A %	ROCKWELL B	IMPACT FT LB [illegible]	[illegible]
A	1650 F 8 HR OQ					[illegible]		
	1650 F 8 HR OQ, 1110 F 4 HR FC	163200	123000	16.0	51.8	[illegible]	[illegible]	[illegible]
	1650 F 8 HR OQ, 1110 F 291 HR FC.	128900	110200	19.0	58.1	[illegible]	[illegible]	[illegible]
	1650 F 8 HR OQ, 1110 F 1000 HR FC	116200	100400	21.0	59.8		[illegible]	
	1650 F 8 HR OQ, 1110 F 1000 HR FC. 1650 F 8 HR OQ, 1110 F 4 HR FC	[illegible]	145400	16.0	[illegible]	[illegible]	[illegible]	[illegible]
B	1650 F 8 HR OQ, 1200 F 4 HR FC	137200	113200	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	1650 F 8 HR OQ, 1200 F 291 HR FC.	103400	85400	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
	1650 F 8 HR OQ, 1200 F 1000 HR. FC	97400	85400	25.0	[illegible]		[illegible]	
	1650 F 8 HR OQ, 1200 F 1000 HR FC 1650 F 8 HR OQ, 1200 F 4 HR FC	[illegible]	143900	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
C	1650 F 8 HR OQ, 1290 F 4 HR FC.	118500	95200	22.0	63.0	[illegible]	[illegible]	[illegible]
	1650 F 8 HR OQ, 1290 F 129 HR FC	97400	84700	28.5	63.0	[illegible]	[illegible]	[illegible]
	1650 F 8 HR OQ, 1290 F 291 HR FC	89900	75700	26.5	54.6	91.5	20.1-18.2	8.0-8.0
	1650 F 8 HR OQ, 1290 F 1000 HR FC	80900	70400	27.5	48.6		12.8-13.4	

BAR STOCK - 3" DIAM. CHEMICAL COMPOSITION - 0.45 C, 0.98 CR, 0.35 Mo, 0.27 V, 0.57 Mn, 0.28 Si

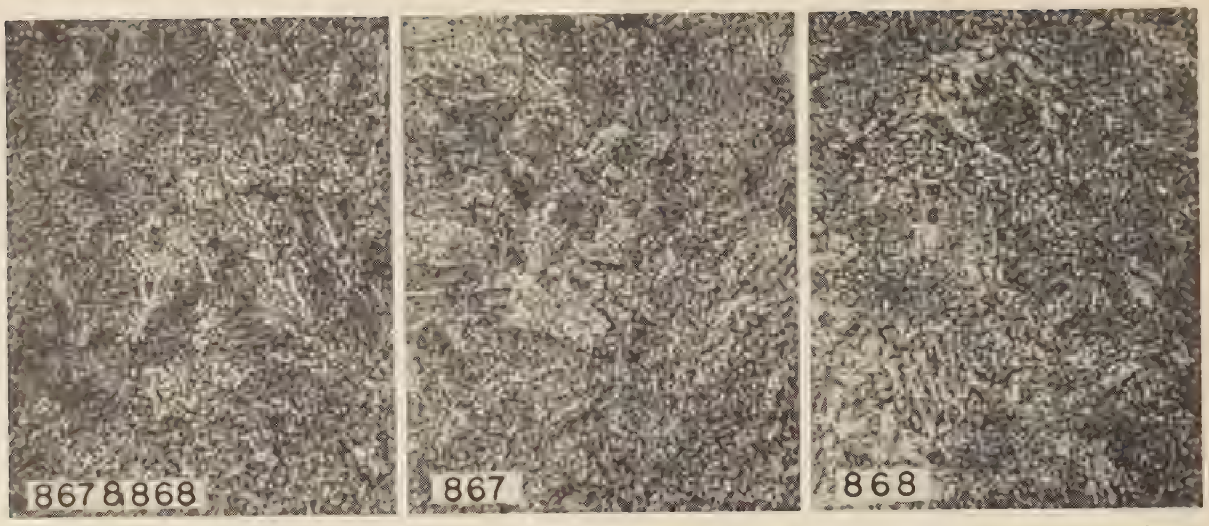


Fig. 17 Micrographs Showing Creep Test Items Nos. 867 and 868 Before Creep, No. 867 After Creep at 950 F, and No. 868 After Creep at 1000 F

(Heat-treatment before creep test, oil-quenched from 1650 F and drawn at 1200 F; etched with 5 per cent nital; ×1000.)

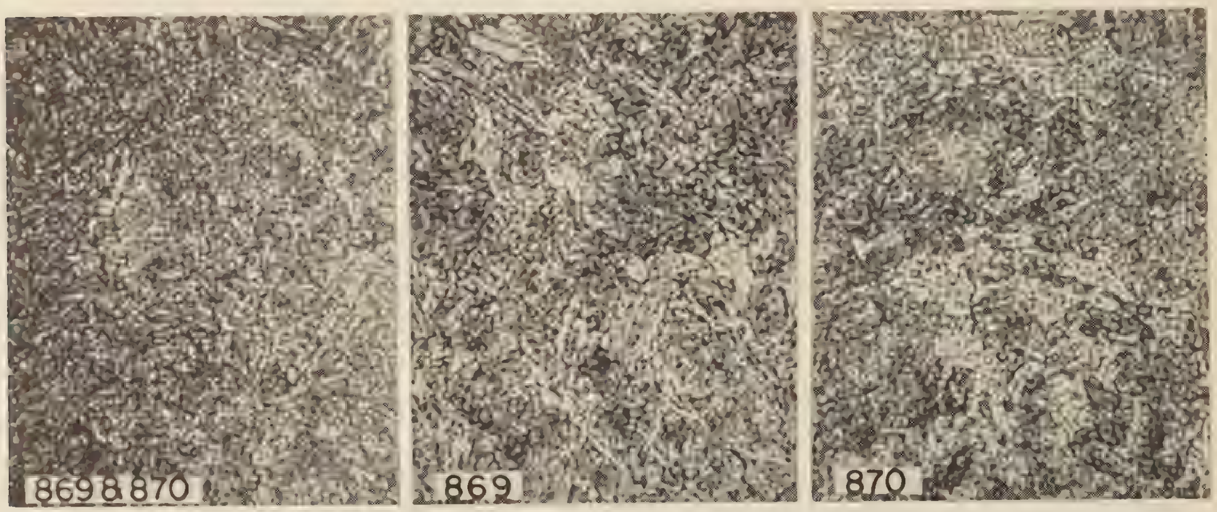


Fig. 18 Micrographs Showing Creep-Test Items Nos. 869 and 870 Before Creep, No. 869 After Creep at 950 F, and No. 870 After Creep at 1000 F

(Heat-treatment before creep test, water-quenched from 1650 F and drawn at 1250 F; etched with 5 per cent nital; ×1000.)

original banded condition. Fig. 23 shows graphically the effect of draw temperature and time upon the elastic limit. At the end of 1000 hr, the structural differences for the three drawing temperatures are reflected in the relative values of elastic limit.

Micrographs showing structural changes were made for each of the three draws after 4, 481, and 1000 hr. These are shown in Fig. 25. Referring to Table 5, it will be noted that, in the completely spheroidized state, as shown in micrograph C-1000, the minimum Charpy value is 13.3 ft-lb. Note also that the Charpy strengths of C-4 and B-481 are the same, though B-481 is partially spheroidized.

Another interesting thing to note is the effect on impact strength of the different draw temperatures, which is shown graphically in Fig. 24. The plotted results of the hardness tests are shown in Fig. 26.

As this is a precipitation-hardening material, Rockwell B hardness tests were made at intervals in an effort to determine the elapsed time for precipitation-hardening at the various draw temperatures. It will be noted that for the 1110 F draw, there are three periods of precipitation-hardening shown, the first occurring in 4 hr or less. This first period does not show in the 1200 and 1290 draws because it was over before 4 hr had elapsed and softening had begun. These three periods may be caused by each of the three alloying elements, chromium, molybdenum, and vanadium combined with carbon or even other more complex carbides.

Conclusion

This series of tests might be extended indefinitely and the results qualified to some extent. Different sizes of bar stock will show a difference in creep and rupture strengths and of course the physical characteristics will vary. Since these tests were made on 4-in-diam material, it is assumed that the results are applicable to large sizes of bolts, but any variation in properties of smaller sizes is safely covered in allowable working stresses.

(Figs. 19–26 follow on pages 661 and 662)

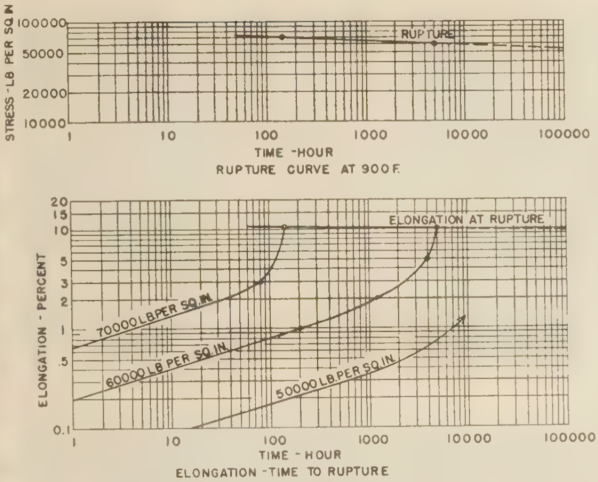


FIG. 19 LONG-TIME RUPTURE TEST, SERIES B7B: PLOTS SHOWING STRESS VERSUS TIME TO RUPTURE AND ELONGATION VERSUS TIME CURVES

(Test temperature 900 F; material oil-quenched and drawn like creep-test item No 866; refer to Fig. 12.)

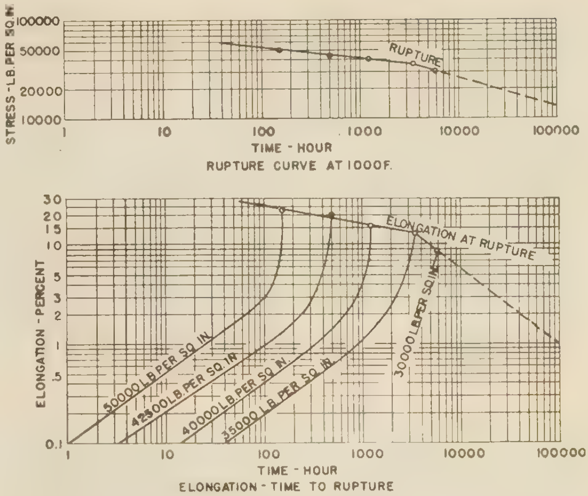


FIG. 20 LONG-TIME RUPTURE TESTS SERIES B7B: PLOTS SHOWING STRESS VERSUS TIME TO RUPTURE AND ELONGATION VERSUS TIME CURVES

(Test temperature 1000 F; material oil-quenched and drawn like creep-test item No. 866; refer to Fig. 12.)

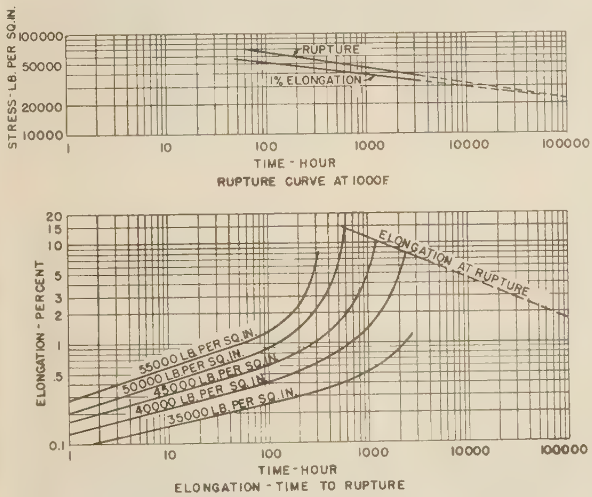
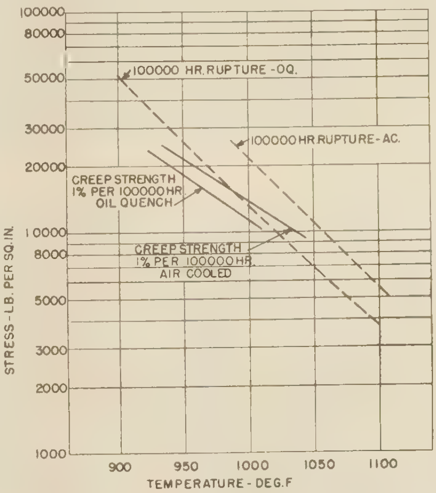


FIG. 21 LONG-TIME RUPTURE TEST SERIES B7A: PLOTS SHOWING STRESS VERSUS TIME TO RUPTURE AND ELONGATION VERSUS TIME CURVES

(Test temperature 1000 F; material air-cooled like creep-test item No. 864; refer to Fig. 11.)



(For creep strength figures see Table 4, items 863 and 864 air-cooled and items 865 and 866 oil-quenched.
Plotted data for both creep and rupture strengths are based on specimens cut from heat-treated half segments of 4-in-diameter stock.)

FIG. 22 COMPARISON OF CREEP AND RUPTURE STRENGTHS IN OIL-QUENCHED AND AIR-COOLED MATERIALS

		100,000 Hr rupture strength, psi	
Air-cooled	1000 F	22000	Fig. 21
Air-cooled	1100 F	5600	
Oil-quenched	900 F	53000	Fig. 19
Oil-quenched	1000 F	13200	Fig. 20
Oil-quenched	1100 F	3700	

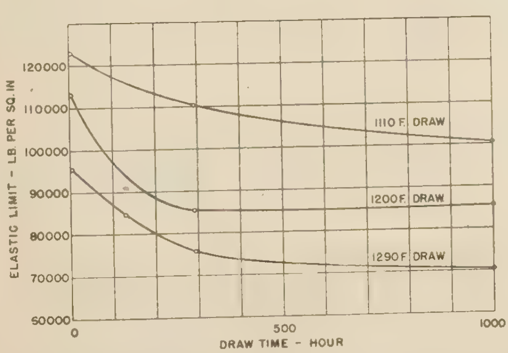


FIG. 23 EFFECT OF DRAWING TEMPERATURE ON ELASTIC LIMIT

Heat-treatment before draw, 1650 F for 8 hr, oil-quenched; refer to Table 5.)

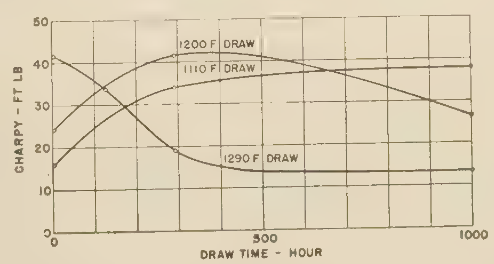
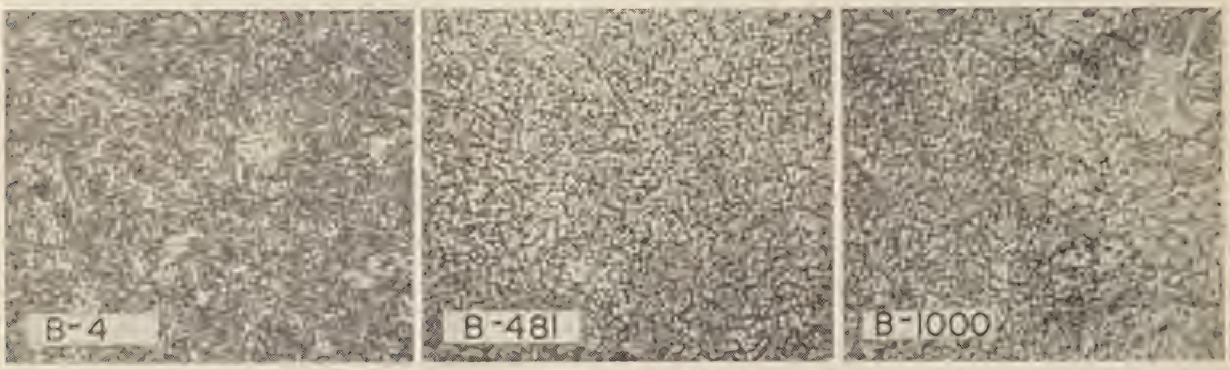


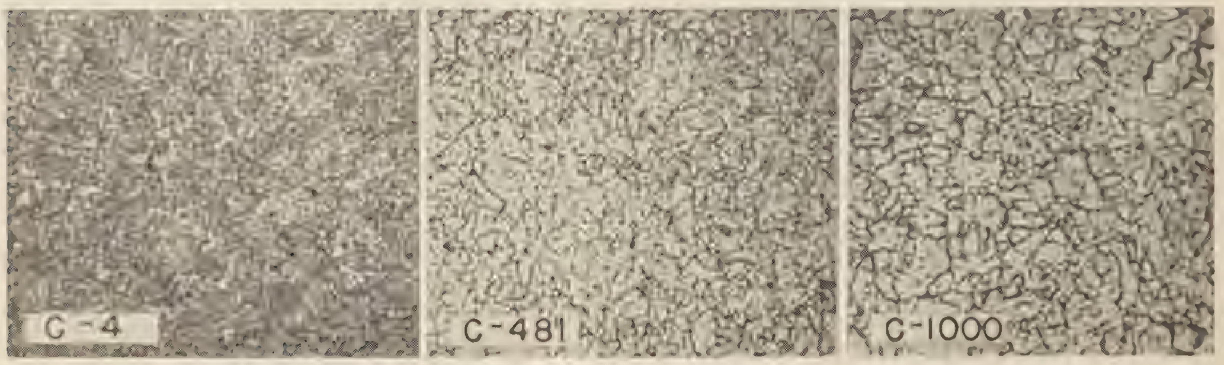
FIG. 24 EFFECT OF DRAWING TEMPERATURE ON KEYHOLE CHARPY IMPACT STRENGTH

(Heat-treatment before draw, 1650 F for 8 hr, oil-quenched. Refer to Table 5.)

TEST SERIES, A



TEST SERIES, B



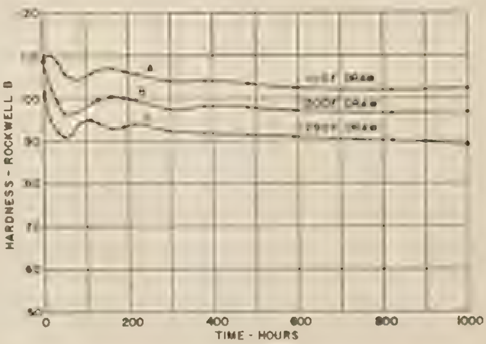
TEST SERIES, C

Fig. 25 Micrographs Showing Structural Change in Material, Oil-Quenched for 8 Hr, From 1650 F, Followed by Various Drawing Temperatures, Specimens Unstressed, ×1000

(Series A was drawn at 1100 F; items Nos. A-4, A-481, and A-1000 had 4, 481, and 1000 hr, respectively. Series B was drawn at 1200 F; items Nos. B-4, B-481, and B-1000 had 4, 481, and 1000 hr, respectively. Series C was drawn at 1290 F; items Nos. C-4, C-481, and C-1000 had 4, 481, and 1000 hr, respectively.)

Fig. 26 (Right) Effect of Drawing Time and Temperature on Rockwell B Hardness

(Heat-treatment before draw, 1650 F for 8 hr, oil-quenched. Chemical composition: 0.45 C, 0.98 Cr, 0.35 Mo, 0.27 V, 0.57 Mn, 0.28 Si. Refer to Table 5 for physical properties at start and after various draw times.)



Discussion

A. J. Herzig[4] and R. L. Wilson.[5] The author has directed attention to several aspects of the selection of bolting steels for high-temperature service which are not generally appreciated. Certainly there is slight agreement as to what constitutes an acceptable standard for judging the merits of a high-temperature bolting steel, but there is a growing realization of the many compromises which may have to be made in the choice of a material for a particular application.

The problem in searching for a good high-temperature bolting steel is to find a material having high strength at elevated temperatures combined with high room-temperature elastic strength, stability on heating, satisfactory notch toughness, and good machinability. It is also important to obtain these desirable properties in the heat-treatment of sizes ranging from $^1/_2$ to 4 in. or more with consistent uniformity both in the section treated and from lot to lot. High temperature strength would mean either the reluctance to relaxation of stress for a fixed strain or would be measured by the creep strain under constant stress.

From the rather meager data available it would seem that the constant-stress tests show somewhat higher relative strength values for air-treated as against quenched steels than are reported in the relaxation tests. This may be due to a persistent effect of a high initial rate of straining in the down-step test. At any rate there is ample evidence, supported by this paper, to indicate a preference for air-treated bolting steels to obtain best high-temperature strength were it not for the variation of room-temperature mechanical properties when the same heat-treatment is applied to a range of sizes.

We are now aware that seemingly small changes in microstructure can cause significant differences in mechanical properties, particularly the notch toughness and creep of steels. The microstructure and related properties will thus be changed by variations in chemical composition of the steel and by the rate of cooling in different sizes and media. For any preferred microstructure, the problem thus becomes one of hardenability of the steel. The hardenability of the steel can be changed by suitable adjustments in the chemical composition to produce the desired microstructure and associated properties by any kind of heat-treatment.

Since the normalizing and tempering treatment gives the highest strength at elevated temperatures, bolting materials should preferably be heat-treated in this manner by adjusting the composition to give a good compromise of room-temperature elastic strength and notch toughness, depending upon the sizes involved. Best all-round results will be achieved by using a normalizing temperature below the coarsening range, and increasing the hardening elements in the steel as section size increases. This might be handled commercially by selective application of steels to several size ranges.

J. J. Kanter.[6] The bolting steel upon which the author reports his extensive and valuable data conforms to a composition which has been known since 1936, as grade B14.[7] The chemical requirements for B14 steel as established in A.S.T.M. Specification A193-40T are as follows:

Element	Per cent
Carbon	0.35 to 0.50
Manganese	0.40 to 0.70
Phosphorus	0.04 Maximum
Sulphur	0.05 Maximum
Silicon	0.15 to 0.30
Chromium	0.80 to 1.10
Molybdenum	0.30 to 0.40
Vanadium	0.20 to 0.30

Minimum tensile requirements established for normalizing heat-treatments for sizes up to $2^1/_2$ in. diam and various draw temperatures are given in Table 6.

TABLE 6 MINIMUM TENSILE REQUIREMENTS FOR NORMALIZING AND VARIOUS DRAW TEMPERATURES

Minimum draw temp, F	Tensile strength, psi	Yield strength, psi	Elongation in 2 in., per cent	Reduction of area, per cent
1000	145000	120000	14	45
1100	135000	115000	15	45
1200	125000	105000	16	50

The properties obtained upon this steel, utilizing an air quench or normalize followed by a draw at 1200 F, are particularly notable as representing the class C physicals of A.S.T.M. Specification A96-39, which have long been recognized as desirable in high-strength bolting materials. This steel B14 was introduced into general high-temperature use upon the discovery that it responded to the normalizing treatment in such a manner as to attain exceptional creep and relaxation resistance at high temperatures and yet possess the high elastic strength implied for class C A96-39, i.e., 105,000 psi minimum yield strength.

Numerous alloy bolting steels were found which developed good creep strength upon normalizing and drawing at 1200 F (the lowest draw permissible for 1100 F service according to Spec. A193), but which failed to attain the elastic strength so essential to a good bolting steel. The chromium-molybdenum-vanadium composition was found to respond to tempering after normalizing in an entirely different manner from a chromium-molybdenum steel of equivalent composition but without a vanadium content. The B14 composition was found actually to increase in hardness

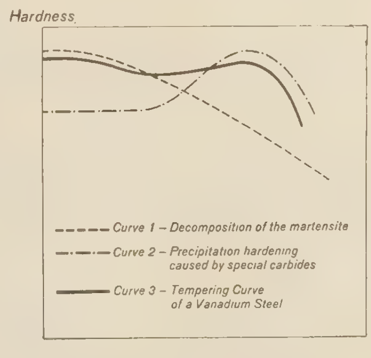


Fig. 27 Diagram Representing Superposed Phenomena Which Occur on Tempering Vanadium Steel Quenched From High Temperature[8]

upon drawing at 1200 F. Whereas, its "as normalized" hardness might be about 280 Brinell upon tempering at 1200 F, this increased to about 300 Brinell. This effect has been explained for vanadium steels by Houdremont, Bennek, and Schrader[8] as being due to precipitation hardening caused by separation of special carbides. Usually an air-hardening steel progressively softens

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[5] Metals Engineer, Climax Molybdenum Company, Canton, Ohio. Mem. A.S.M.E.

[6] Research Laboratories, Crane Company, Chicago, Ill. Mem. A.S.M.E.

[7] "Tentative Specifications for Alloy-Steel Bolting Materials for High-Temperature Service From 750 to 1100 F Metals Temperature," A193-40T, American Society for Testing Materials, 1936.

[8] "Hardening and Tempering of Steels Containing Carbides of Low Solubility, Especially Vanadium Steels," by E. Houdremont, H. Bennek, and H. Schrader, A.I.M.E. Technical Publication No. 585, Class C, Iron and Steel Division, 1934.

upon tempering due to the decomposition of martensite. Air-quenched B14 steel, however, seems to be hardened by a precipitation process which overcomes the softening tendency due to martensite decomposition, schematically illustrated in Fig. 27 of this discussion.

In Fig. 8, the author shows a dilation curve for the furnace cooling of the steel at 240 F per hr. This curve does not give a representative picture of the cooling transformations of this steel at the cooling rates obtained in the size of sections used for bolts when air-cooled. The author's curve, Fig. 8, shows complete transformation at Ar', which represents a completely pearlitic structure, whereas, in the air-cooling of sizes up to $2^1/_2$ in. diam, the rate is usually rapid enough at least partially to suppress the

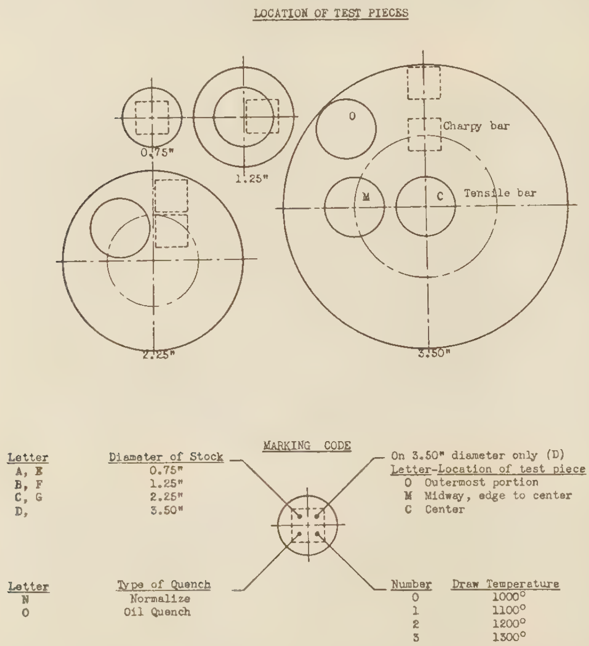


Fig. 28 Effect of Drawing Temperature, Size of Specimen, and Type of Quench on Physical Characteristics of B14 Bolting Steel

transformation to Ar'' with formation of some martensite. A complete investigation of the critical cooling rates of B14 steel is presently under way at the Crane research laboratories which have so far demonstrated that oil quenching in the steel is not, in any but the large diameters, necessary to effect an Ar'' transformation.

TABLE 7 CHEMICAL ANALYSES OF TEST SPECIMENS

Analysis no.	Diameter, in.	Marking	C, per cent	Cr, per cent	Mo, per cent	V, per cent
259391	0.75	A	0.35	0.95	0.35	0.32
259392	1.25	B	0.36	0.94	0.32	0.25
259393	2.25	C	0.45	0.94	0.32	0.26
259394	3.50	D	0.39	0.97	0.29	0.28
260325	0.75	E	0.38	0.95	0.36	0.25
260326	1.25	F	0.37	0.92	0.35	0.25
259394	2.25	G	0.39	0.97	0.29	0.28
258440	0.75	382	0.38	0.94	0.35	0.26

ABLE 8 HARDNESS OF SAMPLES IN QUENCHED CONDITION

		Hardness			
		VPN/30		Bhn	
Sample	Type of quench	Skin	Core	Skin	Core
A-0.75 in.	Normalize	312	311	309	302
A-0.75 in.	Oil-quench	560	490	514	477
B-1.25 in.	Normalize	307	308	297	302
B-1.25 in.	Oil-quench	465	400	465	363
C-2.25 in.	Normalize	326	321	332	311
C-2.25 in.	Oil-quench	470	432	461	...
D-3.50 in.	Normalize	265	251	260	246
D-3.50 in.	Oil-quench	320	315	321	321

TABLE 9 EFFECT OF DRAWING TEMPERATURE, SIZE OF SPECIMEN, AND NORMALIZING ON PHYSICAL CHARACTERISTICS OF B14 BOLTING STEEL

NORMALIZED

MARK	DRAW TEMP. °F	TENSILE STRENGTH P.S.I.	YIELD STRENGTH P.S.I.	PROP. LIMIT P.S.I.	ELONG. IN 2" %	REDUCTION OF AREA %	CHARPY IMPACT FT. LBS.	HARDNESS VPN/30
				0.75" Diameter				
A	1000	139,300	115,800	110,000	17.5	50.0	22.5	31[illegible], 312
A	1100	144,000			17.5	54.3	23.5	331, 315
A	1200	139,300	121,000	118,800	18.0	54.5	22.0	298, 315
A	1200	139,400	121,800	113,200	18.0	57.0		304
E	1200	140,700	121,200	117,500	21.0	58.1	21.0	301, 308
382 f	1200	142,700	125,600	111,500	19.5	56.4	26.5,17.0	315
382 c	1200	143,900	123,300	110,200	17.5	55.3	14.5	312
E	1300	117,800	99,000	98,500	23.0	58.0	40.0	252, 266
				1.25" Diameter				
B	1000	135,000	111,000	101,500	19.0	56.0	20.5	275, 307
B	1100	137,000	111,000	104,000	21.0	57.0	19.0	301, 302
B	1200	138,000	117,800	106,000	20.0	57.0	19.5,19.0	294, 290
F	1200	138,000	117,500	110,000	19.0	56.8		302
B	1300						31.0	275
F	1300	117,000	96,800	98,500	20.5	59.7	35.5	257, 264
				2.25" Diameter				
C	1000	152,500	123,500	108,500	17.0	49.0	12.0	315, 315
C	1000	152,500	121,200	100,000	17.7	49.3	12.5	321, 326
G	1000	140,000	111,000	92,500	18.6	54.0	14.5	293, 312
C	1100	153,000	125,000	102,500	17.0	48.0	13.0	323, 329
C	1100	156,200	127,400	107,400	19.4	52.5	13.0	312, 342
G	1100	144,100	117,000	102,500	20.0	54.3	16.0	304, 308
C	1200	143,500	120,000	90,000	16.0	53.0	15.5	323, 312
C	1200	151,000	125,500	115,500	18.7	51.0	18.0	326, 315
G	1200	138,300	116,000	105,000	19.2	56.1	18.5	305, 308
C	1300	123,400	99,000	89,000	20.0	59.1	27.5	264, 285
G	1300						30.0	274
				3.50" Diameter				
DO	1000	123,000	84,000	72,500	23.0	59.0	20.5	253, 278
DM	1000	121,500	82,500	76,000	21.5	57.0	24.0	247, 251
DO	1100	126,000	88,000	65,000	21.0	60.0	22.0,23.5	262, 268
DM	1100	121,000	85,000	77,500	23.0	58.0	20.5,23.0	254, 244
DO	1200	124,500	85,000	85,000	21.5	60.0	21.0	251, 281
DO	1200	120,000	80,000	82,500	22.0	59.5	23.5	246, 243
DM	1200	119,000	80,000	77,500	22.0	58.0	25.0	24[illegible], 230
DC	1200	114,500	75,500	78,000	22.0	59.0		236
DO	1300						26.0	239, 242

TABLE 10 EFFECT OF DRAWING TEMPERATURE, SIZE OF SPECIMEN, AND OIL QUENCH ON PHYSICAL CHARACTERISTICS OF B14 BOLTING STEEL

OIL QUENCHED

MARK	DRAW TEMP. °F.	TENSILE STRENGTH P.S.I.	YIELD STRENGTH P.S.I.	PROP. LIMIT P.S.I.	ELONG. IN 2" %	REDUCTION OF AREA %	CHARPY IMPACT FT. LBS.	HARDNESS VPN/30
				0.75" Diameter				
A	1000	198,000	180,000	172,000	13.2	46.0	17.0	432, 434
A	1000	197,700	187,500	152,500	15.6	49.5		422
E	1000	200,000	188,500	175,000	12.3	46.6	19.0	420, 404
A	1100	197,000	182,000	152,000			19.5	418, 429
A	1100	196,000	185,000	146,500	15.6	49.8		415
E	1100	190,100	180,000	170,000	15.2	53.0	21.5	409, 402
A	1200	171,000	166,000	150,000	17.5	55.7	28.0	357, 366
A	1200	162,200	156,800	152,500	17.7	56.4		356
E	1200	152,700	145,000	144,000	19.0	58.5	31.5	328, 341
382 f	1200	139,100	131,400	130,650	19.7	61.4	36.0,39.0	318
E	1300	122,300	112,500	110,000	24.2	65.3	43.0	268, 276
				1.25" Diameter				
B	1000	161,500	139,000	110,500	16.0	52.0	16.0	346, 418
B	1000	184,400	170,000	137,000	12.8	50.0		406
F	1000	187,000	173,000	135,000	14.1	50.4	18.0	404, 422
B	1100	171,000	155,000	115,000	13.0	52.0	20.5	396, 366
B	1100	177,000	160,400	127,700	13.5	51.0		385
F	1100	189,000	177,500	150,000	15.5	52.0	19.5	415, 415
B	1200	155,500	145,000	128,500	17.0	57.0	35.5	359, 361
B	1200	145,800	130,000	117,500	18.4	60.0	35.5	317, 346
F	1200	151,200	143,000	135,000	17.8	58.8	36.5,35.0	333, 343
F	1300	124,500	110,500	102,500	21.3	65.9	46.0	269, 280
				2.25" Diameter				
C	1000	206,500	191,000	131,500	13.0	44.0	15.0	432, 422
C	1000	203,200	187,000	135,000	13.8	46.8	13.5	427, 422
G	1000	161,500	134,000	106,000	17.0	51.5	17.5	354, 355
C	1100	196,500	182,500	138,500	15.0	47.0	16.0	413, 432
C	1100	198,100	186,000	135,000	14.2	47.3	17.0	411, 411
G	1100	161,800	159,000	112,000	18.0	56.6	18.0	351, 357
C	1200	167,500	158,500	145,000	18.0	54.0	28.0	353, 348
C	1200	161,500	152,000	135,000	17.0	54.8	31.0	351, 346
G	1200	148,000	129,000	115,000	19.0	59.2	32.5	308, 329
C	1300	124,000	108,000	107,000	22.1	63.2	41.5	263, 278
G	1300						42.5	272
				3.50" Diameter				
DO	1000	145,000	122,500	110,000	18.0	58.0	23.5	312, 309
DM	1000	146,000	120,000	90,000	18.0	54.0	20.0	307, 328
DO	1100	147,200	125,000	107,500	20.0	59.0	21.0,22.0	325, 329
DM	1100	149,000	123,500	107,500	20.0	54.0	19.0,19.5	318, 322
DO	1200	141,000	122,500	92,500	20.5	58.0	25.0	305, 307
DO	1200	139,500	119,500	107,500	20.0	56.5	28.5	297, 318
DM	1200	139,000	120,000	97,500	20.0	58.0	24.5	315, 311
DC	1200	137,500	120,000	107,500	20.0	58.0		308
DO	1300						40.5	262

It has been interesting to correlate the author's information on B14 steel with other data and to study the effect of size of section upon the physical properties obtained both from the air-cool and oil-quench treatments. Particularly, in the case of air cool, where proper results are dependent upon exceeding a critical cooling rate, is it necessary to consider carefully the effect of size of section. An investigation has been made at the Crane re-

search laboratories upon the effect of drawing temperature and size of specimen for both oil quench and air cool. Specimens of $^3/_4$ in., $1^1/_4$ in., $2^1/_4$ in., and $3^1/_2$ in. diam were investigated for tensile properties, hardness, and Charpy impact after 1-hr draw at 1000, 1100, 1200, and 1300 F, one set of specimens being cooled in air from 1675 F, the other quenched in oil from 1550 F. Test specimens were located in the sections as illustrated in Fig. 28 of this discussion. The chemical analyses of the materials used are given in Table 7.

The hardness of some of the samples in the quenched condition was recorded and a tabulation of these figures is given in Table 8.

The results of the tests are given in Tables 9 and 10.

In attempting a comparison of Crane results with the author's, a rather notable difference in material investigated becomes apparent. Whereas, the author's bars all had carbon contents of either 0.45 or 0.46 per cent, all Crane material with the exception of analysis No. 259,393 had a carbon content covering a range of 0.35 to 0.39 per cent. Fig. 29, which summarizes both G.E. and Crane results for Charpy impact strength as a function of diameter of bar, reveals notable dips in the general trends of the curves at G.E. points, these dips are probably attributable to the higher carbon content. This observation is substantiated by

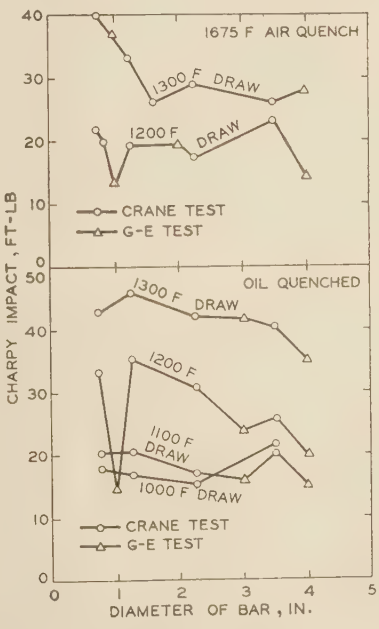


FIG. 29 SUMMARY OF AUTHOR'S AND CRANE TESTS FOR CHARPY IMPACT STRENGTH AS A FUNCTION OF BAR DIAMETER

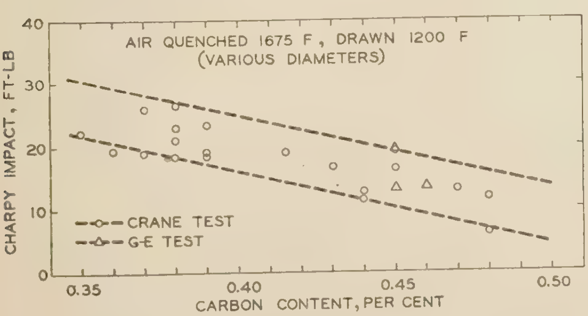


FIG. 30 EFFECT OF CARBON CONTENT ON IMPACT STRENGTH OF B14 STEEL

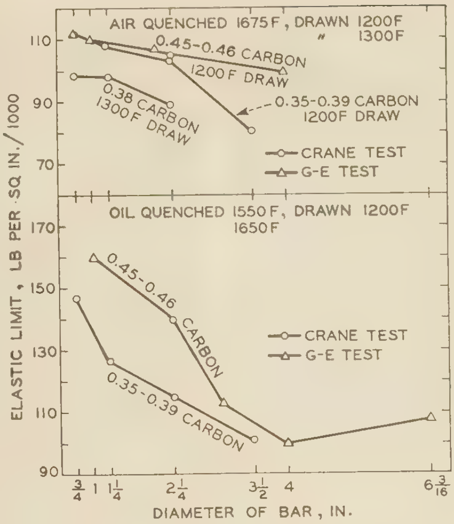


FIG. 31 ELASTIC LIMIT OF B14 STEEL

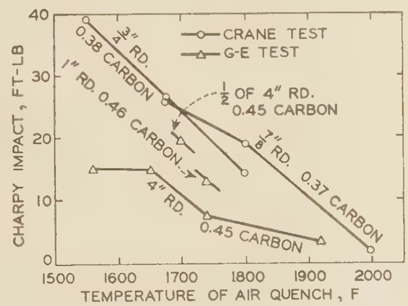
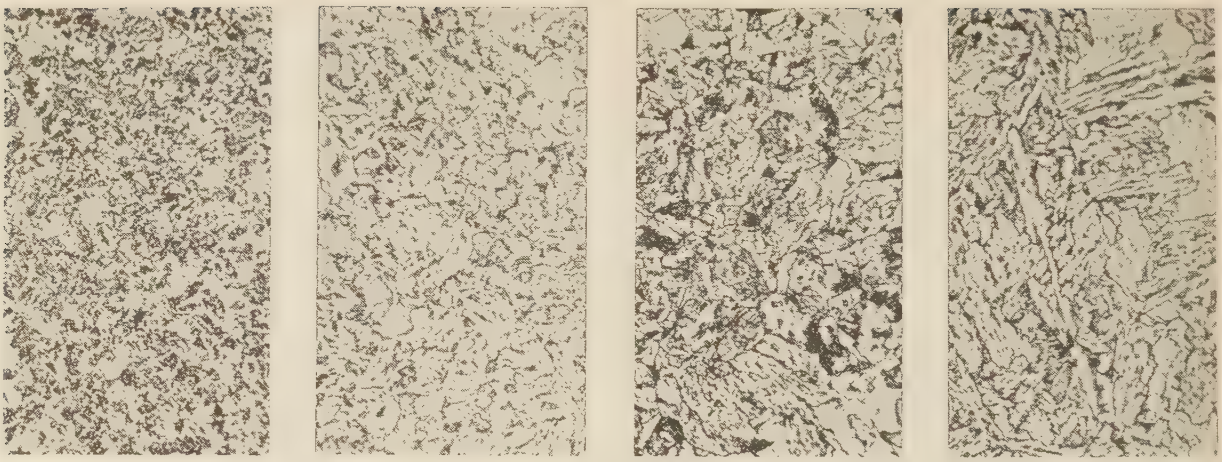


FIG. 32 B14 STEEL, DRAWN AT 1200 F AFTER AIR-COOLING FROM CRITICAL RANGE

Fig. 30, in which a number of Charpy impact results are plotted as a function of carbon content. In Fig. 30, it may be observed that 1675 F air quench and 1200 F draw show a minimum Charpy value of 20 ft-lb for 0.37 per cent carbon, while 0.45 per cent carbon shows a minimum Charpy of 10 ft-lb.

That 0.37 per cent carbon is sufficiently high to obtain the desired class C elastic strength with air-cooling treatment in diameters up to $2^1/_2$ in. is shown by the analysis of elastic-limit data in Fig. 31. Only in diameters as large as $3^1/_2$ and 4 in. does 0.45 per cent carbon appear warranted, if the purpose is to retain elastic strength. Experience has shown that no difficulties, due to insufficient impact strength, are encountered if the carbon content of B14 steel is kept in the range of 0.35 to 0.4 per cent. The wide range of 0.35 to 0.5 per cent in Specification A193 was so set to permit the selection of a carbon content compatible with the air-hardening character of the various diameters of bars. However, since it is clear that nothing useful is gained by high carbon in sizes up to $2^1/_2$ in. diam, and that the Charpy impact suffers so markedly from high carbon, care must be taken to select the proper carbon range within the specification range.

Fig. 2 of the paper shows that the temperature of quench exerts an important influence on impact strength. The importance of this factor is even greater than might be concluded from Fig. 2, if we consider the data available for material having a 1200 F draw. In Fig. 32 of this discussion, G.E. data are given for 0.45 and 0.46 per cent carbon B14 steel together with Crane data for 0.37 and 0.38 per cent carbon B14 steel. While the 4-in-



(a) (b) (c) (d)

Fig. 33 Microstructure of B14 Steel, Air-Cooled and Drawn at 1200 F; ×500

[(a) Air-cooled from 1550 F, $^3/_4$-in. rod, 0.38 per cent carbon, Charpy 39 ft-lb, A.S.T.M. grain 9; (b) air-cooled from 1675 F, $^7/_8$-in. rod, 0.37 per cent carbon, Charpy 26 ft-lb, A.S.T.M. grain 9; (c) air-cooled from 1800 F, $^7/_8$-in. rod, 0.37 per cent carbon, Charpy 19 ft-lb, A.S.T.M. grain 7; (d) air-cooled from 2000 F, $^7/_8$-in. rod, 0.37 per cent carbon, Charpy 2 ft-lb, A.S.T.M. grain 5.]

diam 0.45 per cent carbon ranges from 15 ft-lb Charpy for 1560 F to 3 ft-lb for 1920 F air cool, the lower-carbon Crane material in $^3/_4$ in. and $^7/_8$ in. diam ranges from 40 ft-lb Charpy for 1550 F air cool to 2 ft-lb for 2000 F air cool. This comparison again indicates that, for almost any air-quenching practice, important gains in Charpy impact are to be expected by limiting carbon content to a range between 0.35 and 0.4 per cent.

In Fig. 15 of the paper is shown the microstructure of steel air-cooled from 1700 F and drawn at 1180 F. However, in order to gain perspective on the effect of varying the air-cooling temperature on the microstructure and the physical properties, let us consider the photomicrographs of B14 steel with 0.37 to 0.38 per cent carbon at 500 diam, Fig. 33 of this discussion, representing the $^3/_4$-in. and $^7/_8$-in-diam air-cooled from 1550 F, 1675 F, 1800 F, and 2000 F, respectively, and all drawn at 1200 F. As the air-cooling temperature increases, the austenitic grain size seems progressively to increase from about A.S.T.M. 9 for 1550 F and 1675 F to A.S.T.M. 7 for 1800 F and finally to A.S.T.M. 5 for 2000 F. Moreover, a definite tendency toward Widmanstätten structure has developed by heating to 1800 F and above, not apparent for 1675 F and below. These data would seem to suggest that between 1700 F and 1800 F there is an austenite grain-coarsening effect injurious to the impact strength and high-temperature rupture properties. Rupture tests were made upon 0.505-in-diam bars of the 0.38 per cent carbon, representing both fine and coarse air-cooling structures, by loading to 30,000 psi at 1000 F with the following results:

Heat-treatment	Time to fracture 3000 psi 1000 F, hr	Total elongation, per cent
Air cool 1675 F, draw 1200 F........	2400	20.0
Air cool 1800 F, draw 1200 F........	830	0.5

The author's data for 0.45 per cent carbon steel having 1700 F air cool, 1180 F draw, tested to rupture at 1000 F for a similar time period show better than 10 per cent elongation. Thus, it appears that brittle rupture of B14 steel in the air-cooled condition is only a hazard when heat-treating temperatures above the austenite-coarsening temperatures are used. There is good reason to believe that often times, where failure to achieve good results in the use of this and other alloy steels has been experienced, the failure may be attributable to the application of too high normalizing temperatures.

While through careful control of the heat-treatment and composition of B14, thoroughly satisfactory properties at ordinary and high temperature can be attained, using a 1200 F draw temperature, there may be purposes for which greater toughness is desirable. As shown by Fig. 29 of this discussion, a 1550 F oil quench, followed by a 1200 F draw, gives some gain in the impact strength obtained through air-cooling treatment, but at a great sacrifice of creep and relaxation resistance. By retaining the air quench from 1675 F and increasing the draw temperature to

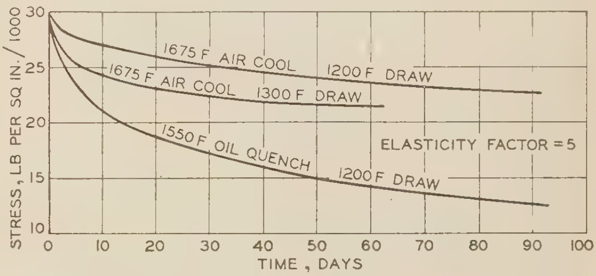


Fig. 34 Curves of Relaxation for B14 Steel

1300 F, the Charpy impact strength appears fully as good as for oil quench and 1200 F draw. Although the room-temperature elastic strength is somewhat reduced by the 1300 F draw (refer to Fig. 31), it is still high enough in sizes up to $2^1/_2$ in. diam to suffice for many bolting applications more than meeting A.S.T.M. A96 class B requirements. Fig. 34 illustrates the tremendous difference in relaxation time for 0.38 per cent carbon B14 steel at 932 F between 1675 F air cool, 1200 F draw treatment, and 1550 F oil quench, 1200 F draw. While 1675 F air cool, 1300 F draw results in loss of relaxation resistance, it is clear that a distinct advantage over the oil-quenching treatment is maintained.

It is reassuring to note in Table 3 of the paper that creep specimens of B14 steel, tested at 950 and 1000 F, do not suffer appreciable loss of impact strength through high-temperature exposure, provided the original value exceeds 20 ft-lb Charpy. Although the results shown by the author which qualify were obtained by oil-quenching treatment, air-cooled B14, with carbon content not exceeding 0.4 per cent carbon, not only shows good impact strength as treated, but also after prolonged high-tem-

perature exposure, as attested by the following results on analysis No. 258,440, $^3/_4$-in-diam, 0.38 per cent carbon, air-cooled from 1675 F, 1200 F draw:

Charpy impact, as heat-treated..................	26.5 ft-lb
Charpy impact, after 2000 hr at 900 F............	27.0 ft-lb
Charpy impact, after 1000 hr at 1100 F...........	33.0 ft-lb

Arthur McCutchan.[9] Some 20 years ago, investigations of the tendency of bolts to become brittle in service at temperatures around 600 F led to the substitution of alloy steels for the mild-carbon-steel, wrought-iron, and screw-stock bolts which had formerly been used. In the intervening years, investigations by English metallurgists[10] showed nickel-chromium bolts (3 to 4 per cent nickel, 0.75 per cent chromium) to be particularly susceptible to embrittlement, as determined by reduction in impact strength. However, until the recent increases in power-plant operating temperatures to around 900 F, breakage of alloy-steel-bolt studs was of infrequent occurrence in this country.

During the last year, cases of bolt breakage or low impact values after service have been reported for the following bolting materials listed in A.S.T.M. Specification A193:

Grade B4, nickel-chromium-molybdenum, SAE 4340
Grade B7, chromium-molybdenum, SAE 4140
Grade B11, tungsten-chromium-vanadium
Grade B12, nickel-chromium, SAE 3140
Grade B13, tungsten-molybdenum-chromium
Grade B14, chromium-molybdenum-vanadium.

Whether inherent lack of structural stability of the alloy bolting materials at these higher temperatures or the more severe stress conditions imposed is responsible for this increase in bolt breakage is open to question. The additional stress imposed on bolts because of difference in temperature between the body of the flange and the bolts during warming up periods is, of course, greater with a 950 F line temperature than with 750 F. This is true because of (1) the higher temperature gradient established between the inner flange mass and the bolts; and (2) the greater rigidity of the flanges necessary for the higher temperature.

While agreeing that notch impact values should be as high as can be obtained consistent with other properties, the writer has observed cases where bolts with Charpy V-notch values of only 8 to 12 ft-lb, but with extremely high tensile, yield, and creep strengths, have given the best service in keeping certain experimental joints tight. Incidentally, had the author used the A.S.T.M. standard V-notch Charpy specimen rather than the keyhole Charpy and the special deep-notched specimen, his results would have been more directly comparable with those of other investigators.

The difference in impact values found for the specimens, represented by items 837 and 838 in the author's Table 3, illustrates the difficulty of drawing conclusions from a limited number of impact tests. According to Table 2, the diameter of stock, composition, and heat-treatment of these two items were identical, yet item 837 showed a drop in Charpy keyhole impact from 12.8 to 3.79 ft-lb after 3870 hr at 932 F while item 838 showed an increase from 13.8 to 15.1 ft-lb. It occurs to the writer that the value of 3.79 ft-lb might be an error in decimal point since this appears to be the only impact value reported to the hundredths place. Because of discordance in usual impact results, reporting values even to tenths of foot pounds implies an accuracy of reproducing results that is of doubtful justification.

The question of bolting performance is receiving increased attention at this time and the author's correlation of impact and creep properties for this one bolting material should stimulate further study of this and other types. The free interchange of such results is of great assistance in the selection of suitable bolting materials.

[9] Engineer, Engineering Division, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

[10] "The Effect of Time and Temperature on the Embrittlement of Steels," by A. M. McKay and R. N. Arnold, *Engineering*, vol. 143, Dec. 15, 1933, p. 647; also, "Embrittlement of Steels at High Temperatures," by H. A. Dickie, *Engineering*, vol. 143, Aug. 4, 1933, p. 108.

J. S. Worth.[11] In presenting the results of so many high-temperature tests on a single bolting steel, the author has provided at least a partial answer to some of the most important questions arising from the use of high-temperature steel. Although the data are so varied in character that generalizing is not possible, the following more or less related trends may be discerned.

A.S.T.M. Specification A-193 requires that the tempering temperature exceed the nominal operating temperature by at least 100 F. This was written in to insure stability of structure and properties of the steel during its service life. The minimum differential was set at 100 F because experience with a number of high-temperature steels showed it to be adequate.

The author's study of the chromium-vanadium-molybdenum steel indicates that a 100 F spread may not make it stable. Photomicrographs and impact tests of items 863 and 864 reveal an unmistakable change occurring in only 2500 hr at 1000 F, although the steel had been drawn at 1180 F, 180 F above the test temperature. Does this mean that the minimum differential of 100 F may be too small for most bolting steels, or that this steel is more difficult to stabilize, or merely that it was not held long enough at the tempering temperature?

We believe that the question will bear further study because of its importance. There is slight agreement as to how much static tensile strength, notch impact resistance, rupture, and creep strength are necessary in high-temperature steels, but the desirability of maintaining the original properties of a material throughout its term of service cannot be questioned.

Although for stability this steel may require tempering at a temperature very substantially over the service temperature, it is capable of maintaining satisfactory room-temperature strength when so treated. In other words, the tensile and impact properties of the steel can be made highly stable with the proper heat-treatment.

This fact should be borne in mind by the user, as the present requirements of Specification A193 will not necessarily insure the application of such treatment.

According to Figs. 20 and 21 of the paper, oil quenching tends to stabilize the material in another respect. It postpones the transition from ductile to brittle fracture in the rupture test. The author states that, whereas, brittle or intercrystalline failure appeared in the air-quenched steel after only 310 hr at 1000 F, it was not obtained in the oil-quenched material for 3400 hr. Whether a steel which becomes brittle after even 3400 hr may be considered safe for most engineering purposes is in itself an important question, but at least it may be said that the effect of oil quenching is in the direction of greater stability. In spite of the foregoing, we do not mean to conclude that the normalizing treatment is less suitable than oil quenching for all high-temperature bolting steels since eventually some steels having a very different behavior may be developed. However, for the chromium-vanadium-molybdenum steel considered in the paper, we are in accord with the conclusion that oil-quenching treatments are the safest to employ.

[11] Assistant Metallurgical Engineer, Bethlehem Steel Company, Bethlehem, Pa.

Author's Closure

The author appreciates the valuable addition of test data and comments, based on experience with this bolting material, presented by the several contributors.

Referring to the comments of Messrs. A. J. Herzig and R. L. Wilson, the author agrees that control of composition and cooling rate from the quench would provide uniform room-temperature physical properties and creep and rupture strengths for different sizes of stock. Admitting that there is merit in such a procedure, the question remains as to how far it would be justifiable to go in using regularly a variety of compositions and treatments, in order to secure a uniformity of final result, the actual improvement of value of which might not be great enough to justify the complication.

The author wonders if too great importance has not been attached in the past to the elastic properties at room temperature which are coming to be considered as less important than formerly for materials which have to operate at high temperature and

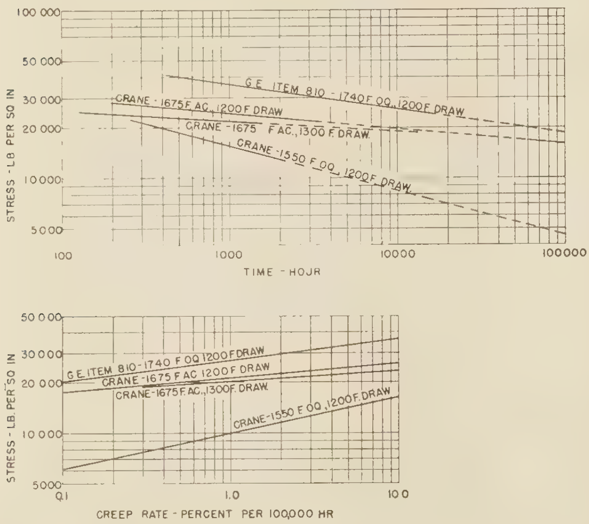


Fig. 35 Comparison of Bolting Performance With Elasticity Factor of 5, Upper Diagram, and Creep Strength, Lower Diagram, for Materials Shown in Fig. 9 and Fig. 34 at 932 F

where care may have been taken in design to avoid bad differentials.

Thus, in order to meet the room-temperature physical-test requirements of A.S.T.M. specifications, particularly the yield strength, the range of heat-treating possibilities is at present narrowed down. Perhaps the specification should be revised for materials which are to be used for 900 to 1000 F application to permit somewhat lower elastic limit with resulting higher creep strength.

Mr. Kanter points out the fact that lower carbon content broadens the range of heat-treatment for high Charpy impact values, which is in agreement with the author's experience in tests on this composition.

In reference to Mr. Kanter's Fig. 34, it will be noted that the oil-quenched specimen is quite inferior in comparison to the two normalized specimens; however, this was oil-quenched from 1550 F and drawn at 1200 F, both of which were too low to develop the optimum creep strength for the oil-quenched condition. This material was only $^3/_4$ in. diam. The only tests of small-size material, described in the paper, were items 810, 837, and 838 on 1 in. diam. These were tested at 932 F, the same temperature as used by Mr. Kanter, and item 810 was oil-quenched from 1740 F and drawn at 1200 F. Plotting item 810, which is shown in Fig. 9 of the paper, and correcting it to the same elasticity factor of 5, to be comparable with Mr. Kanter's data in his Fig. 34, it is evident that, with proper oil-quenched treatment, in this particular case, it will be fully as good as the normalized material. This is shown on the combined plot of Fig. 34 and Fig. 9 of the paper in Fig. 35 of this closure.

One matter of importance to be developed by the discussion is the size of stock and the relation between this and the type of heat-treatment. Thus, when all is said and done, there is not so much difference between the normalizing of small-diameter material and the oil-quenching of large-size stock.

In reply to Mr. McCutchan's inquiry concerning the Charpy value of 3.79 ft-lb for item 837 in Table 2 of the paper, the decimal point is correct as shown. The figure in question was the result of an average, but even so, the author admits there is no justification for quoting impact strengths to hundredths.

When the tests covered by this paper were first instituted, no thought was given to the possibility of publication, and the type of V-notch impact specimen was selected with the idea of producing a notch which would be the nearest approach to an actual bolt thread. When it was decided to publish the test results, these sharp V-notch impact data were not excluded. It is a measure of sensitiveness of materials to sharp notch impact and is a more severe test than any of the generally accepted impact tests.

Mr. Worth agrees with the author as to the desirability of the oil-quenching treatment. In addition, he has pointed out the necessity of drawing this material for maximum stability at a temperature considerably above the operating temperature. With this particular steel a high draw is necessary in stabilizing the carbides. In the oil-quenched condition, the ductility of fracture in the rupture test increases with the increase of draw temperature as shown in tests subsequent to the preparation of this paper.

In conclusion there is no escape from the fact that a satisfactory bolting material must represent a compromise among a number of qualities.

After considering a proper balance of desirable room-temperature physical properties, creep and rupture strengths, structural stability at high temperature, and insurance against failure in tightening the bolts, the following heat-treatment is recommended for material of composition similar to that reported in this paper: 1650 F, oil-quenched; 1250 F, air-cooled.

Continuous Heat-Balance Control of Boiler-Room Operation

By BENJAMIN S. MURPHY,[1] BROOKLYN, N. Y.

The author has attempted to bring out the different factors entering into boiler-room-operation control and to show that all of these factors must be synchronized before any comprehensive plan may be formulated. These factors include such apparently unrelated items as scheduling boilers for cleaning, turbine outages, and personnel, as well as incremental boiler loading and automatic control. The methods used at the Hudson Avenue Station of the Brooklyn Edison Company, Inc., constitute the basis for the presentation.

OUR entire social system has come to depend more and more upon an unlimited, continuous, and low-cost supply of electrical energy from centralized distributing centers. Due to geographical location or political reasons, a few such centers utilize the energy from falling water, but the greater portion must use the stored heat in fuel. To convert this heat to electricity, we burn this fuel under a boiler.

The boiler thus becomes the foundation for our energy supply and, to meet the requirements mentioned, we must insure that the boilers involved will (1) develop the maximum expected output; (2) give this output whenever called upon, and (3) produce it economically.

The designing engineers and the manufacturers provide the operator with a rather intricate mechanism. However, the operator cannot shovel in a scoop of coal at one end and take out an equivalent kilowatt of steam from the other, as often as required and as economically, without contributing talent of his own. It is this talent which we call "boiler control."

By boiler control it is not meant that regulators and devices can be installed to accomplish the desired objectives. It is not intended to detract in the slightest from the effectiveness of automatic control, which is one of our best boilerhouse tools, but rather to emphasize the fact that boiler control depends upon many factors which must all be synchronized. These may be outlined as (1) planning, (2) operating, and (3) checking. There is but a slight difference between the last two items, since an adequate system of checking must be in effect, at all times or, by the time inefficient operation is discovered, material harm may have been done.

Planning

The first step to be undertaken is the development of a long-range plan for the year. Of primary importance in this plan is to know how many boilers we must have either on the line or ready to go on the line when required. Unfortunately, due to first cost, operators never have as many boilers as they would like to have, and so must make the most of the boilers which are provided. A typical yearly boiler schedule is shown in Fig. 1.

Load. First we will estimate the maximum, nonemergency peak for each day of the year and plot this in Fig. 1 as the "expected maximum peak." Now, believing we know the load to expect, the next step is to find the number of boilers necessary to deliver it.

Number of Boilers. If all the boilers are of the same size, the problem is fairly simple. However, since in most of the older stations the boiler size has increased with each addition, it becomes more complex to determine the number required.

At the author's station, not only are the boilers of different sizes, but they operate at different pressures; twelve of the same size at 300 psi, and twenty of three different sizes at 400 psi. The turbines served are three 50,000-kw units operating at 300 psi, and five, from 80,000 to 160,000 kw, at 400 psi. Also, by using reducing valves and desuperheaters, the 300- and 400-psi systems are interconnected, so to some extent the 300-psi turbines may be supplied from the 400-psi boilers.

To overcome the difficulty due to boiler size in our calculations, we reduce them to a unit size or "equivalent boiler." This is done by calling the capacity of the base boiler 10 Mw, or that of our smallest (300-psi) units. Then, on peak-load operation, we have the boilers rated as in Table 1. These are the boiler ratings which reasonably may be expected for normal heavy peak conditions (1) with the average grade of coal supplied, (2) with the amount of overfire liquid fuel that we may have, (3) with reasonable boiler economy, and (4) without making an undue amount of objectionable stack discharge. They are low enough so that, for any actual emergency, such as the loss of a boiler, we may increase the individual steam outputs of the remaining units to compensate for the shortage.

As the base-boiler rating is 10 Mw, the expected load divided by 10 gives the number of equivalent boilers required on any day. So using the "expected-maximum-peak" curve of Fig. 1 and the right-hand scale, we have the number of boilers that will be necessary.

Boiler Scheduling for the Year. The availability factor of the modern boiler is improving constantly, and yet, boilers must come off the line to be cleaned and repaired at fairly frequent intervals; this is especially true for stokers. So, after determining how many boilers we must have at any time, we must fit the individual boilers into their outage periods and make out the schedule for each unit.

The 300-psi turbines and boilers are the least economical and so are used as little as possible. However, when some of the 400-psi turbines are off for overhaul or when the station peak is above 600 Mw, some 300-psi turbines must be used. If the 300-psi steam demand is not too great, the 400/300-psi connection is sufficient, but there are times when 300-psi boilers must be used to back this up. For this reason we will plot, in Fig. 1, the times of scheduled turbine overhauls.

Our boilers are all stoker-fired and experience has proved that more frequent but shorter outages keep them in better condition, so we have abandoned the old practice of many "externals" and one long "internal" outage each year. This has been replaced by scheduling all overhauls for 2 weeks. The length of time between these outages is set at 14 weeks for the regular steaming boilers. Any minor stoker repairs are made during week ends or nights when the boiler may be spared. This schedule is

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Contributed by the Fuels Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

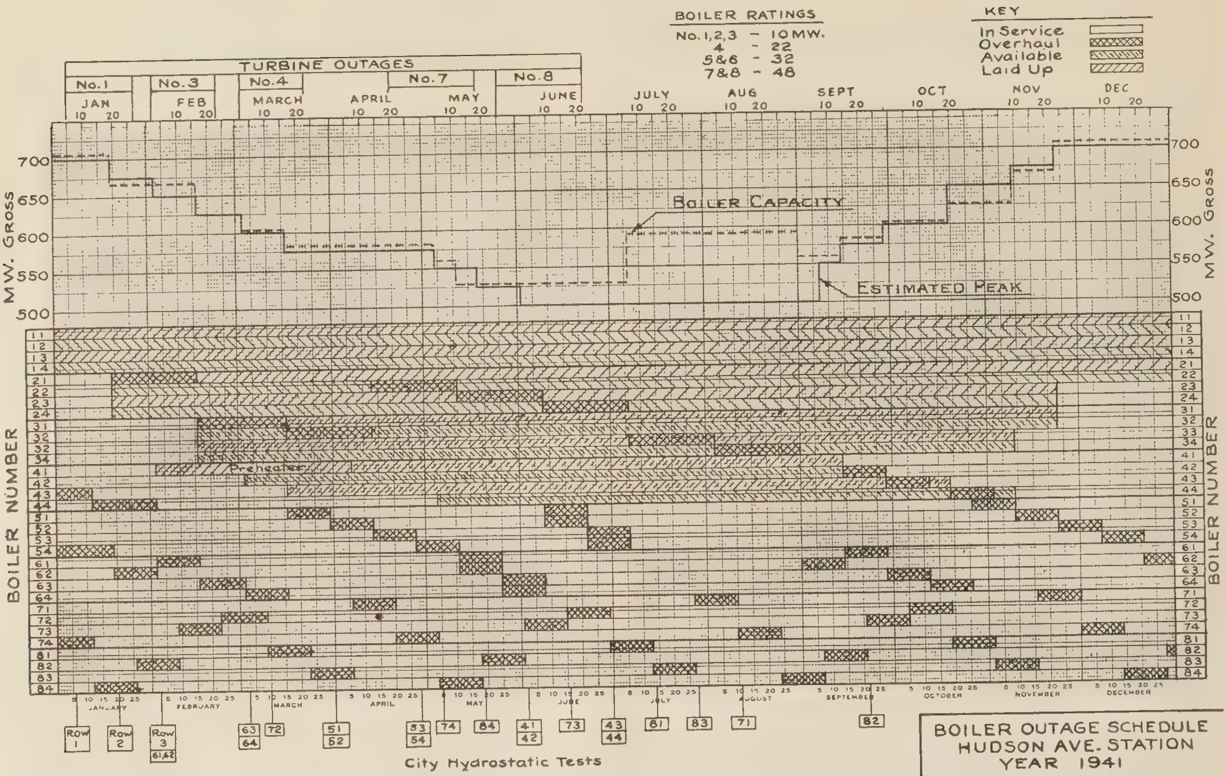

Fig. 1 Yearly Boiler Schedule

maintained unless something quite out of the ordinary occurs, then we get back on it as soon as possible, shortening repair periods, if necessary.

We plan a boiler-maintenance crew of sufficient size to handle three boilers at one time, together with any routine work which may develop. With this crew our object is to have under repair throughout the entire year one boiler from row 7 or 8 (largest), one from row 5 or 6 (next largest), and one other boiler.

This leaves seven boilers of rows 7 and 8, and seven of rows 5 and 6 available or a total of 56 equivalent boilers. Any difference in boilers required is made up first from row 4 (also 400 psi), and next from rows 3 and 2 (300 psi).

The outage periods of rows 5, 6, 7, and 8 are now plotted so that they will fall at the specified times for insurance and city inspections and tests. The status of the other boilers is determined by demand. The specified overhaul periods and the outage, available, laid-up, and steaming periods for the remaining boilers are plotted on the graphical schedule.

The total steaming capacity is calculated, and this line, the dotted one following the solid line, is drawn in Fig. 1, completing the yearly schedule. From this schedule, the chief boiler-room engineer makes out his detailed schedules for work to be done at each outage.

Boiler Scheduling Daily. Each day a "turbogenerator and boiler schedule" is issued covering the period from 4 p.m. that day to 4 p.m. the next day. This schedule gives the number of boilers (and turbines) available for that day's peak, the number to use, and the order in which the turbines are to be taken off during the night and go back on the next morning. This gives the boilerhouse crew the information necessary to bring up their boiler loadings or bring them down to banking to suit the load demand. This is especially so for the 300-psi units on bank except when some 300-psi turbines are being used and are brought up with the rolling of the generator, saving considerable fuel.

The expected peak at Hudson Avenue, including Gold Street, is also given, the latter since all steam to that station is supplied from the Hudson Avenue boilers. The expected boiler capacity as given to the system operator is listed, together with any special notes on hold-offs, etc.

The boiler data required to compile this sheet are taken from the yearly schedule, modified if necessary. All of this scheduling is done by rows not by specific boiler numbers.

TABLE 1 PEAK BOILER RATINGS

Boiler row	Equivalent boiler	Capacity each, Mw
1, 2, and 3	1.0	10
4	2.2	22
5 and 6	3.2	32
7 and 8	4.8	48

TABLE 2 BOILER-LOADING SCHEDULE
(Loading in Pounds of Steam per Hour)

Row number: 4	5 and 6	7 and 8
100000	100000	160000
100000	110000	180000
110000	120000	200000
120000	140000	220000
130000	150000	240000
140000	170000	260000
160000	180000	280000
170000	190000	300000
180000[a]	210000	320000
180000	230000	340000
	Overfire air[c]	
190000	250000	350000
		Overfire liquid fuel[e]
190000	260000	360000
190000	270000	380000
200000	280000	400000
200000	280000	420000
210000	300000	440000
220000[b]	320000[d]	460000
220000	320000	480000
220000	320000	500000[f]

Note: For letter notes see text.

BOILER LOG
HUDSON AVE. STATION
PRODUCTION DEPARTMENT

Monday DAY 9/26/40 DATE

TIME 12		1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12
STA. LOAD MW		265 240	230 200	195 180	180 155	155 160	170 165	170 200	205 260	370 400	420 420	430 430	430 420	380 400	420 420	420 420	420 420	430 410	410 400	380 380	400 460	510 500	480 450	380 350	320 300

BOILER RECORD - 300 LB

BOILER NUMBER	BOILER RATING MW	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12
11	11	L							L								L								L
12	11	L							L								L								L
13	11	L							L								L								L
14	11	L							L								L								L
21	11	L							L								L								L
22	11	L							L								L								L
23	11	L							L								L								L
24	11	L							L								L								L
31	11	A							A								A								A
32	11	A							A								A								A
33	11	A							A								A								A
34	11	A							A								A								A
TOTAL																									

BOILER RECORD - 400 LB.

BOILER NUMBER	BOILER RATING MW	1	2	3	4	5	6	7	8	9	10	11	12	1	2	3	4	5	6	7	8	9	10	11	12
41	22	R							R								R								R
42	22	A							A								A								A
43	22	A							A								A								A
44	22	A							A								A								A
51	32	A	A	A	70	95	110	115	120	210	245	255	255	235	245	235	235	230	225	230	230	275	265	225	150
52	32	R	R	R	70	105	70	B 35	100	200	230	235	270	270	270	270	265	240	220	215	235	265	265	210	160
53	32	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R
54	32	190	165	80	75	100	115	120	120	180	230	230	235	215	230	225	225	225	225	215	235	280	280	200	180
61	32	50	80	135	135	175	115	60	125	220	245	245	260	240	240	240	250	240	240	210	250	290	280	215	175
62	32	190	75	B 20	125	70	B 30	105	125	215	225	260	260	230	250	245	240	230	B 140	60	250	300	280	215	165
63	32	200	160	120	B	B 25	85	130	145	215	215	250	265	220	240	230	225	230	240	235	270	315	300	220	185
64	32	210	160	140	110	B	50	120	175	220	245	245	255	230	230	225	240	230	240	230	270	310	280	210	180
71	45 2	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R	R
72	45 2	330	270	240	220	210	210	100	180	320	350	360	340	330	355	340	360	320	340	340	345	445	400	330	300
73	45 2	100	B 40	B 30	B 30	B 30	140	205	220	340	350	340	340	335	340	330	330	330	335	335	340	425	400	320	280
74	45 2	300	270	230	200	200	210	210	215	315	345	350	330	320	340	340	325	325	335	320	340	420	380	B 220	B 60
81	45 2	290	250	230	220	270	200	210	220	340	350	350	330	340	340	340	335	330	320	320	340	405	375	315	265
82	45 2	B 40	B 30	B 30	B 20	40	150	210	220	320	330	330	330	320	330	330	335	320	310	330	320	400	365	310	280
83	45 2	290	260	240	230	B 50	B 40	126	200	320	350	355	335	160	320	335	330	340	340	320	325	400	385	330	250
84	45 2	310	250	220	210	190	190	220	200	315	340	340	325	320	315	335	335	335	335	315	340	405	365	320	280
TOTAL		2500	2010	1715	1715	1560	1715	1966	2365	3730	4050	4145	4120	3765	4095	4020	4030	3935	3845	3675	4090	4935	4620	3640	2910
DESUPERHEATER		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0

KEY
ENTER AVERAGE HOURLY STEAM FLOW IN 1000 LBS/HR
WHEN BOILER IS IN SERVICE
A—AVAILABLE
L—LAID UP
R—REPAIR
DB—DEAD BANK (AUTOMATICS CLOSED—NO STEAM BEING GENERATED)
LB—LIVE BANK (AUTOMATICS OPEN BUT NO MEASURABLE STEAM BEING GENERATED)
B—FOLLOWED BY STEAM FLOW (AS B60 OR B70) STEAM BEING GENERATED BUT NO COAL BEING FED.
LF—OVERFIRE LIQUID FUEL FOLLOWED BY 1, 2, OR 3 FOR NUMBER OF BURNERS

NOTE
USE SPACE PROVIDED ON BACK OF LOG FOR REMARKS AS SLAGGING LANCING BLOWING DOWN ETC

ENGINEER IN CHARGE
FROM 12 TO 8 C. Meyl
FROM 8 TO 4 M. Vliegenthart
FROM 4 TO 12 I. Church

FIG. 2 BOILER LOG; FRONT

BOILER No	BLOW DOWN No GLASSES BLOWN: BOILER	SCREEN WALL	BANKING TIME: ON BANK	OFF BANK	STATUS CHANGE TIME: OFFLINE FOR REPAIRS	MADE AVAILABLE	LAID UP	ON LINE	REMARKS
11									
12									
13									
14									
21									Not in service
22									
23									
24									
31									
32									
33									
34									
41									
42									
43									
44									
51					—	12^{01}a	—	3^{15}a	
52					—	3^{00}a	—	3^{25}a	
53									
54									
61									
62			2^{05}a	3^{08}a					Lanced
63			3^{08}a	5^{10}a					"
64			4^{03}a	5^{45}a					"
71									
72									
73		3							
74									
81									
82		2							
83									
84									

OVER FIRE AIR
FANS STARTED 4^{00}p
FANS SHUTDOWN 10^{25}p

OVER FIRE LIQUID FUEL
NUMBER OF BURNER HOURS ~ 21 from 7^{00}p To 10^{30}p = 73.5
18 " " 10^{30}p " 11^{00}p = 9.0
TOTAL = 82.5

FIG. 3 BOILER LOG; BACK

The order in which the turbines are to be placed on or removed from the line depends upon the anticipated loads, their heat rates, and the times on orders from the system operator.

Each day the chief boiler-room engineer schedules the boilers he wishes banked and their sequence, the blowdown based on our laboratory report on concentration, any repairs that are to be made, etc., but always bearing in mind capacity and relative economy.

Boiler-Loading Schedule. The daily schedule, based on the yearly one, gives the total number of boilers to be used, and a boiler-loading schedule, Table 2, gives the distribution of load between the different-sized boilers.

Four boilers were installed with each turbine, in line with it. Each group of four has its own steam header, but all of the headers are tied together. Thus while most of the steam for any turbine comes from its own row, it is not necessary, and a turbine may operate with its boilers shut down, so no particular boiler-turbine combination need be considered.

The boiler loadings have been computed from incremental loadings based on boiler tests and should give the best over-all economy. However, they have been modified to suit other operating conditions.

It is fundamental that we wish to burn the minimum amount of coal but we must limit our operating points to those at which a nuisance is not created, due to stack discharge, even if by so doing more coal is burned. Such points are indicated in Table 2 and explained subsequently.

If the same grade of coal were always available, these points would be well fixed but, since the grade varies, we have two such stop limits for each row and do not depend upon the operator knowing exactly what he is burning, and to limit his load accordingly.

In Table 2, for row 4 we find that, for all coals ordinarily received, we are quite safe at 180,000 lb per hr (note *a*) so, unless necessary, we stop loading at this point and hold at this amount until the other boilers have been brought up, although the

theoretical curve would call for further loading of this row before increasing the load on the others. When the other boilers have been brought up, then row 4 may be increased to 220,000 lb per hr (note *b*) and there we must stop, for though we have taken 280,000 lb from a boiler the coal must be exceptionally well suited or there will be trouble.

The first limiting point (note *c*) for rows 5 and 6 is 230,000 lb per hr. When it is necessary to go higher than this, we use overfire air. This consists of six nozzles in line across the back of the boiler connected to special blowers, discharging air at from 9 to 17 in. wg horizontally into the furnace. These high-velocity air streams striking the gases at right angles break them up, causing considerable turbulence, supplying oxygen where required, and knocking down and holding down a large amount of fly cinders and "popcorn." With our average coal and no overfire air, about 280,000 lb per hr is the nonnuisance limit but, in order to provide a safety margin for poorer coals, we set this at 230,000 lb per hr. With the overfire air, we now increase the everyday peak to 320,000 lb per hr (note *d*), which roughly gives a gain of one boiler in eight. Incidentally, the total quantity of air to the furnace is not increased, the underfire air being reduced in proportion.

For the largest boilers, rows 7 and 8, we are fairly safe at about 380,000 lb per hr, but have set the first limit below this at 350,000 lb per hr (note *e*). For higher loads, overfire liquid fuel, either gas tar or fuel oil, is used. The liquid-fuel installation consists of three burners, in a group on the center line at the back of the boiler, discharging horizontally into the furnace over the fuel bed. By means of this arrangement the coal-burning rate of the stoker is held below the nuisance limit and the additional heat is supplied from the liquid fuel together with its necessary air. At the same time, the introduction of these flames causes some turbulence and mixing of the coal-fire gases, supplies ignition to some coal or carbon particles, and has some tendency to keep down any cinders. The top value for the better coals is now 500,000 lb per hr (note *f*) or, for the eight boilers, the liquid overfire fuel has given the equivalent of ten or a gain of two large boilers.

A study each day of the steam outputs, as shown on the boiler-meter charts, would check the boiler loading, and this is done. However, for convenience and for the psychological effect of self-checking, each hour we have the boiler-control operators enter the hourly outputs of the individual boilers on the boiler log, Fig. 2. This log also gives the status of all boilers and shows any deviation from the yearly schedule.

Operation

Not so many years ago, we operated a boiler with only an indicating pressure gage, a water column, a "banjo," and a fireman to play it. If the latter was "not so strong in the back" as to be "too weak in the head" the results were not bad. We depended upon the man and we still do. Years ago, the late Mr. Smoot often said, "my control is only a tool, it cannot think." So, we must not overlook the fact that we are still dependent upon the skill and knowledge of the boiler operator and, therefore, must supply him with the necessary equipment in order that he may utilize this skill.

Coal Measurements. Our coal is floated alongside in barges (1700 to 1900 tons) or in steamers (5000 to 6000 tons) and hoisted by digging buckets to 5-ton self-propelled cars. Each car on its way to the bunkers passes over and is weighed on a track scale and the weights printed on a tape.

The eight rows of boilers are back to back and in the aisles in front of each row are electric-propelled weigh lorries. These lorries take the coal directly from the bunker gates and discharge it into the stoker hoppers. With this arrangement, it is possible to weigh the coal supplied to each boiler, to each row, and to all boilers.

The total weight of all coal hoisted each day is compared with the shipper's weights. If a check of the coal to each boiler each day is required, the amount of labor involved is excessive so our practice is to use only the daily totals. However, if the consumption becomes irregular, then more detailed checks are made.

The bunkers are calibrated at the first of each calendar week and month and, from these estimates and the shipper's weights, the consumption is calculated. It is desirable that the bunkers be leveled before calibration, but too often operating conditions will not permit and the estimate is made "as is." Ordinarily this is done by eye, but if more irregular than usual, or low, a tape is used.

If the calibration is always made by the same person, the error is reduced but, even by different ones, the probable error is ±200 to 500 tons and this for us would be ±70 Btu on the heat rate for the month and ±250 Btu for a weekly calibration. It is obvious from these errors that too much dependence must not be placed on the weekly figure, though the actual error is as a rule less than that given. The weekly rate is of value, however, because if it is used in conjunction with the weekly lorry weights, it gives the trends, and the average will not be far out of line with the monthly rate.

Whether we like it or not, the coal to be accounted for must be that given on the bill of lading. Some irregularities are found

DATE Aug 2 1940 DAY Friday

HUDSON AVENUE- BOILER ROOM

Item	Watch 12-8	8-4	4-12
Ignition			
Coking			
Volatile			
Fuel Bed			
Clinker	✓	✓	✓
Slag	✓		
Dust			
Dirt			✓
Smoke			
Rating			
Remarks	Soft clinker	Test coal in B 71 and 61 at 9 am.	Test coal in B 61 and 71 all watch.
Boiler Room Engineers	J Donegan	I. Church	C Hayl

Enter only abnormal characteristics of the coal. If there is nothing outstanding, place "Normal" on Remarks line and make no other note. Use back of sheet for additional remarks if necessary.

FIG. 4 COAL-CHARACTERISTIC DAILY REPORT; BY BOILER-ROOM ENGINEERS

here when compared with our track-scale weights, the latter figure being only of the order of a check. A few shipments, and very few as a rule, show we are receiving more than we pay for, but most are the other way. This variation may be from 0.5 to 1.6 per cent which would represent some 70 to 220 Btu in heat rate.

These figures appear high; an average would be of the order of 0.6 per cent. However, when it is considered that there is car leakage en route to barge, that there is considerable spillage loading and unloading barges, that all vessels are not completely

cleaned, that moisture variation must be taken into account, etc., it is within reason.

Kind of Coal. From the engineer's viewpoint, we are interested in producing our kilowatt from a minimum weight of coal and with a minimum expenditure of heat but this is not a true "yardstick." In the last analysis it is: "How much does it cost?" Fractions of a cent should be considered rather than pounds or British thermal units.

With this in mind, we are continually seeking fuels that will bring down our cost in cents per kilowatthour, even if by so doing we increase our pounds per kilowatthour. This search consists of testing the coal (1) to determine if its characteristics fit our equipment so that we will be sure to supply the demand, and (2) if it does, whether the cost per unit of output will be reduced.

The tests are not elaborate and extreme accuracy is not necessary. One bunker is emptied and the test coal, 1700 to 3500 tons, placed in it and used exclusively under one of our row 6 boilers and the opposite boiler in row 7. The tests last three days, the first two for noting the burning characteristics and the last day for conducting a 10-hr weighed-coal run at a fixed and fairly high rating on one boiler.

To determine the burning characteristics, we operate both boilers during the day on automatic control with or without overfire air or liquid fuel as the load demands. In the evening of the first day, without the aid of the overfire air or fuel, we find the nonnuisance heavy peak rating for 2 hr and the emergency maximum for 30 min. The same two points are determined the next evening with the overfire air and liquid fuel.

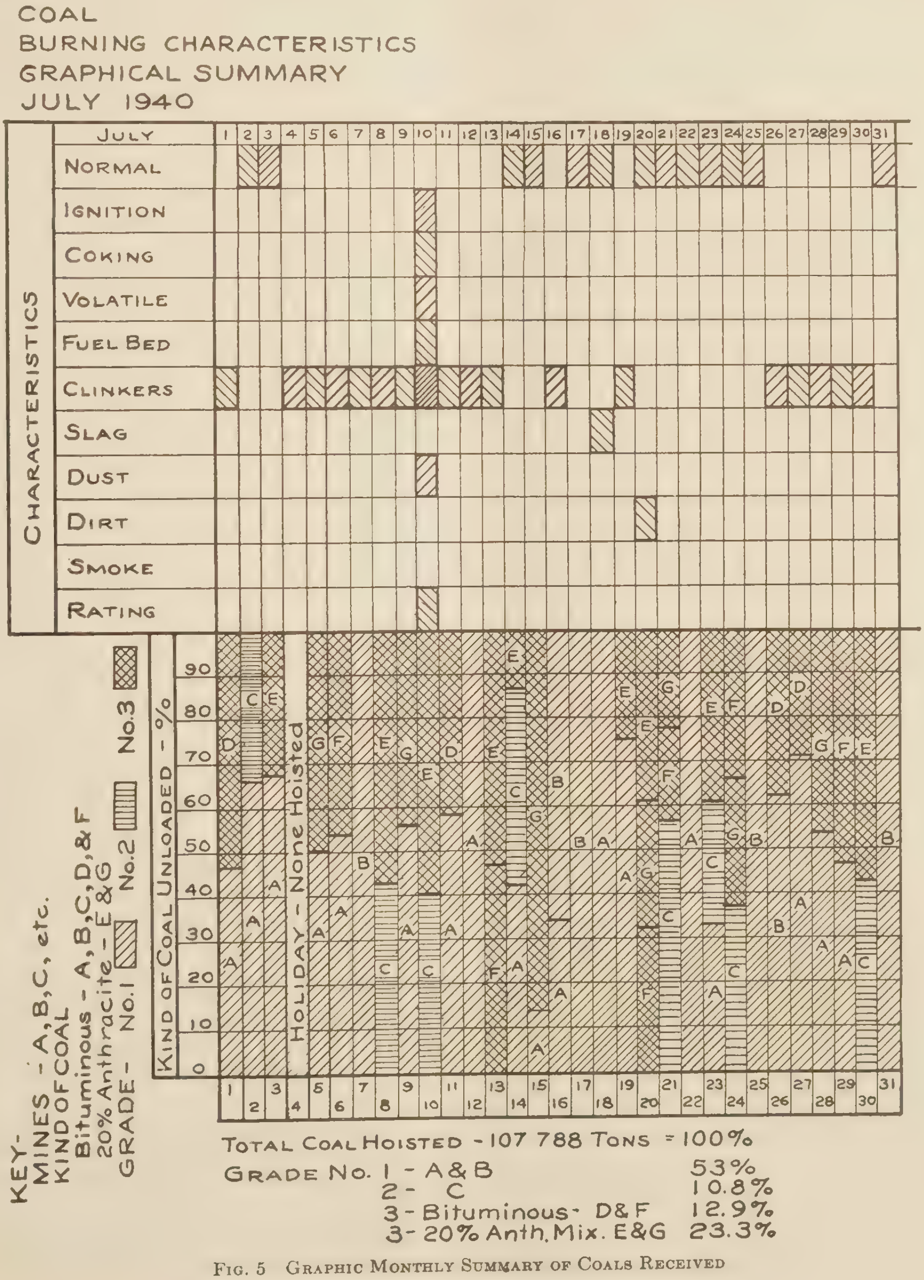

FIG. 5 GRAPHIC MONTHLY SUMMARY OF COALS RECEIVED

The boiler-room engineer reports on these findings as well as the test crew. On the morning of the third day, the stoker hopper of the row 7 boiler is leveled, the rating set at a fixed amount, and all coal to the hopper weighed by the lorry for 10 hr. From this weight and the steam output from the boiler-meter chart, the economic value of the coal, as compared to our standard coal, is determined.

These data are sufficient to reach a decision as to whether more complete testing is necessary and allow us to classify or rate the coal.

We classify our coals from the test data into three grades, Nos. 1, 2, and 3. The No. 1 grade is best suited to our requirements and the No. 3 the least, with a top nonnuisance rating of some 30,000 to 50,000 lb per hr of steam less than No. 1. The No. 2 is somewhere between the others.

The No. 1 coals are straight bituminous, some of the No. 3 are also bituminous, but many are a mixture of one or two kinds of bituminous and 20 per cent of No. 4 (size) anthracite. The latter mixtures have great possibilities, for at once we have 20 per cent of the cost of the fuel at a very much lower price per ton, possibly reducing the unit cost by 25 to 30 cents.

Before such mixtures are successful, however, the bituminous, which acts as a binder and supplies ignition, must be found. In some cases, a single bituminous may answer, in others two may be used. As an example, we have a bituminous which cokes so much that it is too good a binder and although it has fairly high volatile, it ignites almost as slowly as buckwheat. Another coal, though low in volatile, distills very quickly and ignites readily but is so friable that it produces as much cinders as the anthracite and cokes but little. Combining these coals in a 50/30/20 mixture gives excellent results. Now, if some of these bituminous coals may be purchased at a price below that of our No. 1, then we reduce the 80 per cent as well as the 20 per cent.

Assuming such a coal with a unit price of 45 cents less than the No. 1 and our capacity is such that we can get sufficient output, then we can afford to burn 5 to 6 per cent more of it. But all such coals reduce capacity and some limit must be placed upon their use, or else so many more boilers must be carried that we would nullify any saving. Therefore, the coal must be made to fit the load.

We have five bunkers, one over each firing aisle, and each is divided into three parts, the middle one being the larger. The No. 3 grade coal is placed in this larger section for rows 4–5, 6–7, and 8. During light-load periods it is used for all of these boilers and for the heavier loads on rows 5 and 6 with the overfire air and on rows 7 and 8 with the overfire liquid fuel. No. 1 or No. 2 grade coal is used for the heavy loads on row 4 and for any No. 1, 2, or 3 boilers which may be in service, as well as for any abnormal loads which we may have to carry on the other boilers.

In addition to testing the new coals, we make a continuous check on the coals received in the same manner as for the burning tests. This is done by the boiler-room engineers, who each day turn in a "coal-characteristic daily report," Fig. 4. On this they note any abnormal condition found. The data from these sheets are plotted on a current monthly sheet or graphical summary, on which is the daily unloading, classified in grades. This gives a comprehensive idea of the coals used and, if applied intelligently, it is worth-while. Such a sheet is shown in Fig. 5.

Boiler Control Room. Centrally situated on the operating floor of the boilerhouse is the boiler-control room, in which is located most of the control equipment for the making and, if necessary, the emergency distribution of all steam. The equipment may be divided into four groups: (1) Steam-valve control; (2) boiler control for 300-psi boilers; (3) steam control for 400/300 psi; (4) boiler control for 400-psi boilers.

Steam-Valve Control. A diagram board of all boilers, steam lines, and motor-operated valves is mounted on the wall with red and green signal lamps for valve position indication and "close" buttons for some 118 valves, 10 in. to 24 in. in size.

300-Psi Boiler Control. For the 300-psi boilers there are supermasters to control all twelve boilers. These may be operated straight automatic from steam-pressure variation or manually. These boilers have no induced fans but are equipped with forced-draft fans in common. Regulation is provided for their speed, for gas volume by outlet boiler-damper position, and for coal feed by stoker-motor speed control.

400/300 Psi Steam Control. The operating mechanism, gages, thermometers, and flowmeter recorder for reducing the pressure and temperature of the 300-psi steam from the 400-psi boilers is located in this room.

400-Psi Boiler Control. This control consists of a supermaster, either fully or part automatic, which controls five submasters, one for each boiler row. Leak-offs are provided for distribution of load between rows. The regulators for the equipment are listed in Table 3.

TABLE 3 CONTROL REGULATORS FOR 400-PSI BOILERS

	Row number 4	5 and 6	7 and 8
Induced-fan speed	×	—	—
Delivery by vane position	—	×	×
Forced-fan speed	×	—	—
Delivery by vane position	—	×	×
Wind-box damper	×	×	×
Boiler-outlet damper	×	—	—

For row 4, the oiler at the forced-and-induced fans starts and stops them on signal from the boiler-control operator; all speed regulation is from the control room.

All of the forced-draft fans for the other rows are started and shut down by the oiler at the fans, and the delivery is regulated by vane position, automatically.

The Nos. 5 and 6 induced fans are started, speed changed, and shut down by the oiler at the fans and the volume is regulated by vane position.

All of the operation of Nos. 7 and 8 induced fans is by the stoker operator at his boiler panel, and volume control is by vane position, regulated automatically.

Totalizing wattmeters and pressure gages are provided for each group together with direct telephones to the low-tension board, the turbine room, and to the stack observers.

There are two specially trained boiler-control operators on duty at all times and it is the headquarters of the boiler-room engineer. It is also a clearing house for all orders, hold-offs, or other boiler-room information.

Control Equipment at Boilers. As indicated, the boiler-control operator handles most of the equipment for the first four rows, but the last four come more directly under the control of the stoker operator at the boiler.

Personnel. It has been truly said that "a fire room will make or break a powerhouse," and we could go yet further and substitute "fireman" for "fire room." All fires must be inspected at frequent intervals and the operator must understand what he sees and know how to correct it, if necessary. So with all of our planning and with all of our indicating and automatic equipment, we still must depend upon the operator at the boiler.

Our operators are all trained by us. The more promising young boilerhouse maintenance men are used for sick and vacation relief work on operation, first on lorries and then as helpers on the boilers. The company conducts classes in combustion and boiler operation which the men are urged to attend, attendance counting for promotion.

Each watch is in charge of a boiler-room engineer who devotes all of his time to operation. Each stoker operator is assigned

two boilers and, for rows 7 and 8, he has a helper. Each operator signs his work by placing his signature on the boiler-meter chart.

In addition to the regular operators, we have a small technical crew which spends its time in the boilerhouse, working not only with the engineers but directly with the operators. For special tests we call in our technical service department.

Checking

First we plan, then operate, and finally become bookkeepers to determine our operating costs. Our quantities are large, the coal bill alone amounting to more than $500,000 per month, so even a small increase is an appreciable amount. Therefore, to guard against such increases, the checking must be continuous. We have outlined some of these checking measures under the subject of operation, as they are so closely allied.

Steam Generated. Each day, each boiler-meter chart is examined to find:

1 If there has been any deviation from scheduled loading.
2 If there is any spread between steam and air pens.
3 The quantity of steam generated.

The operator writes across the face of the charts any reasons for (1) or (2) and there may be good ones. Also such explanations are made in the "daily-boiler-log" sheet.

The turbine main and heater condensate is metered and this, together with necessary corrections for estimated blowdowns and steam sent out of the station, is compared with the integrated totals from the chart and used for the quantity generated.

Temperatures. A running check is made of temperatures of steam, gas, air (for preheaters), etc., and any deviation from standard is checked at once.

Heat. We have no continuous coal sampling at the present time, the coal being hand-sampled from each 5-ton bunker car. The heat in the refuse is from ash samples from the disposal barges.

Draft. Comparison of draft drops guides us on our lancing or external cleanings.

Any other checking is more or less common to all boiler operation.

An Analysis of the Milling Process

By M. E. MARTELLOTTI,[1] CINCINNATI, OHIO

This analysis of the milling process shows that the path of a milling-cutter tooth is an arc of a trochoid, the parametric equation for which can be derived from known variables of the cut. As a result, the milling process is susceptible to mathematical treatment. Practically, the advantage of such analytical methods is that such elements as the radius of curvature of the tooth path, the clearance and rake angles of the length of the tooth path, the radial thickness of the chip, and their effects upon the quality of milled surface may be evaluated. The paper demonstrates the advantages of the up-milling process in achieving machined surfaces of high quality. This form of analysis will also be found useful in comparing different milling methods.

Introduction

Milling is a process of removing the excess material from the workpiece in the form of small individual chips. These chips are formed by the intermittent engagement with the workpiece of a plurality of cutting edges or teeth integral with or inserted in a cylindrical body known as the milling cutter. This intermittent engagement is produced by feeding the workpiece into the field dominated by the rotating cutter.

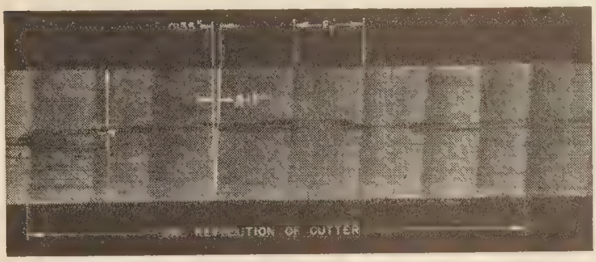

Fig. 1 Tooth Marks on a Milled Surface
(Material, brass; cutter, spiral mill, 8T, 3.89 in. diam, 38 rpm; helix angle, 35 deg; feed rate, 36 in. per min; depth of cut, $^1/_{32}$ in.; $\times 3$.)

The finished surface, therefore, consists of a series of elemental surfaces generated by the individual cutting edges of the cutter, Fig. 1. The method of milling in which the cutter rotates in a direction opposite to or against the motion of the work is usually called "conventional milling," but is better described as "up milling," since the chips are formed while the teeth of the cutter move in an upward direction, away from the finished surface of the work, Fig. 2.

Kinematics of Milling

Owing to the limited period of engagement of each tooth, a milling chip is short and of a variable thickness, Fig. 3. This results from the combination of the translatory motion of the work and the rotary motion of the cutter. Hence, the direction of motion of the tool point is continuously changing with respect to the direction of motion of the workpiece, and the path of the tooth resulting therefrom is not circular, but of the type which is properly described as trochoidal.

The name "trochoid" is given to that family of curves which are the locus of a point (taken along a radial line, originating at the center of a circle), generated while the circle rolls on a straight line X. As indicated in Fig. 4, a point P, chosen on this radial line at a distance from the center of the circle which may be either less, equal, or greater than the radius of the circle, will generate the curves A, C, and B, which are respectively called, "prolate trochoid," "cycloid," and "curtate" or "looped trochoid."

In analyzing the milling process, valuable assistance may be realized from a consideration of the kinematic characteristics, defined by the equation of the path of a tooth. In previous investigations of the milling process[2] it has been customary, for the sake of simplicity, to assume that the path generated by the cutter tooth is circular. However, as will appear from the following analysis, it is not necessary to make this assumption, as the various relationships may readily be calculated from the true trochoidal tooth path. This affords a proper understanding

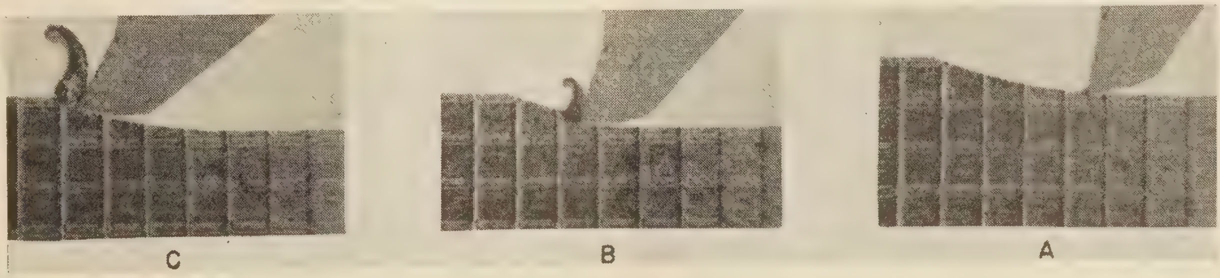

Fig. 2 Milling Chip at Different Stages of Formation
(A, at beginning of tooth path; B, at middle of tooth path; C near end of tooth path.)

of the milling process, and brings to light certain important events, such as the changes in the clearance and rake angles, and radius of curvature of tooth path.

For the purpose of analysis, the system composed of a rotating milling cutter and a translating workpiece may be replaced by an equivalent system in which the work is stationary and the cutter rotating and translating at the same time. In this case, however, the direction of the translatory motion of the cutter or

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Contributed by the Machine Shop Practice Division and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[2] "Chip Thickness in Milling," by C. H. Borneman, *American Machinist*, vol. 82, 1938, pp. 189–190.

"How Thick a Chip?" by A. L. DeLeeuw, *American Machinist*, vol. 83, 1939, pp. 991–992.

feed will be opposite to that of the work, which obtains in the ordinary case.

The idealized version of this arrangement, shown in Fig. 5, and applied particularly to the case of slab milling, can be illustrated by means of a mechanical system, of the rack-and-pinion type, in which the pinion Q of the proper dimension is integral with the spindle and cutter and meshes with a rack Z rigidly supported on the stationary base of the machine. Upon rotation of the spindle, the cutter will be fed to the work in a direction which obtains in an ordinary milling machine.

Since the cutter is translated at a rate corresponding to the feed of the work, the pitch radius r of the pinion may be determined by the relation

$$2\pi rn = F \quad\quad [1]$$

where

F = feed rate, in. per min
n = rpm of cutter and pinion
r = pitch radius of pinion, in.

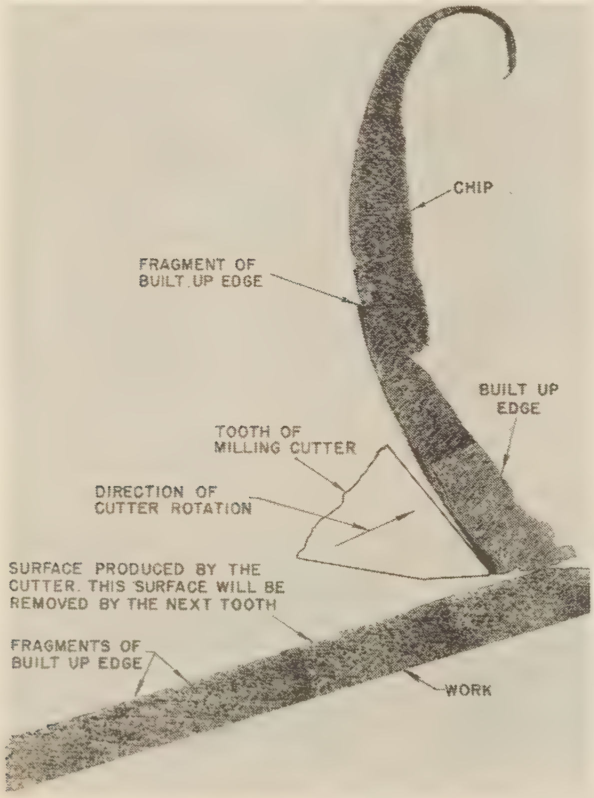


Fig. 3 Complete Up-Milling Chip
(Material, S.A.E. 1112; feed rate, 6¼ in. per min; depth of cut, ⅛ in.; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 40 fpm; ×50.)

In practical application, the surface speed of the cutter is always much greater than the feed rate of the work, consequently R, the radius of the cutter, will be much greater than r, the pitch radius of the pinion.

By referring to Figs. 4 and 5, it is evident that, by allowing the pinion Q to roll on the rack Z, the edge of the tooth will generate a looped trochoid. This can be represented by the parametric equations

$$\left.\begin{aligned} X &= r\,\alpha + R \sin \alpha \\ Y &= R\,(1 - \cos \alpha) \end{aligned}\right\} \quad\quad [2]$$

In Equations [2], α is the angle through which the cutter and pinion have rotated in the direction of the arrow M from the starting point O, which is the origin of the coordinate system XY.

Two consecutive teeth, 2 and 1, Fig. 5, are shown, on an enlarged scale, in their relative position, respectively, at the beginning and near the end of their engagement with the work. Tooth 1 has nearly completed the path A'-N, while tooth 2 is about to contact the work at a point marked A, along the path of the tooth 1.

Tooth and Revolution Marks

As the pinion is rolled on the rack in the direction of the arrow M, tooth 2 moves downward into the work, and not tangentially

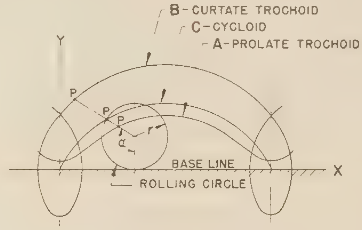


Fig. 4 Trochoids

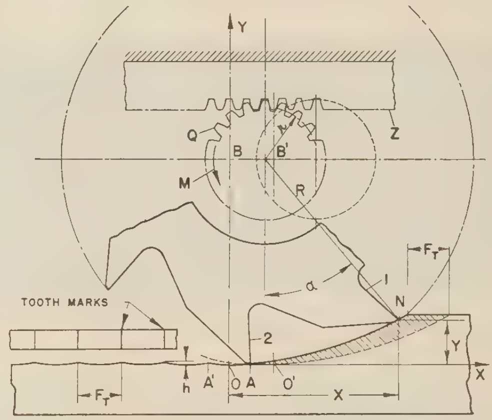


Fig. 5 Path Generated by a Milling-Cutter Tooth

as is usually believed to be the case, until the point O' is reached. The tooth is then at the greatest perpendicular distance from the path to the center B'. Thereafter, the tooth begins to rise toward the upper surface of the workpiece.

A cusp known as the tooth mark results from the intersection of the path of two consecutive teeth, by the direct action of the edge of the tooth upon the work material. The distance between two adjacent tooth marks is known as the feed per tooth, F_t. This is obviously equal to the distance F_t on the upper surface of the work, Fig. 5.

A milled surface, therefore, is composed of innumerable elements of tooth paths, each delimited by the feed per tooth, as shown in profile and enlarged both in Fig. 5 and in the actual reproduction, Fig. 1.

The uniformity of the tooth-mark spacing depends upon the location of the points A A' where the teeth intersect the paths generated by the preceding teeth.

If in the parametric Equations [2] the quantity 2π is added to the angle α, the net result is a translation of a given tooth path, in the direction of the feed, by an amount equal to the feed per revolution; thus, a more general system of parametric equations of the tooth path results

$$\left.\begin{aligned} X &= (2\pi K + \alpha)r + R\sin(2\pi K + \alpha) \\ Y &= R[1 - \cos(2K\pi + \alpha)] \end{aligned}\right\} \ldots\ldots [3]$$

where K is an integer number.

This indicates that the paths generated by the teeth of a milling cutter are congruous, and that they can be obtained from each other by a displacement equal to the feed per tooth, or a multiple thereof.

By eliminating the parameter α between the parametric Equations [2], the Cartesian form of the looped trochoid of a milling-tooth path will be obtained

$$X = r\cos^{-1}\frac{R-d}{R} \pm [(2R-d)\,d]^{1/2} \ldots\ldots\ldots [4]$$

where d indicates the instantaneous depth in place of Y.

The equations which have been derived, whether of the parametric or of the more involved Cartesian form, furnish means for determining the shape of the trochoid curve generated by any tooth of a milling cutter, as a function of a few variables of the cut, which are generally known. From the Cartesian equation, for instance, the value of X for any point, taken with respect to the origin O, can be obtained when the radius of the rolling circle r, the radius of the cutter R, and the depth d at different positions of the tooth path are known. Thus the tooth path may be mathematically reconstructed.

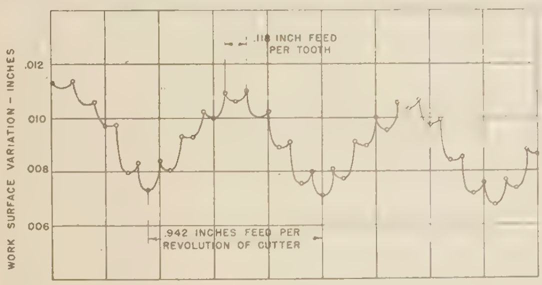


Fig. 6 Variations Produced by Tooth and Revolution Marks on a Milled Surface

(Material, brass; cutter, spiral mill, 8T, 3.89 in. diam, 38 rpm; helix angle, 35 deg; feed rate, 36 in. per min; depth of cut, $^1/_{32}$ in.)

Since the tooth path, in approaching and leaving the point O, is symmetrical with respect to that point, the points of intersection (such as A and A') will be located at one half the feed per tooth from either side of point O. This is exactly true, however, only under the conditions seldom obtained in practice, viz., that the teeth have the same distance from the center of the cutter, and that the feed per tooth is constant and the teeth are equally spaced.

As can be seen from Fig. 1, any variation in the values of these quantities will also affect the height h, but this effect is usually small, and thus may be disregarded. The magnitude of h can be calculated from the following equation, which has been derived from the parametric Equation [2]

$$h = \frac{F_t^2}{8\left[R + \dfrac{F_t \times T}{\pi}\right]} \ldots\ldots\ldots\ldots\ldots [5]$$

where

h = height of tooth mark above point of lowest level, in.
F_t = feed per tooth, in.
R = radius of cutter, in.
T = number of teeth in cutter

The value of h, calculated with Equation [5], and for 0.118 in. feed per tooth, 8 teeth, 3.894 in. diam of cutter, and using a spiral mill, was 0.000766 in.

The average value of h obtained from the measurement of 52 tooth marks produced in an actual cut on brass, under the foregoing conditions, was 0.000710 in., thus showing a satisfactory agreement between the calculated and observed values. Fig. 1 shows this surface, which was purposely milled at an unusually high feed rate in order to show the tooth marks clearly. The actual variations in the surface are shown in Fig. 6.

In addition to the tooth marks, which correspond to the tooth frequency, a slab-milled surface shows periodic variations having a wavy appearance, the frequency of the waves being equal to the frequency of rotation of the cutter. The amplitude of the wave (or height of the revolution mark) is a function of the eccentricity of the cutter and arbor, the so-called "high tooth" on the cutter, and the periodic variation in the deflection of the arbor, caused by the presence of the keyway and possible uneven conditions on the arbor supports.

The height of a revolution mark, in the case shown in Fig. 6, is 0.004 in. It may be observed also that the tooth marks follow the undulation of the revolution marks. The presence of one or the other or both types of marks alters the physical condition of the surface and consequently the quality of its finish.

While the height of a tooth mark can be reduced by increasing the radius of the cutter, and by decreasing the feed per tooth until the tooth marks become scarcely distinguishable, particularly at the lower feed rates, the revolution marks, on the other hand, cannot be prevented unless special care is taken in grinding the teeth of the cutter within close limits and making sure that the runout of the cutter and arbor is reduced to the lowest possible value.

Radius of Curvature of Tooth Path

The difference in the height of a tooth mark, produced under various operating conditions, is due to the instantaneous radii of curvature of the path of the tooth. This is a function of the radius of the cutter, its speed, the depth of cut, and the feed rate, factors which enter into a milling operation.

In conventional milling (up milling), the radius of curvature at any point in the path of the tooth is expressed as

$$\rho = \frac{\left[R^2 + \left(\dfrac{F_t T}{2\pi}\right)^2 + \dfrac{F_t T}{\pi}(R-d)\right]^{3/2}}{\dfrac{F_t T}{2\pi}(R-d) + R^2} \ldots\ldots\ldots [6]$$

The variables are

d = depth of cut, in. (at point under consideration)
R = radius of cutter, in.
F_t = feed per tooth, in.
T = number of teeth in cutter
ρ = the instantaneous radius of curvature in inches

The radius of curvature obtained from Equation [6] is the instantaneous radius of a small portion of the path of the tooth in the neighborhood of a point determined by a given combination of the variables mentioned.

By assuming certain values for three of these variables, the relationship has been determined between the radius of curvature and the depth of cut, the revolutions per minute of the cutter, and the feed rate, Figs. 7, 8, and 9.

From an inspection of these results, the following conclusions may be drawn:

1 A large radius of curvature is obtained at shallow depth of cut. This condition obtains particularly within the length of a tooth path delimited by the feed per tooth. Beyond this

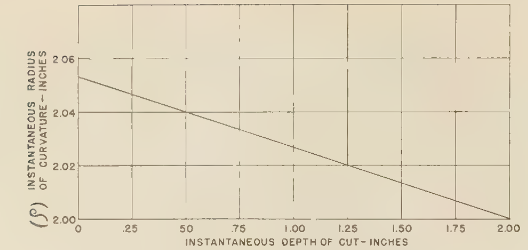


FIG. 7 RADIUS OF CURVATURE OF TOOTH PATH VERSUS DEPTH OF CUT

(Radius of cutter, 2 in.; feed rate, 10 in. per min; cutter speed, 6 fpm.)

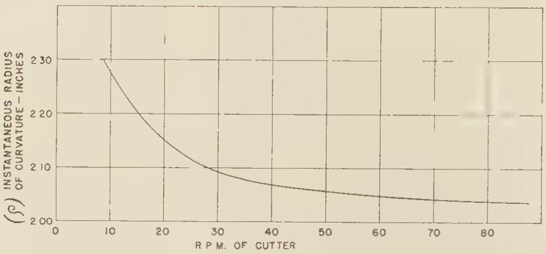


FIG. 8 RADIUS OF CURVATURE OF TOOTH PATH VERSUS REVOLUTIONS PER MINUTE OF CUTTER

(Radius of cutter, 2 in.; depth of cut, $^1/_4$ in.; feed rate, 10 in. per min.)

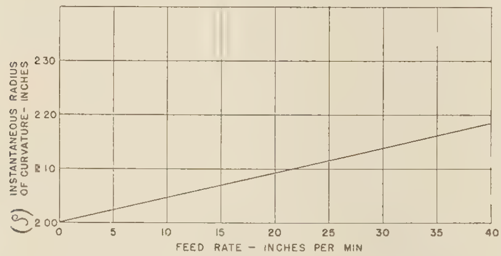


FIG. 9 RADIUS OF CURVATURE OF TOOTH PATH VERSUS FEED RATE

(Radius of cutter, 2 in.; depth of cut, $^1/_4$ in.; cutter speed, 60 fpm.)

elemental portion of the tooth path and, as the tooth travels toward the upper surface of the workpiece, the radius of curvature decreases, reaching a minimum value equal to the radius of the cutter when the depth of cut is likewise equal to the radius of the cutter, Fig. 7.

2 For any given cutting condition, as we increase the cutting speed and, consequently, the revolutions per minute of the cutter, the radius of curvature is correspondingly decreased. For very high values of the revolutions, it tends to approach, as a limiting value, that of the radius of the cutter, Fig. 8.

3 When the feed rate is increased, the radius of curvature increases proportionately with it. The relationship between radius of curvature and feed is shown in Fig. 9.

If we consider the variation of the radius of the curvature, in relation to the variables of the cut, and on the basis of experimental evidence, we can conclude that it is always desirable to have a set of conditions such as will give, at any time, the largest possible radius of curvature. It is well known that more efficient metal removal is usually obtained for relatively large values of the feed per tooth, Figs. 10 and 11.

Since the feed per tooth is a function of the feed rate, the number of teeth and the revolutions per minute of the cutter, as expressed in the equation

$$F_t = \frac{F}{T \times n} \qquad [7]$$

where

F = feed, in. per min
T = number of teeth in cutter
n = rpm of cutter

it is apparent therefore that a large value of the feed per tooth will be obtained for any given number of teeth and feed rate, by choosing a low value for the revolutions per minute of the cutter. This will automatically give us a large radius of curvature.

An illustration of the effect of the cutting speed of a milling cutter on the actual power required to remove a given volume of metal (in this particular case, machinery steel) is given in Fig. 12. From this illustration, it will be seen that, for a constant feed rate, the minimum power required to remove the same amount of metal is always obtained at the lowest cutting speed. The power increases as the cutting speed is increased.

If the same volume of metal is removed under two different conditions, by adjusting the feed rate and the depth so as to give the same metal removal, it is found that the most efficient removal of metal is obtained at a shallow depth and the highest feed rate, as shown in Fig. 13. As an example, if the feed rate in one case is 10 in. per min and the depth of cut $^1/_4$ in., and in the second case the feed rate is 20 in. per min and the depth of cut $^1/_8$ in., the power required at the cutter is 14 and $12^1/_2$ hp, respectively.

It is also evident from the factors entering into the expression for the radius of curvature that, by increasing the radius of the cutter, it is possible to make the arcs of the trochoid yet flatter and thereby reduce the height between the cusps and the trough of each elemental surface, thus improving the quality of the finished surface.

From the foregoing considerations, it is apparent that the radius of curvature can be used to indicate, both qualitatively

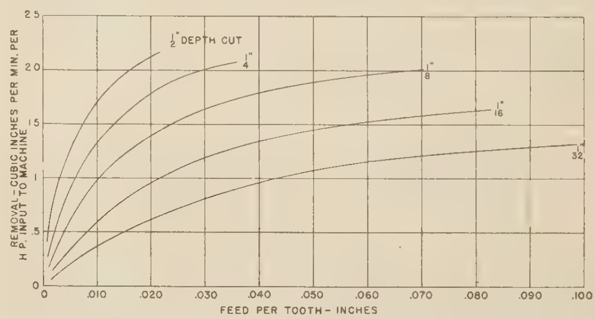


FIG. 10 METAL REMOVAL, CUBIC INCHES PER MINUTE PER HORSEPOWER INPUT TO MACHINE VERSUS FEED PER TOOTH

(Material, cast iron; cutter, spiral mill, 8T, 4 in. diam; rake angle, 10 deg; clearance angle, 3 deg; helix angle, 25 deg; width of cut, 4 in.; cutting speed, 66 fpm.)

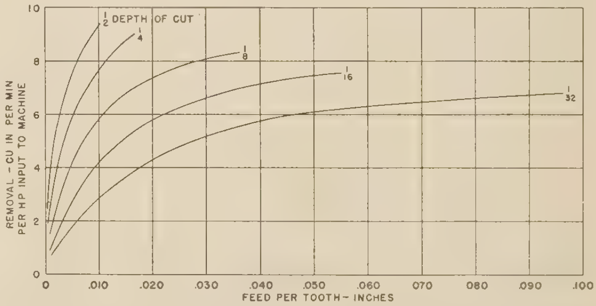


FIG. 11 METAL REMOVAL, CUBIC INCHES PER MINUTE PER HORSEPOWER INPUT TO MACHINE VERSUS FEED PER TOOTH

(Material, S.A.E. 1112 steel; cutter, spiral mill, 8T, 4 in. diam; rake angle, 10 deg; clearance angle, 3 deg; helix angle, 25 deg; width of cut, 4 in.; cutting speed, 66 fpm; cutting fluid, soluble oil and water.)

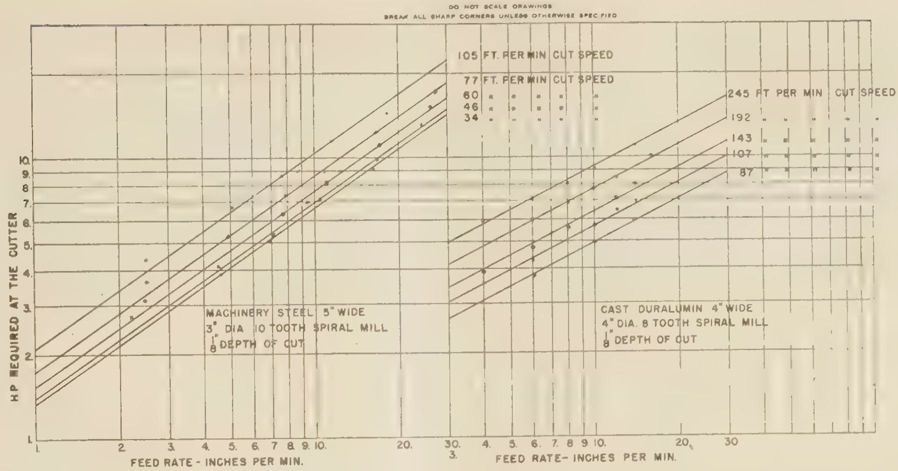


Fig. 12 Horsepower at Cutter When Milling Steel and Duralumin at Different Cutting Speeds Versus Feed Rate

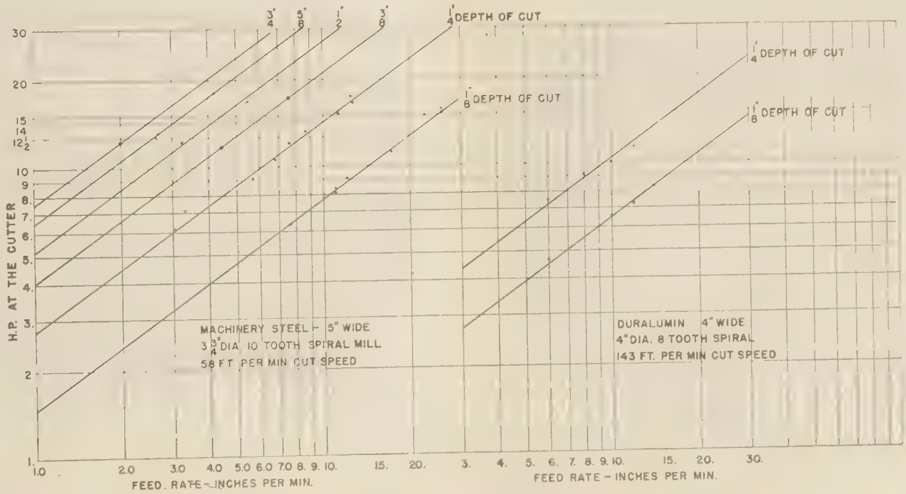


Fig. 13 Horsepower at Cutter When Milling Steel and Duralumin at Different Depths of Cut Versus Feed Rate

and quantitatively, the effects produced on the path of the tooth by the changes in the values of one or more of the variables of the cut. Consequently, these variables can be adjusted to obtain any desired results.

In addition to the effects mentioned, which are inherent in the methods of generating a milled surface, there are other effects which are induced by the flowing metal as the excess material is removed in the form of chips.

Quality of Finish in Milled Surfaces

In all metal-cutting processes, it is found that, at the very beginning of the contact of the cutting edge of the tool with the work material, and for a short distance thereafter, the machined surface is shiny in appearance and of uniformly good finish, while later on, the surface quality changes to that of a dull, coarse finish. This transition is illustrated in Fig. 14.

The length of the shiny surface in the direction of the cut seems to depend upon the kind of material, the depth of cut, the clearance and rake angles, and particularly on the presence, strength, and quality of a film in the nature of an oxide, or an oil film of molecular dimensions, existing on the cutting edge of the tool and its adjacent surfaces at the time of its first contact with the work.

When this film is eventually removed by the combined effect

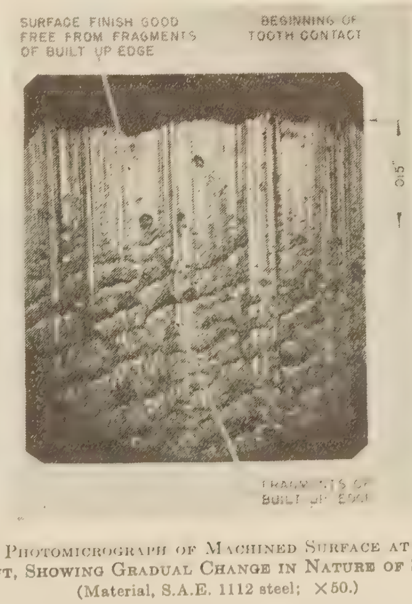


Fig. 14 Photomicrograph of Machined Surface at Beginning of Cut, Showing Gradual Change in Nature of Surface (Material, S.A.E. 1112 steel; ×50.)

of temperature and pressure developed at the root of the chip, and by the wiping action resulting from the motion of the chip, the surface of the tool becomes chemically very active; and therefore the material at the under surface of the chip, which is in a nascent (or freshly formed) condition, will bond readily to the tool.

When this has occurred, the resistance to the motion of the plastic metal of the chip in contact therewith will be increased. This will cause the metal in the vicinity of the cutting edge to be retarded in its motion, and thus the crystals of the chip material become elongated in a direction roughly parallel to the face of the tool.

As the motion of the chip continues, a portion of this material stressed beyond the elastic limit will finally rupture from the main body of the chip and cling to the tool. As the tool continues to advance, the highly stressed material piles up and continues to grow both along the face and the region below the flank of the tool, thus forming the so-called built-up edge. Eventually an unstable condition is reached when a portion of material located on the outer boundary of the built-up edge, no longer sufficiently supported by the tool and under the action of the forces produced in the formation of the chip, will shear off periodically from the underbody and escape as small irregular fragments, both with the chip and the work.

In a metal-cutting process where a tool remains engaged with the chip from the beginning to the end of the cut, and where the path of the tool covers the full length of the final surface, the presence of the "built-up or pseudo edge" is particularly detrimental to the formation of a good surface finish.

The use of a cutting fluid is known to reduce the size of the built-up edge and, in special cases, to prevent its formation.[3] Under normal conditions, however, the built-up edge will never

[3] Symposium, "Machining of Metals," American Society for Metals, 1938.

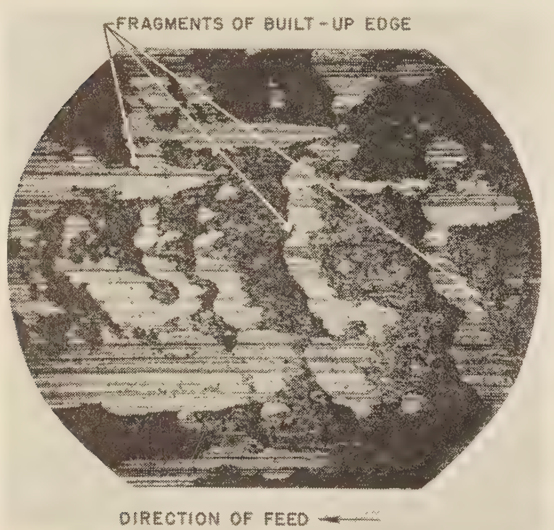


Fig. 15 Photomicrograph of Machined Surface Showing Fragments of Built-Up Edge
(Material, S.A.E. 1112 steel; ×50.)

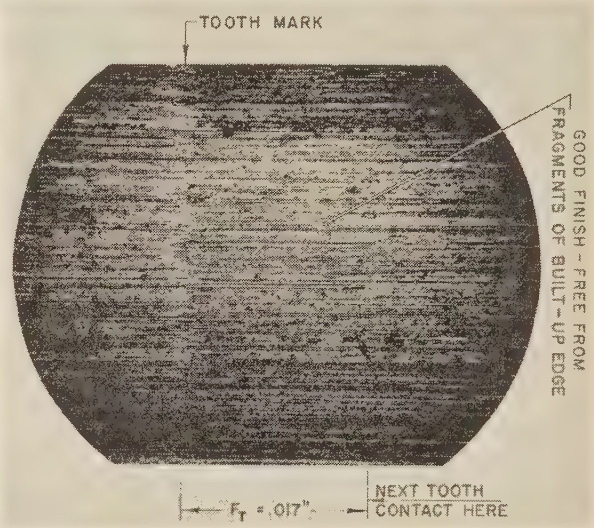
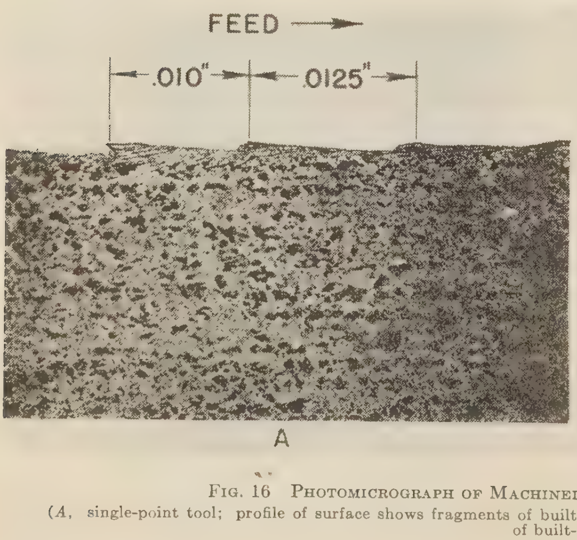


Fig. 17 Photomicrograph of Milled Surface Showing Tooth Mark and Position of Next Tooth Contact
(Material, S.A.E. 1112; feed rate, 6 1/4 in. per min; depth of cut, 1/8 in.; utter, spiral mill, 10T, 4 in. diam; cutting speed, 40 fpm; cutting fluid, soluble oil and water; ×50.)



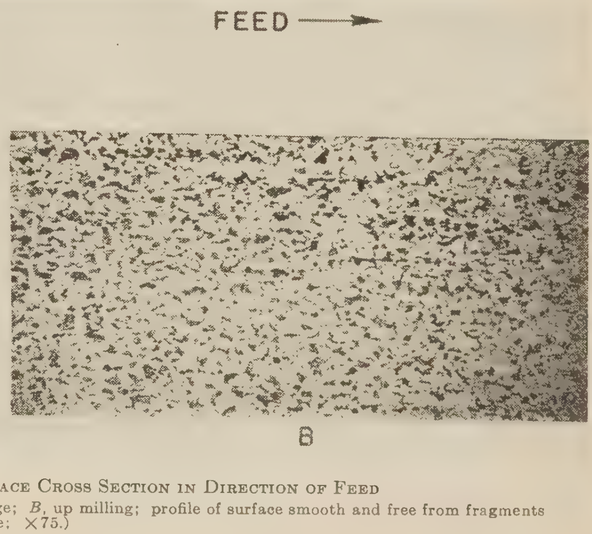


Fig. 16 Photomicrograph of Machined Surface Cross Section in Direction of Feed
(*A*, single-point tool; profile of surface shows fragments of built-up edge; *B*, up milling; profile of surface smooth and free from fragments of built-up edge; ×75.)

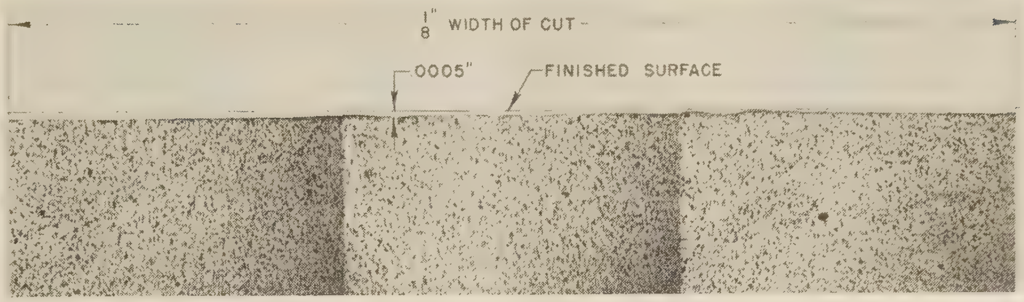


Fig. 18 Photomicrograph of Profile of Milled Surface Normal to Feed
(Produced under same conditions as in Fig. 17; ×50.)

completely disappear, and its fragments will be found on every machined surface in the form of small irregular portions of work material in a highly worked state and aligned in a general direction roughly normal to the direction of the tool path, Fig. 15. A cross section of this surface in the direction of the tool path is shown in Fig. 16 (*A*). Here can be seen the pronounced change in the quality of the finished surface produced by the fragments of the built-up edge.

In the case illustrated in Fig. 14, the element of smooth surface produced at the beginning of the cut extends approximately $^1/_{64}$ in. in the direction of the motion of the tool, but in up milling, this surface extends for a relatively great distance beyond the point where the cut began, Fig. 17.

A milled surface is composed of innumerable elemental surfaces which are generated in the early stages of a tooth path, when the thickness of the chip is extremely small and the cutting edges of the teeth are still covered with the cutting fluid used, and, in addition, with an oxide film formed during the idling period prior to the engagement with the work.

Therefore, as the cutting process continues, successive chips will be formed under nearly identical conditions, and it may be expected to find this quality of surface finish as indicated in Figs. 17 and 16 (*B*), to extend to all elements of the final milled surface. A profile of this surface taken at right angles to the feed within the region of a tooth mark is shown in Fig. 18.

The photomicrographs, Fig. 19, of the actual finish obtained, when milling various kinds of work material with a spiral mill, prove that these deductions are in good agreement with actual practice.

Blemishes on the finished surface are caused, in some cases, by fragments of chips adhering to the teeth and being caught between them and the work in subsequent engagements. This happened in the case of aluminum, Fig. 19 (*A*, *B*), and of stainless steel (Rezistal), Fig. 19 (*C*), all of which were cut dry. When stainless steel was milled with soluble oil as the cutting fluid, a definite improvement in the quality of finish was obtained, as shown in Fig. 19 (*D*). The specimens of copper, brass, and duralumin, Fig. 19 (*E*, *F*, and *G*), show a remarkably good surface finish, particularly copper. The steels including S.A.E. 3115 milled dry, Fig. 19 (*H*), with soluble oil and water, Fig. 19 (*I*), and tool steel, Fig. 19 (*J*), also with soluble oil and water, show indications of small fragments of the built-up edge, due to an early bonding of the material of the chip with that of the tool, thus causing the formation of a small built-up edge almost at the beginning of the cut.

The finish obtained on a cast-iron specimen is shown in Fig. 19 (*K*). The irregularities on this surface are due to successive ruptures caused by the brittleness of the material.

Additional proof that the character of the elemental surfaces is maintained for a certain distance beyond the point *A*, Fig. 5, where the tooth first contacts the work, is shown in the cross section of the chip and the adjacent surface of the work at the beginning of the up-milling cut, performed with a spiral mill on a specimen of S.A.E. 1112 steel, Fig. 20.

The actual point of tooth contact with the work can be determined approximately from a direct measurement of the chip shown in Fig. 20. Here we find that this chip is approximately 0.0008 in. thick and 0.0125 in. long.

Now it has been found that the actual length L_c of a chip bears a definite relation to the length L_s of the resulting surface and the conditions under which it was formed. This may be called the "cutting ratio."

Various values of this ratio obtained on specimens of S.A.E. 1112 steel with a single-point tool, and with various cutting fluids, are listed in Table 1, and the magnified views of the corresponding finished surfaces in Fig. 21.

Comparing the finish of the milled surface, Fig. 17, obtaining at the beginning of the cut, with that of the surfaces shown in Fig. 21, it is found that the finish of the former surface falls between that of the surfaces (2) and (3) of Fig. 21. Hence, the cutting ratio L_c/L_s can be assumed equal to the mean values of the ratios obtaining in these surfaces, or the value of 0.382.

Then, the beginning of the chip, shown in Fig. 20, would be at a distance $L_s = \dfrac{L_c}{0.382} = \dfrac{0.0125}{0.382} = 0.0325$ in. back of its present location, or in the neighborhood of the point marked *A*.

Since the feed per tooth, F_t, used in machining this specimen was 0.0170 in., then the point of intersection of the following tooth will be approximately at *B*, that is, 0.0170 in. from the point where the previous tooth entered. Thus, the element of the finished surface produced by the first tooth is the distance *A*-*B*, Fig. 20, which is well within the region free from fragments of built-up edge, as shown in Fig. 17.

The gradual deterioration of the quality of the milled surface along the path of the milling-cutter tooth is shown in the photomicrograph, Fig. 22 (*A*, *B*, *C*, *D*). At the beginning of the cut, Fig. 22 (*A*), the quality of the finish produced is similar to that obtained in Fig. 14. In Fig. 22 (*B*), taken at a later stage of the tooth passage along its path, the surface shows small fragments of built-up edge; in Fig. 22 (*C*) the surface is rough and shows the extent of nonuniformity produced by large fragments of the built-up edge. This character is maintained in Fig. 22 (*D*), which illustrates the surface at the end of the tooth path.

The cross section of chip and work taken along the ascending portion of the tooth path in up milling is illustrated in Fig. 23. In this case the numerous and rather regularly spaced fragments of the built-up edge are left behind by the advancing tooth at the rate of approximately 2000 per sec. This gives an indication of the rate at which the built-up edge forms and breaks down. This will continue until the chip is finally severed from the work

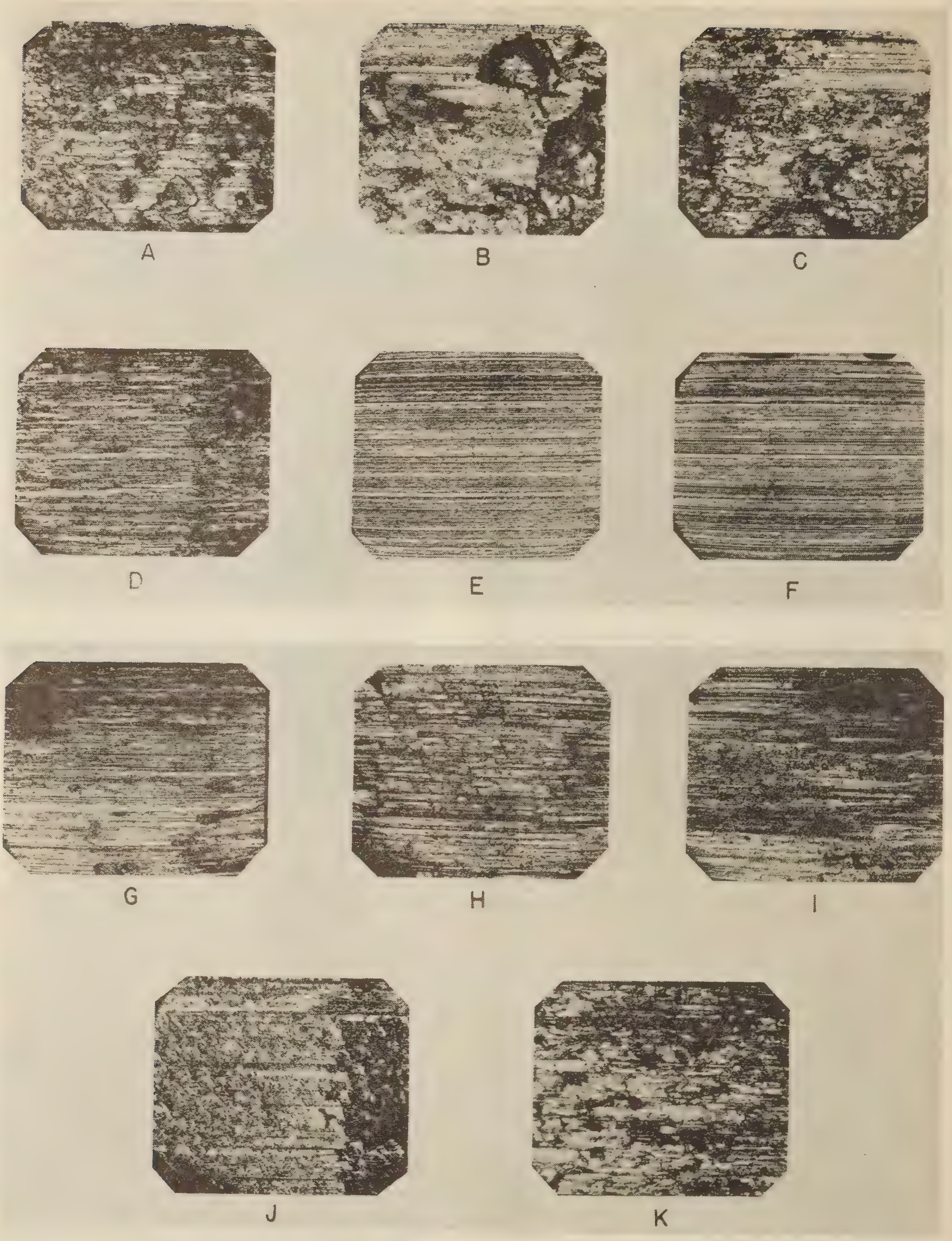


Fig. 19 Quality of Surface Obtained When Milling Various Kinds of Work Material

(*A*, aluminum, cut dry; depth of cut $^1/_{16}$ in., width of cut $^1/_4$ in.; feed rate, 24 in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 675 fpm. *B*, aluminum, cut dry; depth of cut $^1/_{16}$ in., width of cut $^1/_4$ in.; feed rate, $6^1/_4$ in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 230 fpm. *C*, stainless steel, Rezistal, cut dry; depth of cut $^1/_{16}$ in., width of cut $^1/_4$ in.; feed rate, 3 in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 41 fpm. *D*, stainless steel, Rezistal, soluble oil and water; same as under *C*. *E*, copper, cut dry; depth of cut $^1/_{16}$ in., width of cut $^1/_4$ in.; feed rate, $6^1/_4$ in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 230 fpm. *F*, brass, cut dry; conditions same as in *E*. *G*, duralumin, cut dry; conditions same as in *E*. *H*, S.A.E. 3115 steel, cut dry; depth of cut $^1/_{10}$ in., width of cut $^1/_4$ in.; feed rate, 3 in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 40 fpm. *I*, S.A.E. 3115 steel, soluble oil and water; depth of cut $^1/_{16}$ in., width of cut $^1/_4$ in.; feed rate, 3 in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 63 fpm *J*, tool steel, soluble oil and water; depth of cut $^1/_{16}$ in., width of cut $^1/_4$ in.; feed rate, $3^1/_8$ in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 51 fpm. *K*, cast iron, cut dry; depth of cut $^1/_{16}$ in., width of cut $^1/_4$ in.; feed rate, $6^1/_4$ in. per min; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 63 fpm; $\times 50$.)

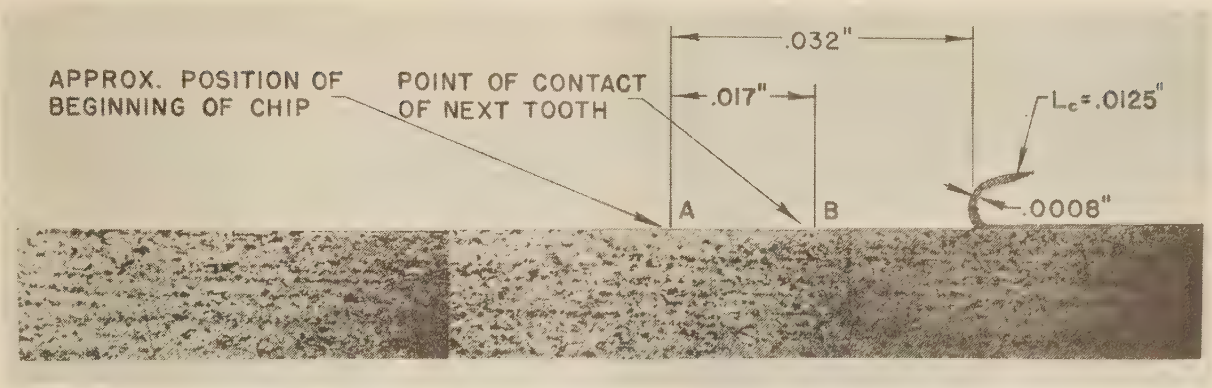


Fig. 20 Photomicrograph of Work and Chip at Beginning of Cut
(S.A.E. 1112 steel; ×75. Other conditions same as in Fig. 17.)

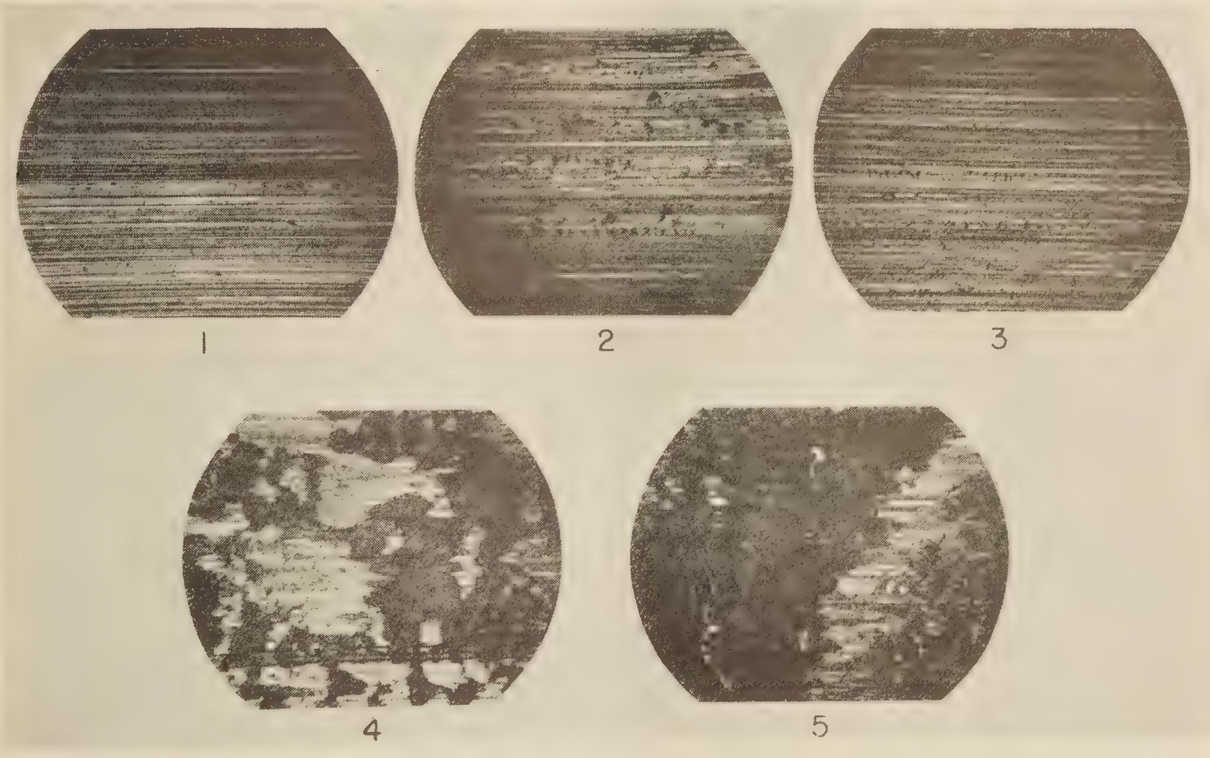


Fig. 21 Quality of Surface Obtained With Single-Point Tool and Various Cutting Fluids
(Material, S.A.E. 1112; depth of cut, 0.005 in.; speed, 5 1/2 in. per min; ×75.)

TABLE 1 CUTTING LENGTH RATIO L_c/L_s
(Material, S.A.E. 1112; depth of cut, 0.005 in.; speed, 5 1/2 in. per min)

Surface no.	Cutting fluid	L_c/L_s	Magnification
1	Carbon tetrachloride	0.493	×75
2	Acetic acid	0.417	×75
3	Methyl alcohol	0.357	×75
4	Turpentine	0.250	×75
5	Benzene	0.213	×75

when the remnant of the built-up edge will pass off with it, Fig. 3.

In up milling, the fragments of the built-up edge appear always along that portion of the tooth path which will be removed by subsequent teeth. Therefore, it is quite evident that a machined surface, having a desirable quality of finish, can be produced by duplicating indefinitely the conditions obtaining at the very beginning of the cut; and, of all known metal-cutting processes, up milling is perhaps the only one which, by virtue of its characteristics, offers such a possibility.

The statement has often been made that the incipient motion of a milling-cutter tooth into the work is accomplished by a momentary sliding for a certain distance, depending upon the hardness of the metal being cut and on the rigidity of the cutter and work support.

From Fig. 24, however, it will be seen that a tooth contacts the work at a point E on the ascending portion of the path generated by the previous tooth with an angle of approach ϕ.

This angle is included between the tangent to the curve at E and the horizontal tangent to the points such as D' and C'.

The value of this angle can be determined from the equation

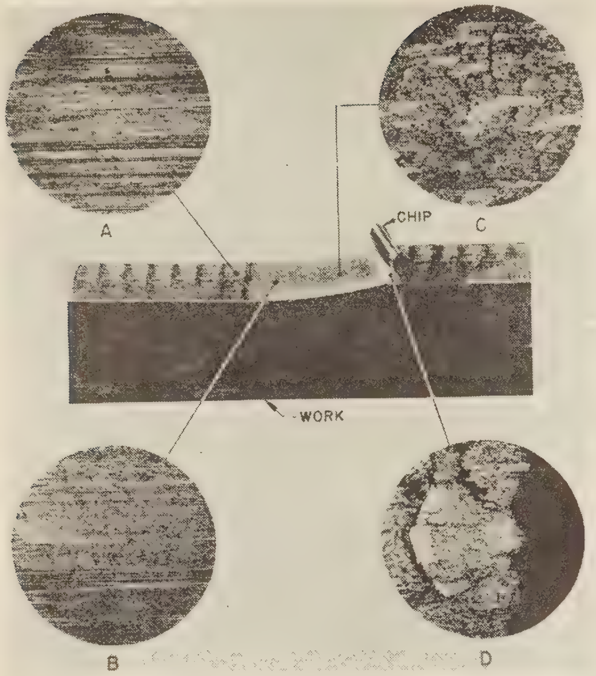


Fig. 22 Change in Surface Quality Along Tooth Path
(Material and conditions same as in Fig. 17. Photomicrographs ×50; workpiece ×4.)

$$\phi = \tan^{-1}\left[\frac{2\pi\,(2\,R\,d - d^2)^{1/2}}{2\pi\,(R - d) + F_t\,T}\right] \qquad [8]$$

where

ϕ = angle of tooth approach to work, radians
R = radius of cutter, in.
d = depth of cut (instantaneous), in.
F_t = feed per tooth, in.
T = number of teeth in cutter

The depth of cut h, obtaining at this point, is equal to the depth of the tooth mark. It will be seen that this angle at this instant

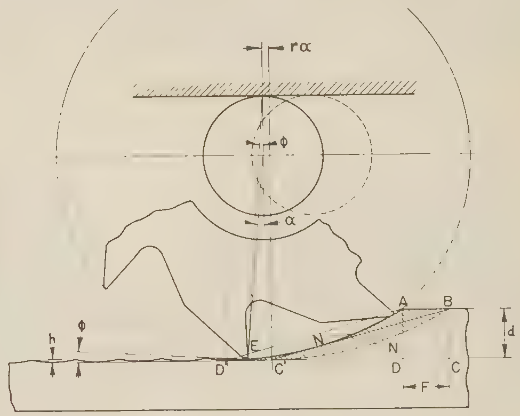


Fig. 24 Angle of Approach of a Milling-Cutter Tooth

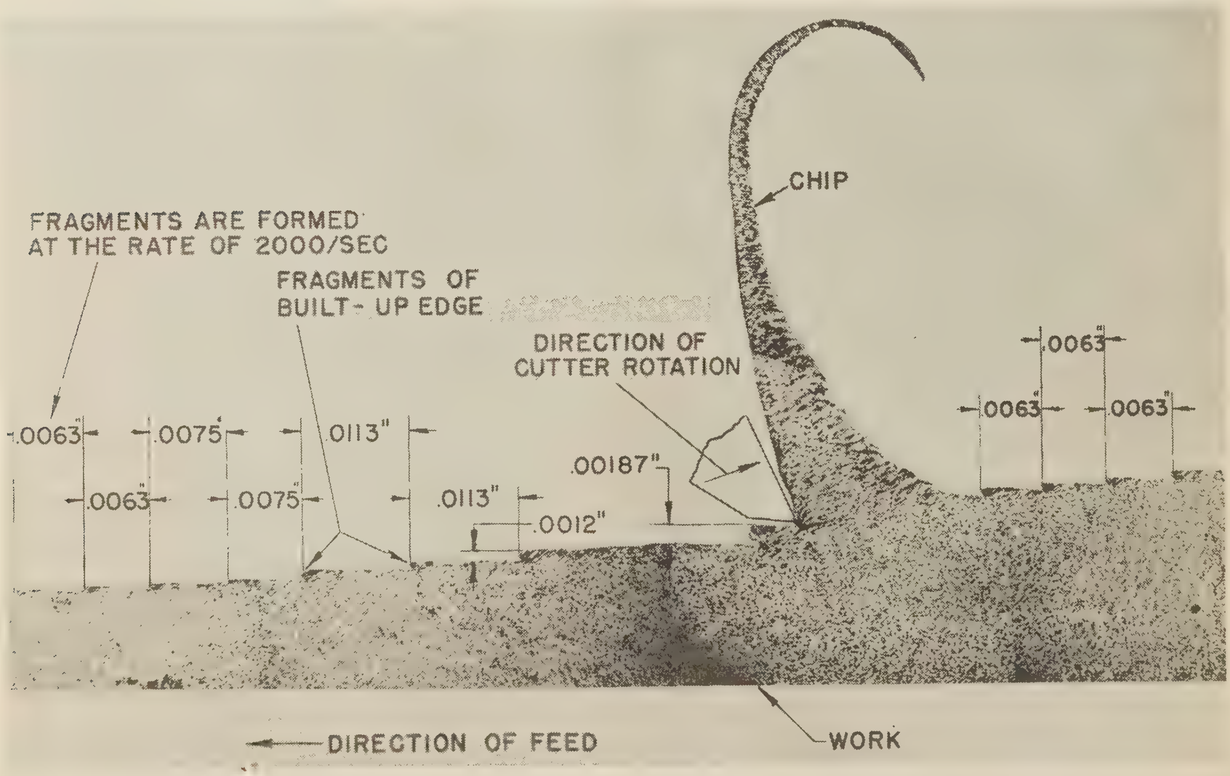


Fig. 23 Photomicrograph of a Milling Chip in Process of Formation
(Material and conditions same as in Fig. 17; ×50.)

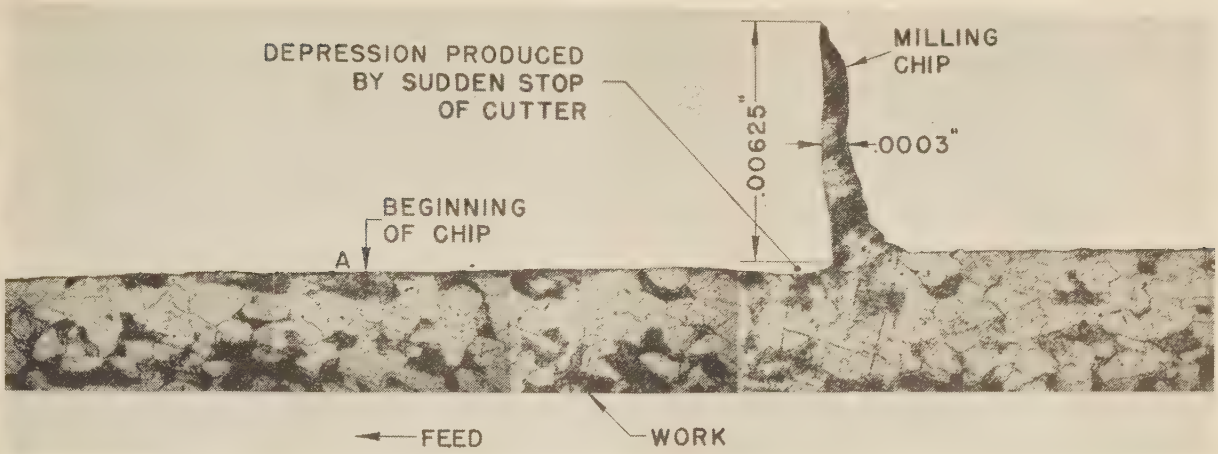


FIG. 25 BEGINNING OF A MILLING CHIP
(Material and conditions same as in Fig. 17; ×600.)

is small but not zero, as is usually believed to be the case. Therefore, a small angular displacement α of the tooth will result with respect to the position C', where the tooth is moving in the direction of the tangent, D'-C'. Incidentally, Equation [8] may be used to determine the angle of approach obtaining at any other point of the tooth path.

For this reason, the tooth approaches and contacts the work while moving in a downward direction, until the point C' is reached, where the upward movement of the tooth begins.

Furthermore, at the beginning of the cut, the thickness of the chip is extremely small, and the length of the cutting edge engaging the work at this instant is usually short for the following reasons:

1 Because the majority of cutters used in milling have either helical teeth with the full length of the cutting edge at a constant distance from the axis of rotation (spiral mills), or straight teeth parallel thereto, but with their cutting edge following a line, the points of which are at variable distances from the center of rotation of the cutter (formed cutters). In either case, there results a gradual engagement of the tooth.

2 In the cutters where the teeth and their respective cutting edges are parallel to the axis of rotation, the length of the cutting edge is inherently limited by their particular use (slotting cutters and saws).

3 Even in those cases (now rare) in which wide straight-tooth milling cutters are used, a full length of contact of the cutting edge at the beginning of the cut can be assured only by grinding the teeth exactly to the same dimensions. Obviously this is a practical impossibility. Therefore, however small the variations in the distance and alignment of each cutting edge relative to the axis of rotation and the fact that each tooth always engages the surface generated by the previous tooth, a limited actual length of engagement with the work at the beginning of the chip will usually result.

4 Since the teeth of a milling cutter are ground to a sharp edge, the area of contact will be exceedingly small. Hence, for materials the hardness of which is within the machinable range, the force applied thereto for producing the intensity of stress needed to cause the material to yield in a plastic manner will also be very small, and penetration of the tooth into the work will readily ensue upon contact.

In view of the foregoing reasons, it is unlikely that sliding of a tooth on the work will ever occur in practice.

It might be argued that, under actual operating conditions, the point of engagement of a tooth is usually very close to the point C', Fig. 24, of maximum distance from the center of the cutter and, therefore, a momentary sliding may be expected.

In the photomicrograph, Fig. 25, in which is shown the cross section of a chip and the adjacent portion of the work at a magnification of 600, there is no evidence whatever of sliding. The minimum thickness of this chip is 0.0003 in. and its length 0.00625 in. This proves, incidentally, that a tooth of a milling cutter can pick up a very thin chip. The beginning of the chip would be placed in the neighborhood of point A for a cutting ratio of 0.5. From an inspection of this illustration, it appears evident that the crystals, of combined carbon and ferrite adjacent to the freshly milled surface, show only very slight indication of local distortion by the action of the cutting edge of the tooth.

In the region A, where the chip began to form, we fail to see a discontinuity in the surface, which may be taken to indicate either sliding or rubbing of the tooth on the work, or sudden digging in, coincident with the picking up of the chip. The chip itself shows, in remarkably clear detail, the crystals composing it. These crystals have maintained their identity intact during the process of being severed from the main body of the work.

The clean-cut outline of the finished surface, and the side of the chip which contacted the face of the tool, suggest that a rather small work of plastic deformation was involved during the formation of this portion of the chip.

Thus, it appears certain that in up milling the cutting edge of a tooth will not slide on the work, but will actually cut as soon as contact between the work and tooth is established.

Clearance and Rake Angle

In analyzing the trochoidal tooth path generated in milling, it was not necessary to consider the shape of the tooth, since in this case it was sufficient to assume a point on the edge thereof at the maximum distance from the geometric center of the cutter.

In actual milling, however, a tooth has definite dimensions, and its geometric shape must conform with definite requirements, among which are the following:

(*a*) The contact between cutter and work must be established on the cutting edge of the teeth, and at no time should interference develop between the body of the tooth and work.

(*b*) On account of requirement (*a*), the flank of a tooth should be located on a plane deviating from the tangent to the periphery of the cutter, by an angle Δ, known as the "clearance angle" or "relief."

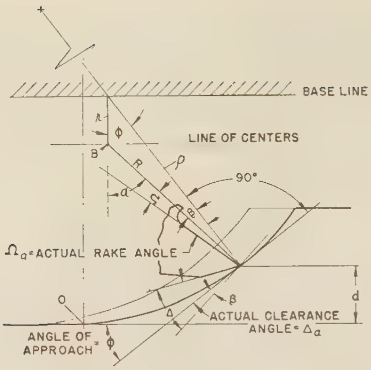


FIG. 26 CLEARANCE AND RAKE ANGLES ALONG PATH OF MILLING-CUTTER TOOTH

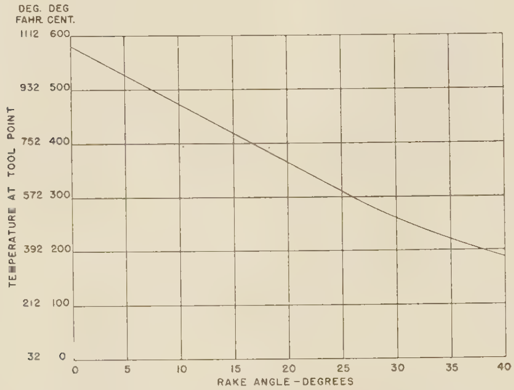


FIG. 27 TEMPERATURE AT TOOL POINT VERSUS RAKE ANGLE
(Data taken from single-point cutting tool, described by Von C. Salomon.[4])

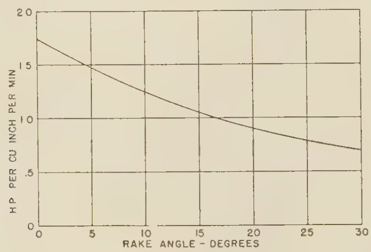


FIG. 28 HORSEPOWER PER CUBIC INCHES PER MINUTE VERSUS RAKE ANGLE[5]

(c) To improve cutting action, the face of the tooth is formed by a plane which deviates from a radial plane by a small angle Ω known as the "rake angle" or "undercut."

(d) The angle included between these two planes (which intersect each other on the imaginary line constituting the theoretical edge) is the tooth angle. This is always less than 90 deg.

The relative position of these angles is illustrated in Fig. 26.

The clearance angle, therefore, is provided for the specific purpose of preventing an interference between the flank of the tooth and the surface of the work. It also has an important function, i.e., that of limiting the area of the flank exposed to the action of the fragments of the built-up edge escaping with the work. It has no particular influence on the power required to remove metal. This angle must be made sufficiently large to take care of the condition of operation of a milling cutter and thus prevent rubbing on any portion of the tooth path.

The rake angle has a determining influence upon the type of chip produced, the pressure, and temperature at the cutting edge of the advancing tooth, and on the power required in the removal of the excess metal. The relationships between the rake angle temperature and power are shown in Figs. 27 and 28.

A large rake angle is known to ease the flow of metal along the face of a tooth, to reduce the cutting temperature, and to favor the formation of a continuous chip with a small built-up edge, and to lessen the rubbing pressure of the chip on the tool and the abrasion resulting therefrom. As a general rule, with an increase in rake angle, a longer tool life, a better finish, and more efficient metal removal may be expected. In practice constant rake angles greater than 20 deg are seldom used, except in the case of aluminum alloys, since the improvement obtained therefrom may be in some cases offset by an early breakage of the cutting edge of the tooth, resulting from the combination of the wearing away of the nose of the tool, higher local temperature, and decarburization of the material of the tool point.

In a milling cutter, the clearance and the rake angles are by no means constant; they are affected by the ever-changing

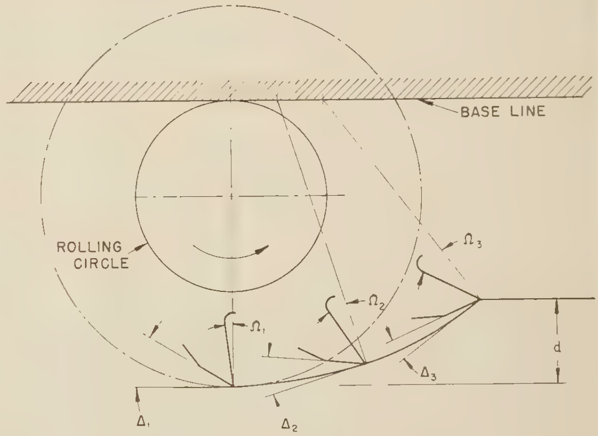


FIG. 29 CHANGES IN CLEARANCE AND RAKE ANGLE WITH POSITION OF TOOTH ON ITS PATH

position of the tooth, and by the curvature of the tooth path. This is illustrated in Fig. 29, showing a tooth in three different positions. It is apparent that the following relation exists among the various angles in the three different positions of the tooth

$$\Omega = \Omega_1 < \Omega_2 < \Omega_3$$
$$\Delta = \Delta_1 > \Delta_2 > \Delta_3$$

where Ω and Δ are the original rake and clearance angles of the cutter.

The actual rake angle or undercut in any point in the tooth path is the angle included between two lines, viz., the parallel to the face of the tooth, and the normal to the trajectory of the tooth at the point under consideration, as is indicated in the case shown in Fig. 26. Since the angle β, included between the radius R of the cutter and the radius of curvature ρ, varies continuously along the path of the tooth in the manner indicated in Equation [9], a variable rake angle will result from the beginning to the end of the engagement of the tooth with the work. Since the teeth of a milling cutter are provided with a rake angle Ω, the actual rake is, therefore, obtained by adding to it the angle β; but

$$\beta = \cos^{-1}\frac{R-d}{R} - \tan^{-1}\left[\frac{2\pi\sqrt{2Rd-d^2}}{2\pi(R-d)+F_t T}\right] \quad \ldots [9]$$

[4] *Die Werkzeugmachine*, December 15, 1929, p. 483.
[5] "Elements of Milling—Part 2," by O. W. Boston and C. E. Kraus, Trans. A.S.M.E., vol. 56, 1938, paper RP-56-1, Fig. 5, p. 358.

hence

Ω_a = actual rake angle = Ω

$$+ \cos^{-1}\frac{R-d}{R} - \tan^{-1}\left[\frac{2\pi\sqrt{2Rd-d^2}}{2\pi(R-d)+F_tT}\right] \ldots [10]$$

By means of Equation [10], it is possible to compute the actual rake angle obtaining under given cutting conditions. It will be found that, in ordinary cases, the increase in the rake angle is of the order of only a few degrees.

The clearance angle is determined by the amount that a tooth path deviates from a circle, Fig. 26, and is measured by the angle β, made by the radius of the cutter and the radius of curvature ρ.

If the path were circular, the radius of curvature would coincide with the radius R of the cutter, and the angle β would be zero. Hence, the minimum angle required is the angle β. This is obtained from the following equation in which the angle β is expressed as a function of the variables of the cut

Minimum clearance angle = β

$$= \cos^{-1}\frac{R-d}{R} - \tan^{-1}\left[\frac{2\pi\sqrt{2Rd-d^2}}{2\pi(R-d)+F_tT}\right] \ldots [11]$$

The actual clearance angle at any point of the tooth path is the result of the difference between the original clearance angle Δ, ground on the flank of a cutter tooth, and the value of the minimum clearance β obtained with Equation [11].

Δ_a = actual clearance angle = Δ

$$+ \tan^{-1}\left[\frac{2\pi\sqrt{2Rd-d^2}}{2\pi(R-d)+F_tT}\right] - \cos^{-1}\frac{R-d}{R} \ldots [12]$$

At the point O, where the tooth is tangent to the path, the minimum clearance angle is zero, and the actual clearance assumes the value of the original clearance angle. As the depth of cut increases, the actual clearance will be reduced by the increasing minimum clearance angle, and eventual rubbing on the flank of the tooth will develop when the two angles are equal.

Therefore, Equation [11] can be used to compute the minimum theoretical clearance angle required for any particular job. To this must be added a small angle to allow for the slight expansion of the material of the newly formed surface, and of the fragments of the built-up edge escaping with the work. It will be found that, on the basis of this formula, the clearance angle will assume values far below those which are now accepted in shop practice.

Tests which have been conducted to prove the validity of this formula, however, confirm the desirability of using small clearance angles of the order indicated by Equation [11]. This has been corroborated by actual application to difficult cutting jobs. Charts based on this equation, showing the clearance angle required for a given cutter diameter, depth of cut, and feed rate, can readily be prepared for use in the shop.

It is advisable to provide the teeth of a milling cutter with the clearance pertaining to given operating conditions rather than to use a comprehensive angle for all possible applications of the cutter.

The changes, occurring in the actual clearance and rake angles in a tooth of a milling cutter as it travels along its path, are important in so far as they affect the quality of finish, the power consumed, and the life of the cutter.

The derivation of Equations [10] and [11] is given in Appendix 1.

Length of Tooth Path

The total length of engagement of the tool with the work, necessary to complete a machining operation, together with the concomitant abrasive action on the cutting edge by the newly generated surface and the fragments of the built-up edge passing off with the work, has a determining influence upon the life of the cutting tool. In milling, only a small fraction of the actual milled surface, approximately equal to the feed per tooth, is used to form the finished surface; the remainder is removed by the tooth following, and is, therefore, completely lost.

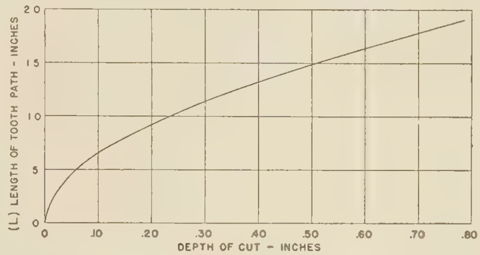


Fig. 30 Relation Between Length of Tooth Path and Depth of Cut

(Diam of cutter, 4 in.; feed rate, 6 in. per min; feed per tooth, 0.021 in.)

Consequently, a minimum actual length of tooth path is a desirable goal. This length is a function of the variables of the cut, as expressed in the equation

$$L = R\cos^{-1}\left(1-\frac{d}{R}\right) + \frac{F_t \times T}{\pi D}(Dd-d^2)^{1/2} \ldots [13]$$

where

L = length of tooth path, in.
d = depth of cut, in.
F_t = feed per tooth, in.
D, R = diameter and radius of cutter, respectively, in.
T = number of teeth in cutter

The derivation of Equation [13] is given in Appendix 2.

For any given condition, in which the depth of cut and the feed rate are known, the only variable on which we can operate to reduce L is the diameter of the cutter and the number of teeth. The feed per tooth, in accordance with the foregoing analysis, should be maintained as large as is consistent with the operating conditions of the job in hand.

A typical example illustrating the results obtained in the application of Equation [13] is shown in Fig. 30. From this chart, we can get some general idea of the advantage obtained in limiting the amount of stock removal to a minimum consistent with the design and the method of manufacturing a given part.

Assuming the cutting conditions, as shown in Fig. 30, and further, that the piece to be machined is 12 in. long, and that the feed per tooth is 0.021 in. with a 10-tooth 4-in-diam spiral mill, and 30-fpm cutting speed, then the total number of engagements made by a single tooth, per piece milled, will be

$$\frac{12}{10 \times 0.021} = 57$$

If the depth of the cut is $^1/_4$ in., then from Fig. 30, the length of tooth path will be 1.04 in. and the total length of surface milled by a tooth will be $57 \times 1.04 = 59.28$ in. If the depth of cut is reduced to $^1/_8$ in., the length of tooth path will be 0.72 in., and then the total length milled per piece will be $57 \times 0.72 = 40.47$ in. This means that the total length of surface milled per tooth in the latter case will be 18.81 in. shorter than the length milled in the former case, under otherwise similar cutting conditions. If the number of pieces milled were 100, then the total saving in milled surface will amount to 1881 in., or approximately 157 ft. Furthermore, by reducing the depth of the cut,

the thickness of the chip (and thus the work done by it) is likewise reduced.

Thickness of Undeformed Milling Chip

Knowing the conditions of the cut and the length of the tooth path, it is now possible to determine the average thickness of the undeformed chip. This can be readily accomplished by considering the equivalence between the areas of the rectangle *A-B-C-D* (the sides of which are equal to the feed per tooth F_t and depth of cut d), the parallelogram *A-B-C′-D′*, and the geometric figure *A-N-D′-C′-N-B*. This area, less the area of the triangle *D′-E-C′* is equal to the area *E-A-B*, which is the area of the cross section of the undeformed chip, Fig. 24.

However, the area of the triangle *D′-E-C′* is nearly equal to $\frac{h \times F_t}{2}$ and, due to the very small value of the quantity h, as determined by Equation [5], we can disregard the effects of this area on the total area of the undeformed chip. Consequently, we can write

Area of undeformed chip section =

$$(A\text{-}B\text{-}E) = F_t \times d \qquad [14]$$

Average undeformed chip thickness =

$$t_{\text{avg}} = \frac{F_t \times d}{L} \qquad [15]$$

where L is the length of tooth path, as given in Equation [13].

The variation of the average undeformed chip thickness obtained in a specific case, and as a function of the depth of cut, is shown in Fig. 31.

The average thickness of the undeformed chip can be used to advantage in determining the cutting ratio, namely, the ratio between the length of the actual chip and the length of tooth path, and in estimating the amount of metal removal by a single tooth. Since, in the measurement of either the power or the force involved in a cutting process, average values are generally obtained, it appears more logical to refer these values to the average thickness of the undeformed chip, which is a function of all the variables of the cut, rather than to only one of these variables, as in the case shown in Fig. 10. Here, the efficiency of metal removal is shown plotted against the feed per tooth. These data, replotted on the basis of the undeformed cross section of the chip, are shown in Fig. 32.

By plotting the efficiency of metal removal (cubic inches per minute per horsepower input to the machine) against the average chip thickness, t_{avg}, the difference between the efficiency obtaining in two extreme cases, such as $^1/_{32}$ in. and $^1/_2$ in. depth of cut, is considerably less than on the basis of the feed per tooth F_t. If the efficiency of metal removal is computed on the basis of power consumed at the cutter, then the efficiency obtaining under the various cases represented in Fig. 33 tends to disappear and a single curve could be used to cover the range of depth of cut shown therein. In this case, the average chip thickness could be used in establishing the relationship between the conditions of the cut and the corresponding power required.

This was done for the values shown in Fig. 33, and the following empirical equation was obtained

$$E = 13.2\, t_{\text{avg}}^{0.417} \qquad [16]$$

where

E = metal removal, cu in. per min per hp at cutter

t_{avg} = average thickness of undeformed cross section of chip, in.

Instantaneous Thickness of Undeformed Milling Chip

The instantaneous radial thickness of the undeformed cross section of the chip, however, is one of the basic elements of the milling process. This is a segment intercepted by two consecutive tooth paths on the normal to the point of the tooth path under consideration. In the case of a circular tooth path, Fig. 34, the instantaneous radial thickness *MN* is a segment of the radius of the cutter, while in the case of a trochoidal tooth path,

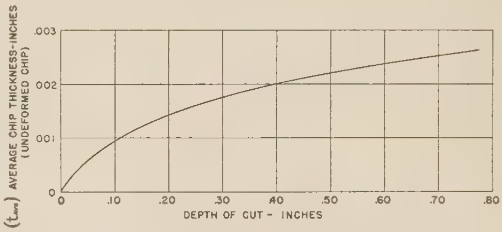


Fig. 31 Average Chip Thickness (t_{avg}) Versus Depth of Cut
(Diameter of cutter, 2 in.; feed rate, 6 in. per min; feed per tooth, 0.0046 in.)

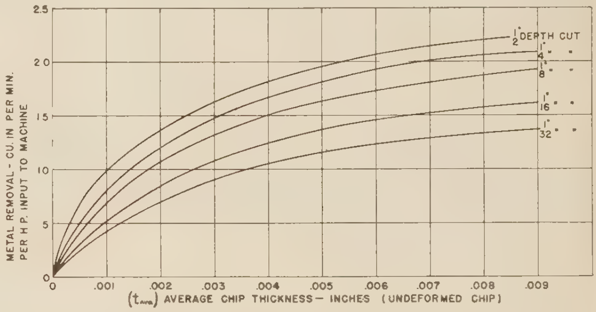


Fig. 32 Metal Removal, Cubic Inches per Minute per Horsepower Input to Machine Versus Average Chip Thickness (t_{avg})
(Material, cast iron; width of cut, 4 in.; cutter, spiral mill, 8T, 4 in. diam; cutting speed, 55 fpm.)

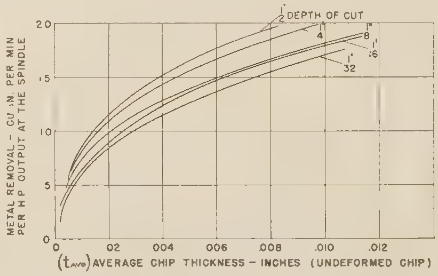


Fig. 33 Metal Removal, Cubic Inches per Minute per Horsepower Output at Spindle Versus Average Chip Thickness (t_{avg})
(Conditions same as in Fig. 32.)

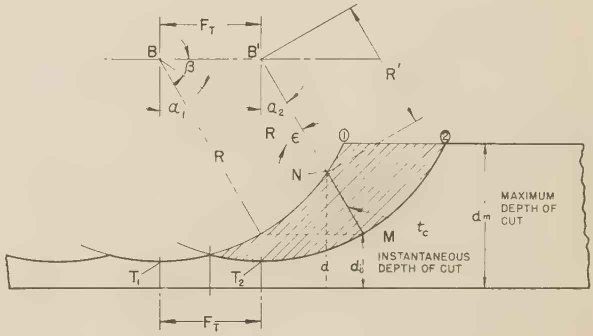


Fig. 34 Instantaneous Thickness (t_c) of Undeformed Cross Section of Chip
(Tooth path approximated to arc of circle.)

Fig. 35, the segment $T_1'' T_2'$ is part of the radius vector ρ_2 (on the radius of curvature) normal to the point T_2' and limited by the instantaneous center of rotation C_1 and the point T_2' on the tooth path.

The corresponding analytical expression, for the case in which the path of a tooth is assumed to be circular, is known, but will be derived here in the form which has been found to be more suitable for comparison with that for the trochoidal tooth path.

From the geometry of Fig. 34, and for the position M of tooth 2 therein indicated, the following equation has been derived for the thickness t_c of the cross section of the undeformed chip with the assumed circular path

$$t_c = R + F_t \sin \alpha_2 - (R^2 - F_t^2 \cos \alpha_2^2)^{1/2} \quad [17]$$

The derivation of Equation [17] is given in Appendix 3. The angle α_2 appearing in this equation gives the angular displace-

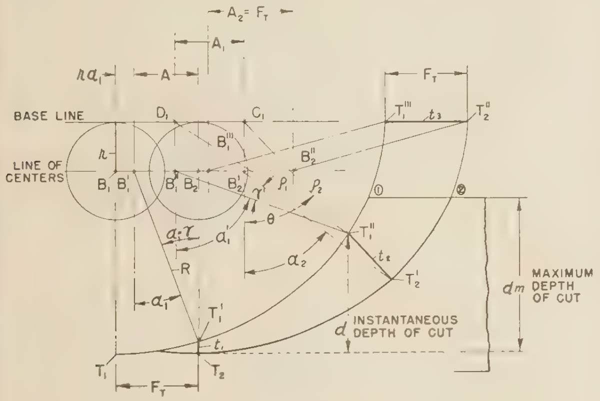


FIG. 35 INSTANTANEOUS THICKNESS (t) OF UNDEFORMED CROSS SECTION OF CHIP
(Actual tooth path.)

ment of the tooth 2 with respect to the vertical drawn from the center B' and corresponding to the position M, of the tooth.

This angle can be obtained from the known data as follows

$$\alpha_2 = \sin^{-1} \frac{(2 R d - d^2)^{1/2} - F_t}{[R^2 - 2 F_t(2 R d - d^2)^{1/2} + F_t^2]^{1/2}} \quad [18]$$

where

R = radius of cutter, in.
d = instantaneous depth of cut at point N, in.
F_t = feed per tooth, in.

It has been found, however, that for the values obtaining in practical milling the expression under the radical in Equation [17] is approximately equal to R^2, hence a simplified form of this equation is

$$t_c = F_t \sin \alpha_2 \quad [19]$$

Substituting in this equation the expression for the angle α_2 obtained in Equation [18], the following equation for the thickness t_c is obtained

$$t_c = F_t \frac{(2 R d - d^2)^{1/2} - F_t}{[R^2 - 2 F_t(2 R d - d^2)^{1/2} + F_t^2]^{1/2}} \quad [20]$$

A similar equation can be derived for the thickness t of the undeformed cross section of the chip, obtaining when the true path of the milling-cutter tooth is considered. From the conditions represented in Fig. 35, and for the position T_2' of the tooth 2, the expression for the thickness at this point t_2 is

$$t_2 = \rho_2 + A_1 \sin \theta - (\rho_1^2 - A_1^2 \cos \theta^2)^{1/2} \quad [21]$$

This is a more general equation of the thickness of the undeformed cross section of the chip, since it includes the approximate case of the circular tooth path. In fact, Equation [17] can be obtained directly from Equation [21] by eliminating, in this equation, the terms containing r, the pitch radius of the pinion. In Equation [21], the thickness t_2 is expressed as a function of the distance ρ_2 from the center of the instantaneous rotation C_1 to the point T_2' on the path of the tooth (2), the distance ρ_1 from the instantaneous center of rotation D_1 to the point T_1'' of the path of the tooth (1). These segments are, therefore, normal to the tangent to the path at their respective points T_2' and T_1''. The thickness t_2 is also a function of the angle θ made by ρ_2 with the vertical drawn from the center of instantaneous rotation C_1, and of the quantity A_1, which is the distance between the center D_1 and C_1.

When Equation [21] is used within the limits of practical milling operations, it can be simplified without greatly affecting the values obtained therewith. It will be found, therefore, that the term under the radical sign may be assumed equal to ρ_1^2 and that ρ_1 can be taken equal to ρ_2. Hence Equation [21] will assume the form

$$t_2 = A_1 \sin \theta \quad [22]$$

but, since

$$\sin \theta = \frac{R \sin \alpha_2}{\rho_2}$$

it follows that

$$t_2 = \frac{A_1 R \sin \alpha_2}{\rho_2} \quad [23]$$

The quantity ρ_2 is a function of the variable of the cut, as indicated in the equation

$$\rho_2 = (r^2 + R^2 + 2 r R \cos \alpha_2)^{1/2} \quad [24]$$

In Equation [23] the thickness t_2 is expressed as a function of the variables of the cut in which are included the angles α_1 and α_2. The independent variables in a practical case are the feed rate, depth of cut, and diameter of the cutter. Of these variables the one which determines the value of the angle α_1 and α_2 is the depth of cut (refer to parametric Equation [2]). At any instant, this is given by the maximum vertical distance from the point T_1'' to the finished surface of the work. Eventually this will assume the value dm, the actual depth of cut, which corresponds to the maximum thickness of the undeformed cross section of the chip. When the distance d of point T_1'' is used as the independent variable the angle α_1 can be determined. It is, however, necessary to determine the relationship between the angle α_1 and the angle α_2. An approximate relationship between these two angles can be found by considering the change occurring in both angles as the tooth 2 travels along its path.

For the position T_2 of the tooth 2, the angle α_2 is zero, while the angle α_1 has assumed a value determined by the angular displacement required to bring the radius R from the position T_1 to T_1'. Coincident with the rotation of radius R, there is a translation of the center B_1 of the cutter to B_1' along the line of centers by an amount $r\,\alpha_1$, resulting from the motion of the pinion on the base line. The value of α_1 at this particular instant can be readily determined from the parametric Equations [2] by assuming that $\sin \alpha_1$ is equal to the arc α_1, and this in turn equal to γ. The justification for this assumption is found in the fact that the angle α_1 is, in this case, very small. After making the necessary substitution and solving for γ there results

$$\gamma = \frac{F_t}{r + R} \quad [a]$$

The distance A between centers B_1' and B_2, which was originally equal to F_t, is now

$$A = F_t - r\alpha \quad \text{[b]}$$

As tooth 2 proceeds along its path, the angle γ changes and, for the new position B_1'' and B_2', the centers B_1 and B_2 of the cutter assume a new value γ', which is equal to the difference between α_1' and α_2

$$\gamma' = \alpha_1' - \alpha_2 \quad \text{[c]}$$

When tooth 2 has reached the position T_2'', α_1 and α_2 are equal, hence γ vanishes. Therefore, the following positions are possible

$$\left.\begin{array}{lll} \alpha_2 = 0 & \gamma = \alpha_1 & A = F_t - r\,\alpha_1 \\ \alpha_2 = \alpha_2 & \gamma' = \alpha_1' - \alpha_2 & A_1 = F_t - r(\alpha_1' - \alpha_2) \\ \alpha_2 = \alpha_1 & \gamma'' = 0 & A_2 = F_t \end{array}\right\} \quad \text{[d]}$$

Assuming a linear variation for the angle γ between the point T_2 and T_2'' in which the angle α_2 varies from zero at T_2 to approximately $\pi/2$ at T_2'', any intermediate value γ' of γ, will be obtained as follows

$$\gamma' = \gamma\left(\frac{\pi - 2\,\alpha_2}{\pi}\right) \quad \text{[e]}$$

But from Equation [c], and with the further approximation $\pi = 3$, there results

$$\alpha_2 = \frac{\alpha_1' - \gamma}{1 - {}^2/_3\gamma} \quad \text{[f]}$$

The angle α_1' which appears in this equation is a function of the independent variable d (instantaneous depth of cut) and is obtained by solving the second equation of the system [2]. Hence

$$\alpha_1' = \cos^{-1}\frac{R - d}{R} \quad \text{[g]}$$

Therefore, from Equations [a], [g], and [f], the value of α_2 in radians is obtained

$$\alpha_2 = \frac{(r + R)\cos^{-1}\dfrac{R - d}{R} - F_t}{r + R - \dfrac{2}{3}F_t} \quad \text{[25]}$$

By making the necessary substitution in Equation [d], the quantity A can be expressed as a function of the given variables of the cut

$$A_1 = F_t\left(\frac{R + \dfrac{2}{3}\,r\cos^{-1}\dfrac{R - d}{R} - \dfrac{2}{3}F_t}{R + r - \dfrac{2}{3}F_t}\right) \quad \text{[26]}$$

Returning now to Equation [23], it is possible to substitute for the quantity ρ_2, α_2, and A_1 their corresponding expressions, Equations [24], [25], and [26], which have been obtained in the foregoing analysis. The result is

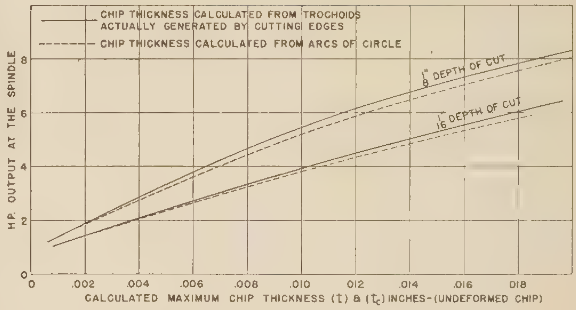


Fig. 36 Horsepower Output at Spindle Versus Maximum Thickness (t) and (t_c) of Undeformed Cross Section of Chip
(Material, cast iron; width of cut, 4 in.; cutter, spiral mill, 8T, 4 in. diam; cutting speed, 63 fpm.)

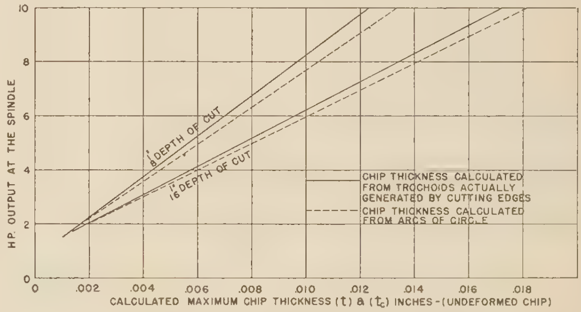


Fig. 37 Horsepower Output at Spindle Versus Maximum Thickness (t) and (t_c) of Undeformed Cross Section of Chip
(Material, S.A.E. 1112; width of cut, 4 in.; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 63 fpm; cutting fluid, soluble oil and water.)

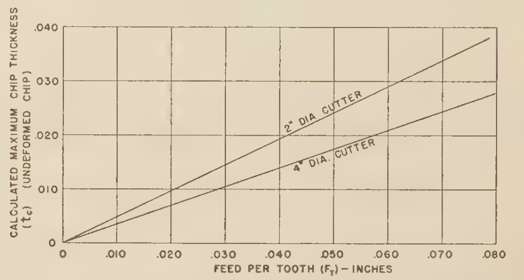


Fig. 38 Maximum Thickness (t_c) of Undeformed Cross Section of Chip Versus Feed per Tooth (F_t)
(Cutter, spiral mill, 8T, 4 in. diam; depth of cut, 1/8 in.; cutting speed, 63 fpm; cutter, spiral mill, 4T, 2 in. diam; cutting speed, 63 fpm.)

The results of the actual power consumed at the cutter with a standard spiral mill on cast iron and S.A.E. 1112 steel have been plotted in Figs. 36 and 37, against the radial thickness t_c and t of the undeformed section of the chip, obtained from Equations [20] and [27], respectively.

On the basis of equal power consumed, it is only when the feed

$$t_2 = \frac{R\,F_t\left[\dfrac{R + \dfrac{2}{3}\left(r\cos^{-1}\dfrac{R-d}{R} - F_t\right)}{R + r - \dfrac{2}{3}F_t}\right] \times \sin\left[\dfrac{180}{3.14}\left(\dfrac{(r+R)\cos^{-1}\dfrac{R-d}{R} - F_t}{r + R - \dfrac{2}{3}F_t}\right)\right]}{\left\{r^2 + R^2 + 2\,r\,R\cos\left[\dfrac{180}{3.14}\left(\dfrac{(r+R)\cos^{-1}\dfrac{R-d}{R} - F_t}{r + R - \dfrac{2}{3}F_t}\right)\right]\right\}^{1/2}} \quad \text{[27]}$$

rate is unusually high that the difference between t and t_c and, consequently, the error in evaluating the thickness, will be increased if Equation [20] is used.

Although the difference between t_c and t becomes increasingly greater as the feed rate is increased, this does not justify the use of the more complicated expression for t_c given by Equation [27]. Therefore, it is suggested that the simpler Equation [20] be used in practical application. Particularly appropriate is this choice when we consider that in actual practice the values of t range between 0.001 and 0.01 in. In this region, the error in the evaluation of t is negligible.

It is customary to use the feed per tooth as a criterion in estimating the tooth load on a milling cutter. This, however, in the light of the information contained in Equations [20] and [27], supplemented by the example illustrated in Fig. 38, leads to incorrect appreciation of the actual thickness of the metal being removed by a tooth.

The feed per tooth is determined by the feed rate of the work, the number of teeth, and revolutions per minute of the cutter, Equation [7]. As long as these are maintained constant, a change either in the depth of cut or the diameter of the cutter will not affect the feed per tooth; a change, however, will result in the radial thickness of the undeformed section of the chip.

Moreover, when the feed per tooth is increased, the corresponding radial thickness t_c becomes progressively smaller than the feed per tooth. The foregoing conditions are illustrated in Fig. 38. Since the radial thickness t_c determines the depth of the section of metal which is directly under the action of the advancing tooth, this is a truer indication of the work involved during the process of forming a chip than the feed per tooth.

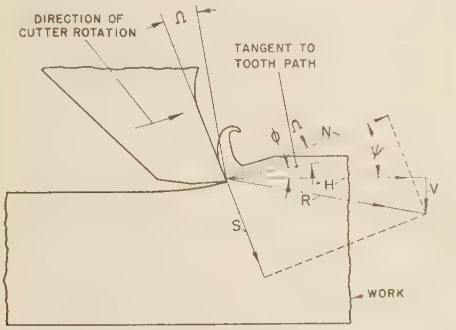


Fig. 39 Force Diagram Showing Relationship of Forces at Edge of a Milling-Cutter Tooth

Horizontal and Vertical Components of the Cutting Force

Of particular interest is the relationship between the radial thickness t_c and the experimentally determined average values of the components of the cutting force R in the direction of the feed and normal thereto.

In the vector diagram, Fig. 39, showing a milling-cutter tooth in the process of forming a chip, these forces are indicated by vectors H and V, respectively, and they are shown as applied to the work.

Their resultant R can be solved in two components, S and N, parallel and normal to the face of the tooth, respectively.

From the known value of R and knowing the angle ψ made by N with the horizontal, it is possible to determine the intensity of N and S.

The angle ψ, however, is equal to the sum of the angle ϕ (Equation [8]) made by the tangent to the tooth path with the horizontal and the actual rake angle Ω, both of which vary with the position of the tooth along its path.

Since the intensities of the vertical and horizontal components of R are average values, it is therefore sufficient in many cases to assume also an average value for ψ.

Hence, the corresponding intensities of N and S could be computed from the known values of H and V. These, for the case of slab milling on cast iron and S.A.E. 1112 steel specimens, are given in Figs. 40 and 41, respectively. The values of the forces were obtained by calibrating with known loads the structure of the milling machine on which these tests were conducted. This method offered the advantage of operating under conditions

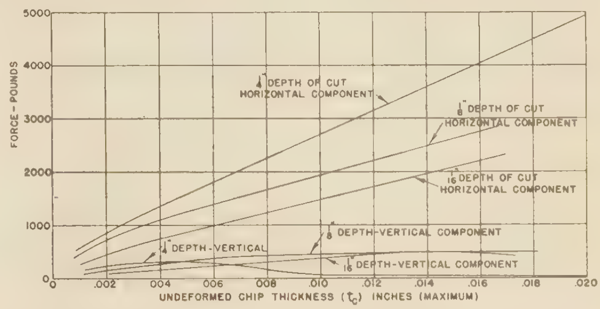


Fig. 40 Horizontal and Vertical Components of Cutting Force Versus Thickness (t_c) of Undeformed Cross Section of Chip

(Material, cast iron; width of cut, 4 in.; cutter, spiral mill, 10T, 4 in. diam; cutting speed, 63 fpm.)

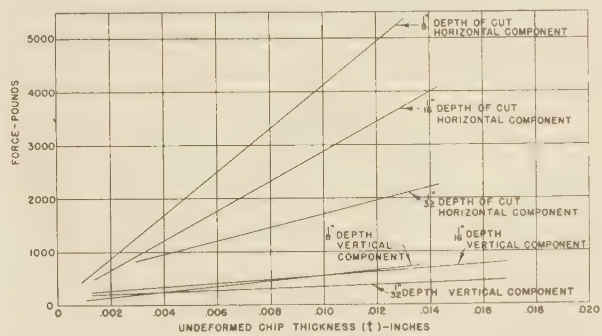


Fig. 41 Horizontal and Vertical Components of Cutting Force Versus Thickness (t_c) of Undeformed Cross Section of Chip

(Material, S.A.E. 1112; width of cut, 4 in.; cutter, spiral mill, 10T, 4 in diam; cutting speed, 63 fpm; cutting fluid, soluble oil and water.)

identical with those obtaining in practice and, furthermore, of increasing the reliability and accuracy of the results by the simplicity and rigidity of the setup.

The values obtained indicate that:

(a) With good approximation the horizontal and vertical components of the cutting force are proportional to the maximum thickness t of the undeformed cross section of the chip.

(b) The horizontal component increases rapidly with the depth of cut.

(c) The intensity of the vertical component is considerably less than that of the horizontal component. Furthermore, the vertical component shows a marked tendency to remain unchanged at shallow depth of cut and eventually to decrease and possibly become negative with deeper cuts.

In the case of $^1/_4$ in. depth of cut on cast iron, Fig. 40, this decrease of the vertical component begins with a value of $t_c = 0.006$ in.

For depths of cut above $^1/_8$ in., a reversal of the direction of the vertical component may take place as the tooth approaches the end of its travel through the work.

Conclusions

In the analysis presented, it has been shown that the path of a milling-cutter tooth is an arc of a trochoid, the parametric equations for which can be derived from known variables of the cut.

Thus the milling process may be approached from a mathematical viewpoint. This is particularly helpful in the evaluation of such elements as the radius of curvature of the tooth path, the clearance and rake angles of the length of the tooth path, the radial thickness of the chip, and their effects upon the quality of milled surface, power consumed, and cutter life. This method of analysis, supplemented with actual photomicrographs of surface and chip, has been used to prove that a tooth of a milling cutter will not normally slide on the work but will actually penetrate immediately upon contact.

Furthermore, it has been shown that, in the up-milling process a good quality of machined surface is usually obtained.

When comparing different milling methods, it will be found that a better perspective of the advantages and disadvantages inherent in each method will be obtained by resorting to a more accurate determination of such elements as the length of tooth path, the actual clearance and rake angles, and thickness of the undeformed cross section of the chip.

In metal-cutting problems, it is always necessary to weigh a number of factors in relation to the desired result; this may be either economical operations of the machine, long tool life, or quality of finish on the machined surface. The proper solution of this problem may be greatly aided by a better understanding of the inherent operational functions of the basic elements of the process chosen for carrying out a given machining operation.

Acknowledgment

The author wishes to express his appreciation to Mr. Hans Ernst, research director; to Mr. J. C. Campbell, laboratory engineer, who conducted the tests and assisted in the preparation of the charts; and to Mr. E. Suhr, laboratory assistant, for the preparation of the specimens and relative photomicrographs.

Appendix 1

The derivation of an expression for rake and clearance angles is given as follows:

In the instantaneous position of the tooth, shown in Fig. 26, the actual rake angle is

$$\Omega_a = \Omega + \beta$$

But Ω is the known rake angle of the tooth and

$$\beta = \alpha - \phi$$

From Equation [2]

$$\alpha = \cos^{-1}\frac{R - d}{R}$$

and ϕ is the slope of the tangent to the curve, given by

$$\tan\phi = \frac{dy/d\alpha}{dx/d\alpha}$$

Differentiating Equation [2] with respect to the parameter α, and then replacing y with d

$$\tan\phi = \frac{\sqrt{2\,R\,d - d^2}}{r + (R - d)}$$

Substituting for r, the expression obtained from Equations [1] and [7]

$$\tan\phi = \frac{2\,\pi\sqrt{2\,R\,d - d^2}}{2\,\pi\,(R - d) + F_t\,T}$$

and

$$\phi = \tan^{-1}\frac{2\,\pi\sqrt{2\,R\,d - d^2}}{2\,\pi\,(R - d) + F_t\,T}$$

Therefore

$$\beta = \cos^{-1}\frac{R - d}{R} - \tan^{-1}\frac{2\,\pi\sqrt{2\,R\,d - d^2}}{2\,\pi\,(R - d) + F_t\,T}$$

The actual rake angle is

$$\Omega_a = \Omega + \cos^{-1}\frac{R - d}{R} - \tan^{-1}\frac{2\,\pi\sqrt{2\,R\,d - d^2}}{2\,\pi\,(R - d) + F_t\,T}$$

Since the actual clearance angle is

$$\Delta_a = \Delta - \beta$$

it follows

$$\Delta_a = \Delta + \tan^{-1}\frac{2\,\pi\sqrt{2\,R\,d - d^2}}{2\,\pi\,(R - d) + F_t\,T} - \cos^{-1}\frac{R - d}{R}$$

Appendix 2

The derivation of the length of tooth path is as follows:

The differential of the arc of trochoid from parametric Equations [2] is

$$ds = \sqrt{(r + R\cos\alpha)^2 + R^2\sin^2\alpha} \times d\alpha$$

also

$$ds = \sqrt{r^2 + R^2}\sqrt{1 + \frac{2\,r\,R\cos\alpha}{r^2 + R^2}} \times d\alpha$$

Developing $\sqrt{1 + \frac{2\,R\,r\cos\alpha}{r^2 + R^2}}$ in binomial series, and disregarding the terms of higher power than the first

$$ds = \sqrt{r^2 + R^2}\left[1 + \frac{r\,R}{R^2 + r^2}\cos\alpha\right] \times d\alpha$$

Integrating between the limits $\alpha = 0$, $\alpha = \alpha$

$$L = \int_0^{\alpha}\sqrt{r^2 + R^2}\left[1 + \frac{r\,R}{R^2 + r^2}\cos\alpha\right] \times d\alpha$$

$$= \sqrt{r^2 + R^2} \times \alpha + \frac{r\,R}{\sqrt{r^2 + R^2}}\sin\alpha$$

If r^2 is disregarded, then

$$L = R\,\alpha + r\sin\alpha$$

From Equation [2]

$$L = R\cos^{-1}\frac{R - d}{R} + \frac{r}{R}\sqrt{2\,R\,d - d^2}$$

Appendix 3

The derivation of Equation [17] is as follows:
From Fig. 34

$$t_c = R - R'$$

$$R' = \frac{R \sin (90 - \alpha_1)}{\cos \alpha_2} = \frac{R \cos \alpha_1}{\cos \alpha_2}$$

But

$$\alpha_1 - \alpha_2 = \epsilon$$

and

$$\alpha_1 = \epsilon + \alpha_2$$

therefore

$$R' = \frac{R \cos (\alpha_2 + \epsilon)}{\cos \alpha_2} = R \frac{\cos \alpha_2 \cos \epsilon - \sin \alpha_2 \sin \epsilon}{\cos \alpha_2}$$

$$= R \left(\cos \epsilon - \frac{\sin \alpha_2 \sin \epsilon}{\cos \alpha_2} \right)$$

Since

$$\sin \epsilon = \frac{F_t \cos \alpha_2}{R}$$

$$\cos \epsilon = \sqrt{1 - \frac{F_t^2 \cos^2 \alpha_2}{R^2}} = \frac{1}{R} \sqrt{R^2 - F_t^2 \cos^2 \alpha_2}$$

hence

$$R' = R \left(\frac{1}{R} \sqrt{R^2 - F_t^2 \cos^2 \alpha_2} - \frac{F_t \sin \alpha_2 \cos \alpha_2}{R \cos \alpha_2} \right)$$

$$= \sqrt{R^2 - F_t^2 \cos^2 \alpha_2} - F_t \sin \alpha_2$$

and

$$t_e = R + F_t \sin \alpha_2 - \sqrt{R^2 - F_t^2 \cos^2 \alpha_2}$$

The derivation of Equation [21] follows:
From Fig. 35

$$t_2 = \rho_2 - \rho_2' \quad \text{where} \quad \rho_2' = C_1 T_1''$$

but

$$\rho_2' = \frac{\rho_1 \sin \beta}{\sin (90 + \theta)}$$

and

$$\beta = 180 - \{90 + \theta + \epsilon\} = 90 - (\theta + \epsilon)$$

where

$$\epsilon = C_1 T_1'' D_1 \text{ and } \beta = T_1'' D_1 C_1$$

therefore

$$\rho_2' = \frac{\rho_1 \cos (\theta + \epsilon)}{\cos \theta}$$

Upon substitution

$$t_2 = \rho_2 - \rho_1 \frac{\cos (\theta + \epsilon)}{\cos \theta} = \rho_2 - \rho_1 \cos \epsilon + \rho_1 \frac{\sin \epsilon \times \sin \theta}{\cos \theta}$$

Since

$$\frac{A}{\sin \epsilon} = \frac{\rho_1}{\cos \theta}$$

$$\sin \epsilon = \frac{A \cos \theta}{\rho_1}$$

and

$$\cos \epsilon = \sqrt{1 - \frac{A^2 \cos^2 \theta}{\rho_1^2}}$$

Results

$$t_2 = \rho_2 + A \sin \theta - \sqrt{\rho_1^2 - A^2 \cos^2 \theta}$$

Discussion

O. W. Boston.[6] This analysis of the actual curvature of travel by the tooth point of a plain milling cutter as it removes a chip, having a definite depth of cut and feed as it deviates from a true circular path, is very interesting. Such analyses have been made previously by Dr. Eng. C. Salomon,[7] who presents a new theory of metal removal by milling, and by N. N. Sawin.[8]

For all cutters which cut on the periphery of the tooth, the length of the chip removed by each tooth depends principally upon its diameter and the depth of cut, as the feed per tooth is relatively very small in comparison. Fig. 42 of this discussion shows an end view of plain- or side-milling cutters removing chips

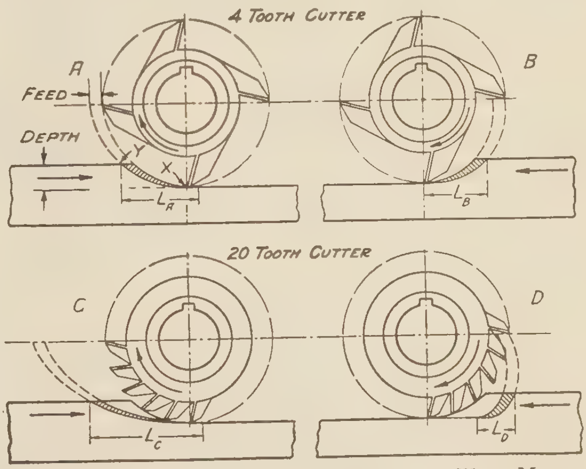


Fig. 42 Graphical Analysis of Chip Formation When Milling Up and Milling Down With Coarse- and Fine-Tooth Cutters

(With thick chips removed more efficiently, from a power standpoint, than thin chips, it is seen that the 4-tooth cutter is more efficient when cutting down at B than when cutting up. It is seen, also, that the fine-tooth cutter is more efficient when cutting down at D than when cutting up. Further, it is seen that the fine-tooth cutter when cutting down is more efficient than the coarse-tooth cutter when cutting down, the feed per tooth in all cases being the same.)

by the peripheral cutting edges of the teeth. The cross-hatch area at A indicates the sectional area of the chip removed by a single tooth. The feed per tooth, exaggerated, and the depth of cut are indicated. The tooth comes in contact with the surface of the work at the point X and leaves the work at the point Y. The curve XY is that discussed by the author. The tooth point rubs over the surface of the work at the point X as the material is fed to the right, as indicated by the arrow, while the cutter rotates clockwise until the normal force between the tooth and the work is sufficient to cause the cutting edge to dig in. If the cutting edge is dull, a considerable force between the work and cutter

[6] Professor of Metal Processing, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[7] "Die Fräsarheit," by C. Salomon, *Werkstattstechnik*, vol. 20, 1926, pp. 469–474.

[8] "Theory of Milling Cutters," by N. N. Sawin, *Mechanical Engineering*, vol. 48, 1926, pp. 1203–1209.

Note: The articles of ref. (7) and (8) are abstracted in "Bibliography on the Cutting of Metals," A.S.M.E. 1930.

is necessary before a chip starts to be formed. The thickness of the chip at right angles to the path of the cutting edge is practically zero at the point X, and reaches maximum at the point Y. The cross-sectional area of the chip removed is equal to the product of f, the feed per tooth in inches, multiplied by d, the depth of cut in inches. If the diameter of the cutter in inches is D, the average thickness of the chip removed is

$$\text{A.T.C.} = f\sqrt{\frac{d}{D}\left(1-\frac{d}{D}\right)}$$

The horizontal length of the cut as indicated by $L_A = \sqrt{d(D-d)}$. From the A.T.C. formula, it is found that the average thickness of chip having a feed equal to f, and a depth equal to d, is less than the average thickness of chip having a feed equal to $2f$ and a depth equal to $\frac{d}{2}$, even though the cross-sectional areas of the two chips are equal.

The author has used average thickness of chip as a basis for comparing power or energy determinations. This was previously done by Parsons.[9] The writer has proved to his satisfaction that A.T.C. is not a good criterion for such a basis, but rather that a formula for energy in foot-pounds per chip is $E = C\,w f^x d^y$, where C is a constant and w is the width of cut in

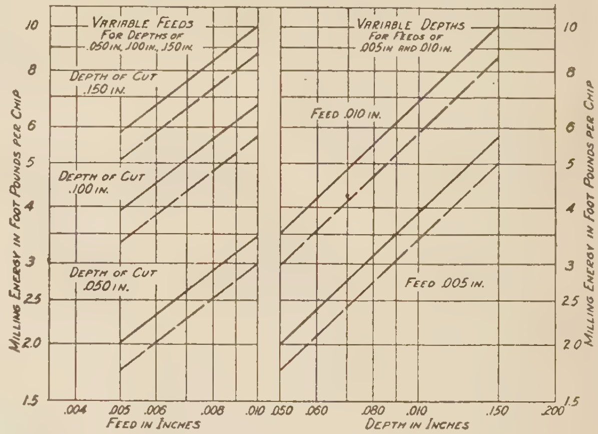


FIG. 43 ENERGY AT TOOL POINT REQUIRED IN FOOT-POUNDS PER CHIP FOR VARIOUS FEEDS AND DEPTHS OF CUT

(Operation of milling brass with lard oil and a side-milling cutter 0.347 in. wide, 3.5 in. diam, and 15-deg rake; when milling up, solid line, and when milling down, dashed line.)

[9] "Power Required for Cutting Metal," by Fred Parsons, Trans. A.S.M.E., vol. 45, 1923, pp. 193–227.

TABLE 2 NET ENERGY AND HORSEPOWER FORMULAS, WITH VALUES OF THE CONSTANT C, FOR MILLING DIFFERENT MATERIALS, BOTH UP AND DOWN, WITH VARIOUS CUTTING FLUIDS

(Formula: $E = Cwf^x d^y$, in which, C = constant for cutter, material, and cutting fluid; E = energy in foot-pounds per chip; w = width of cutter in inches having 15-deg. rake; f = feed in inches per chip; and d = depth of cut in inches)

Material cut	Formulas: Energy, ft-lb. per chip	Formulas: Hp. per cu. in. per min.	Hp./cu. in./min.[a] Up	Hp./cu. in./min.[a] Down	Oil[b] No.	Values of C Up	Values of C Down
S.A.E. 1020 steel, A-1	$E = Cwf^{0.64} d^{0.78}$ both up and down	$= \frac{C}{33{,}000\, f^{0.36} d^{0.22}}$	1.388	1.169	1	5520	4650
			1.084	0.952	4	4320	3790
			1.000	0.796	5	3980	3170
			1.084	1.043	6	4320	4150
			1.043	0.980	8	4150	3900
			0.930	0.854	10	3700	3400
Cold-rolled low-carbon steel	Up: $E = Cwf^{0.73} d^{0.87}$	$= \frac{C}{33{,}000\, f^{0.27} d^{0.13}}$ (Up)	0.936	....	5	6800	..
	Down: $E = Cwf^{0.67} d^{0.87}$	$= \frac{C}{33{,}000\, f^{0.33} d^{0.13}}$ (Down)	....	0.775	5	..	4260
S.A.E. 3150 steel (KF)	$E = Cwf^{0.70} d^{1.00}$	$= \frac{C}{33{,}000\, f^{0.30}}$	1.327	1.278	1	11000	10600
			1.180	1.082	10	9780	8980
S.A.E. 6140 steel (A-13)	$E = Cwf^{0.72} d^{0.90}$	$= \frac{C}{33{,}000\, f^{0.28} d^{0.10}}$	1.44	1.52	1	10630	11220
			1.348	1.427	8	9950	10520
Free-cutting screw-stock steel (No. 414)	$E = Cwf^{0.77} d^{0.86}$	$= \frac{C}{33{,}000\, f^{0.23} d^{0.14}}$	0.954	0.784	6	8180	6720
High-speed steel	$E = Cwf^{0.73} d^{0.84}$	$= \frac{C}{33{,}000\, f^{0.27} d^{0.16}}$	1.437	1.525	1	9700	10400
			1.437	1.525	8	9700	10400
Cast iron (CF)	$E = Cwf^{0.41} d^{0.56}$	$= \frac{C}{33{,}000\, f^{0.59} d^{0.44}}$	0.685	0.740	1	595	643
Leaded screw-stock brass (BF)	$E = Cwf^{0.76} d^{0.96}$	$= \frac{C}{33{,}000\, f^{0.24} d^{0.04}}$	0.381	0.358	5	3830	3600
			0.381	0.358	6	3830	3600
Annealed leaded screw-stock brass (187-A)	$E = Cwf^{0.84} d^{0.87}$	$= \frac{C}{33{,}000\, f^{0.16} d^{0.13}}$	0.307	0.307	5	3700	3700
			0.3095	0.307	6	3730	3700
Annealed unleaded brass (50-A)	$E = Cwf^{0.77} d^{0.96}$	$= \frac{C}{33{,}000\, f^{0.23} d^{0.04}}$	0.573	0.491	5	6040	5170
Pure copper	$E = Cwf^{0.62} d^{0.94}$	$= \frac{C}{33{,}000\, f^{0.38} d^{0.06}}$	1.233	1.049	1	5980	5090
			0.795	0.695	8	3860	3375
Bakelite	Up: $E = Cwf^{0.21} d^{0.75}$	$= \frac{C}{33{,}000\, f^{0.79} d^{0.25}}$ (Up)	0.1432	....	1	74	..
	Down: $E = Cwf^{0.29} d^{0.83}$	$= \frac{C}{33{,}000\, f^{0.71} d^{0.17}}$ (Down)	....	0.1475	1	..	130
Wood fiber	Up: $E = Cwf^{0.49} d^{0.72}$	$= \frac{C}{33{,}000\, f^{0.51} d^{0.28}}$ (Up)	0.0685	....	1	121	..
	Down: $E = Cwf^{0.61} d^{0.91}$	$= \frac{C}{33{,}000\, f^{0.39} d^{0.09}}$ (Down)	....	0.0618	1	..	281

[a] Feed in inches, 0.010; depth of cut in inches, 0.125.

[b] Cutting fluid 1 is dry cutting.
Cutting fluid 4 is a soluble oil, 1 part oil to 10 parts water.
Cutting fluid 5 is a No. 2 lard oil.
Cutting fluid 6 is a light mineral oil.
Cutting fluid 8 is a light mineral oil containing 10 per cent No. 2 lard oil.
Cutting fluid 10 is a sulphurized light mineral oil.

inches. Such a formula holds not only for the energy in milling, but for the torque and thrust in drilling, and the cutting force in turning. Values of energy in foot-pounds per chip are shown for various combinations of depth and feed in Fig. 43 of this discussion, in which a log-log scale is used. At the left of Fig. 43, the milling energy in foot-pounds per chip for various values of feed per tooth for three depths of cut is shown, namely, 0.05, 0.10, 0.15 in., when cutting brass, consisting of 65.5 per cent copper, 34.1 per cent zinc, 0.25 per cent lead, and 0.10 per cent iron.[10] The solid lines represent the energy values when cutting up, as shown at A in Fig. 42, and the dashed lines the values when cutting down, as shown at B in Fig. 42. It is interesting to note that all six lines are parallel, the tangent of the angle of the slope being 0.77 which becomes the exponent x of f in the energy equation. It is seen that, if the depth for a given feed is doubled or tripled, the energy increases almost but not quite in the same proportion. The variable-feed curves show that cutting down requires less energy than cutting up, although this relation does not hold true for all metals, as shown in Table 2 of this discussion.

At the right in Fig. 43 are shown the milling-energy values in foot-pounds per chip for variable depths for each of two values of feed per tooth. Again, four straight lines are obtained, all of which are parallel. It is seen that, for a given depth of cut, if the feed is doubled from 0.005 to 0.010 in. per tooth, the energy increase is in the proportion of 2 to 3.5. This shows that thick chips require less energy in proportion than thin chips. The tangent of the angle of slope of these lines is 0.96 which becomes the exponent y of the depth d.

From the results of Fig. 43, an equation giving the relation between the net energy in foot-pounds per chip E, the width of the cutter w, the feed per tooth f, and the depth of cut d, all expressed in inches, may be written

[10] "The Elements of Milling," by O. W. Boston and C. E. Kraus, Trans. A.S.M.E., vol. 54, 1932, paper RP-54-4, pp. 71–92.

$$E = Cwf^{0.77}d^{0.96}$$

C representing a constant which, when cutting up, is 6040 and, when cutting down, is 5171. The energy values have been found to vary directly with the width of the cutter w. Similar equations for other materials when cutting up and down and using various types of cutting fluids are shown in Table 2 of this discussion for relatively light cuts for which the depth of cut is 0.125 in. and the feed per tooth 0.01 in. This equation shows that the energy per chip increases with an increase in feed and depth of cut only as the 0.77 power of the feed and the 0.96 power of the depth of cut. This proves the desirability, from a power standpoint, of taking heavy feeds.

The average thickness of chip (A.T.C.) is a function of the feed and depth of cut as explained. Fig. 44 of this discussion shows the net energy required at the tool point to remove various metals, such as free-cutting brass, cold-rolled steel (low-carbon), S.A.E. 1020 steel, and cast iron for various values of A.T.C., but for three different depths of cut, namely, 0.01, 0.1, and 1 in., respectively. The feed was such as to give the A.T.C. indicated.

The cubic inches of metal removed per horsepower-minute for milling free-cutting brass takes the form of one straight line when plotted over the A.T.C. on log-log paper, when the depth of cut is 0.01 in. Another straight line is obtained, somewhat below the first, for the same feeds when the depth of cut is 0.1, and a third straight line is obtained when the depth of cut is 1 in. This indicates that, for a given depth of cut, a straight power line is obtained, when the feed is varied to obtain different values of A.T.C. for a given depth of cut. For a constant value of A.T.C. such as 0.0006, the cubic inches per horsepower-minute for cutting brass are 2.5 when the depth is 0.01, but only about 2 when the depth of cut is 0.1 in. Again, 2.5 cu in. of the brass are removed per hp per min when the A.T.C. is 0.0006 in., 0.0013 in., and 0.0023 in., respectively, for the three depths of cut. The results for the other metals show also that A.T.C. is not a safe basis for computing energy values. The cold-rolled steel being milled with lard oil is the one exception, as the variation of depth does not destroy the continuity of the three curves.

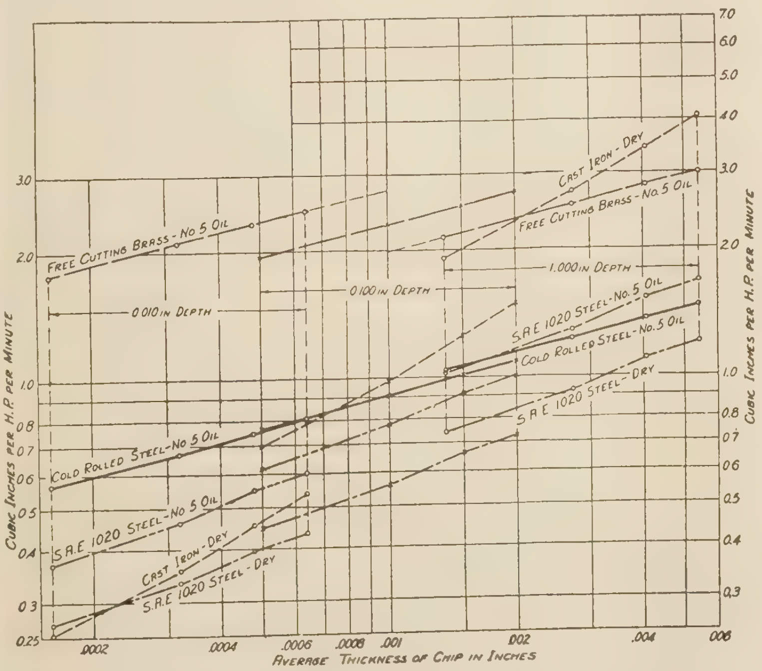


Fig. 44 Relation Between A.T.C. and Cubic Inches per Net Horsepower-Minute When Cutting Several Materials at Three Different Depths of Cut

(The curves are based on the horsepower formula given in Table 2 of this discussion, but plotted over the A.T.C., determined from the formula. All values are for milling up. Oil No. 5 is a No. 1 lard oil.)

The results of Fig. 44 indicate the desirability of taking small depths of cut when milling free-cutting brass. The reverse appears to be indicated when milling cast iron and steel. From this it is clear that the A.T.C. alone is not a proper basis for determining power requirements.

Net-energy and horsepower formulas with values of constants for milling different materials both up and down, with a variety of cutting fluids, are given in Table 2 of this discussion. These formulas and values are for the sizes of cut and cutter as indicated in the table heading.

The cutting condition, shown at A, Fig. 42 of this discussion, represents that most commonly used. The cutter rotates clockwise, while the work is fed to the right, both motions being indicated by arrows. In this manner, the cutting tooth cuts against the motion of the work. This is referred to as cutting up, or against the feed. At B the cutter rotates clockwise, while the work is fed to the left. This is called climb or down cutting, in which the cutting action is with the feed. The feed per tooth of the 4-tooth cutter is the same at A as at B. An analysis of the two chips formed

shows that the chip at A is thinner and longer than the chip at B. Illustrations at C and D represent, respectively, chips produced by a tooth of a 20-tooth cutter when feeding up and down. Again, it is seen that the chip at C is much longer and much thinner than the chip at D. Therefore, the shape of the chip produced by a cutter of a given number of teeth is influenced by the cutting being done up or down. Also, the difference in shape of the chips is greater, the greater the number of teeth in the cutter.

The author discusses the influence of films of various types on the built-up edge during the formation of a chip. The writer is not in agreement with his statements; the writer's views on the matter were presented in the form of a discussion to a previous paper[11] by Ernst and Merchant.

O. R. Schurig.[12] Thoroughgoing studies of the kind presented in this paper are basic in establishing the fundamentals of everyday shop operations which are in common use and which are so little understood, except for such studies. A more complete understanding of the processes involved will permit operators of machines, or those who control shop practices, to speed up production, or to obtain an improved surface quality, or to increase the life of machine tools, as the case may be.

The author mentions certain advantages resulting from upmilling, including smoother surfaces, because the built-up edge, if any, is not close to the cutting edge when the cut is started, and the initial cut is the one which remains on the final surface. It is also understood that a built-up edge is later formed, as the cutting edge advances and cuts more deeply into the metal.

The writer, therefore, wishes to inquire whether the built-up edge formed on the tool face at the end of the cut remains there as the tool leaves the metal and, if so, why the presence of the built-up edge does not affect the smoothness of cutting at the beginning of the next cut after 1 revolution?

A. E. Richard de Jonge.[13] The author has chosen for his investigation a highly interesting subject which required clearing up beyond what was known heretofore. His analysis of the milling process must, therefore, be received with gratification by the profession. It should be kept in mind, however, that the subject is by no means a new one. The first clear statements about the milling process were made by Prof. F. Reuleaux, in 1900, who published them in the second part of his "Lehrbuch der Kinematik."[14] He was the first to draw attention to the fact that the paths of the teeth of a milling cutter are trochoids (he calls these curves cycloids). He also was the first to discuss the two modes of feeding the material to the revolving cutter, namely, in the direction of rotation of the cutter, and against it. Today, these two processes are called down-milling and upmilling respectively. He had tests made here in the United States, by Messrs. Pratt and Whitney, on whether the first or the second mode of feeding was preferable and produced smoother and better work. The results were indecisive as the marks due to tooth errors were much more pronounced than were the tooth marks. Other tests made at that time for him by Messrs. Ludwig Loewe and Co., of Germany, favored down-milling. Reuleaux also gave a simple formula which showed that in order to obtain tooth marks of small pitch, the circumferential velocity of the cutter must be high, and the speed of the workpiece low. Conversely, if the pitch of the tooth marks is to be great so that the work appears smooth, the speed of the cutter must be low and the speed at which the workpiece is fed to the cutter high.

The author's investigation can be divided into two sections, a practical or experimental one, and a mathematical one. The author has called the latter "Kinematics of Milling." This section forms the greater part of the investigation although the experimental section appears to be the more important one as it gives, for the first time, proof of what actually happens when a chip is removed. Indeed, Figs. 3, 23, and particularly Fig. 25 show what occurs in the formation of a chip, and how little the basic structure of the material is disturbed thereby. The various photomicrographs given by the author amplify and round out the picture of what takes place in milling various metals and what the quality of the surface is that may be obtained. Of value are also the diagrams Figs. 10, 11, 12, 13, 27, 28, 32, 33, 36, 37, 40, and 41, which, as the author has stated privately to the writer, were obtained from actual tests, although the test values are marked only in Figs. 12 and 13. Fig. 6 is of interest in so far as it confirms the statement by Reuleaux that the revolution marks are far more pronounced than are the tooth marks.

Regarding the kinematic analysis, the author, apparently, has gone the long and thorny path of analytical geometry combined with differential calculus, instead of making use of the modern methods of graphical kinematics. Thus, in Appendix 1 for example, there is no need for differentiation to find tan ϕ, because this function can be read off directly from the geometrical figure. The radius of curvature of the tooth path (Equation 6) was also obtained by differential quotients, as stated by the author to the writer in the private communication mentioned. This also is unnecessary as, by the modern kinematic methods, the formula can easily be obtained by similar triangles, making use of the velocities of the tooth generating the trochoid and of the instantaneous center. Most of the other formulas can be read off directly from the respective figures. Yet, the results obtained by the author are correct; only, with respect to Equation 5, it should be mentioned that an approximation has been made and a certain term neglected which, however, is permissible due to the small magnitude of the quantities involved.

As the author has chosen for the title of his paper "An Analysis of the Milling Process," one would have expected that he would have discussed the entire subject. In fact, he has discussed only one half of it, having confined himself to up-milling. Down-milling is just as important, however, and according to Reuleaux and the Loewe experiments cited is even preferable. Yet, this does not actually seem to be the case, and it would be very interesting to see both the mathematical analysis and its experimental verification for the case of down-milling.

In down-milling the rolling centrode (or polode) rolls upon a straight line near the surface of the workpiece instead of, as in up-milling, on a straight line beyond the center of the milling cutter, reckoned from the surface of the workpiece. Consequently, the curvature of the tooth path is far greater, that is, the radii of curvature are very much smaller, than in up-milling. This has an effect on the clearance angle of the teeth, the tooth having to be much sharper. When the cutter has straight instead of helical teeth, it hits the surface of the workpiece at a spot of solid material, hence with an impact, and thus sets up vibrations and chattering; while, in up-milling, the cutter starts its cut at the thinnest part of the chip and not at the thickest part, thus gradually increasing the energy given off to form the chip. If the cutter has helical teeth, there is an impact only at the instant when the front part of the tooth starts cutting, but this impact is greatly reduced if the previous tooth or teeth are

[11] "Chip Formation, Friction, and High Quality Machined Surfaces," by Hans Ernst and M. E. Merchant, Trans. American Society for Metals, 1940, preprint No. 53.

[12] General Engineering Laboratory, General Electric Company, Schenectady, N. Y.

[13] Adjunct Professor, Polytechnic Institute of Brooklyn, Brooklyn, N. Y. Mem. A.S.M.E.

[14] "Die Praktischen Beziehungen der Kinematik zur Geometrie und Mechanik," Lehrbuch der Kinematik, vol. 2, Braunschweig, 1900, pp. 685–689.

still cutting. The distortion of the material forming the chip may, however, be greater than in up-milling, and it would be highly interesting to see the experimental evidence, which the author perhaps is able to supply. As the cutting edges of the teeth have to be much thinner, the life of the cutter seems to be greatly reduced, but here too the author can probably give actual figures which should be of interest.

One would also like to see the actual evidence for the smoothness, or rather roughness, of the surface due to the tooth marks so as to be able to compare the surface finish with that in the case of up-milling. Of interest would also be the formation of the "built-up edge" in down-milling, which should have an important bearing on the actual surface finish. Of importance would, further, be diagrams of the power consumption per cubic inch of metal removed, for various metals, such as the author has given for up-milling. Perhaps, the author might clarify all these points and, thereby, contribute a further valuable piece of information to the very useful facts he has already given in the present paper. It should be stated once more that the profession owes him a debt of gratitude for the very careful investigation he has made of the milling process.

Author's Closure

The author appreciates the comments and the criticisms of Professor de Jonge who has read the paper and checked the derivation of the various equations. He notes the absence of any reference to the down-milling method, although from the title of the paper one would expect a complete analysis of the two methods of milling ("up-milling" and "down-milling"). The paper would have been more appropriately titled: "An Analysis of the Milling Process: Part I—Up-Milling," since the author has actually made a parallel analysis of the two methods and has on hand the information mentioned by Professor de Jonge. It is the author's hope that this material, together with an analysis of the mechanism required to permit down-milling operations, may be presented in the near future in a second paper.

Space limitation prevented the publication of this material in its entirety in the initial paper, and the same reason prevents the author from elaborating on this subject at this time.

The author finds that the analytical method used, supplemented with actual tests, is conducive to a better understanding of the characteristic differences between the two methods of milling, and also makes it possible to visualize, from a practical viewpoint, the advantages and disadvantages inherent in both methods.

Professor Boston takes exception to the author's suggestion that the average thickness of the undeformed cross section of the chip may be used for establishing the relationship between the conditions of the cut and the corresponding power required, on the basis that the results he has obtained with the A.T.C. formula derived by Parsons[9] gave rise to some apparent inconsistencies.

Quoting Professor Boston:[10] "It is obviously inaccurate to use the A.T.C. as the variable in a general milling formula . . . . . This is unfortunate, because the average thickness of the chip seems to be a logical basis for determining cutting qualities."

In his investigation, Professor Boston has used the so-called pendulum-type milling machine, also known as "chip tester." This machine was developed and used by Professor Airey and C. J. Oxford, and was described[15] by them some years ago.

In the chip tester, one fundamental element of milling is eliminated, i.e., the feed of the work. Consequently, in operation, a circular tooth path is obtained instead of the true path which, as shown by the author, is trochoidal. It follows then that with the chip tester there is no geometric difference between up- and down-milling except for the point where the tooth contact with the work begins. For given cutting conditions, the average thickness of the chip is identical in the two cases, and the energy involved in forming the chip should also be the same.

[15] "On the Art of Milling," by John Airey and C. J. Oxford, Trans. A.S.M.E., vol. 43, 1921, p. 549.

The values of hp per cu in. per min, given by Professor Boston in Table 2 of his discussion, for the two methods of milling, indicate that in some cases there is a difference in favor of down-milling; in other cases, there is no difference; and, in yet other cases, the values for down-milling are higher than those for up-milling. Since these differences are small, one is inclined to believe that, because of the limitations of his experimental apparatus and, furthermore, to the difficulty in setting the work accurately for the required feed per tooth, the variations found by Professor Boston are due to experimental errors rather than to actual differences resulting from the two methods of formation of the chip.

It is to be noted also that, with Professor Boston's chip tester, the rake and clearance angles are constant throughout the tooth path, while in actual milling they vary continuously from the beginning to the end of the chip, as shown by the author.

For the conditions investigated by Professor Boston, the instantaneous radial thickness of the chip can be exactly calculated by means of Equation [17] of the paper. This was derived for the case in which the tooth path is circular. For the purpose of this discussion, the simplified form, Equation [19], of this equation is used

$$t = F_t \sin \alpha \qquad [28]$$

where

t = instantaneous thickness of undeformed section of chip, in.

F_t = feed per tooth, in.

α = angle through which radius vector has rotated from beginning to end of tooth path

The angle α can be evaluated in terms of the elements of the cut from the equation for Y in the system of Equations [2] of the paper. When this is done Equation [28] of this closure, assumes the form given by Parsons for the maximum radial thickness of the chip

$$t = 2f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)}$$

where

f = feed per tooth, in.

d = depth of cut, in.

D = diameter of cutter, in.

From this formula Parsons derived the following expression for the so-called "average chip thickness" by dividing by 2 his expression of the maximum chip thickness

$$\text{A.T.C.} = f\sqrt{\frac{d}{D}\left(1 - \frac{d}{D}\right)}$$

This is identical with the expression

$$\frac{F_t \sin \alpha}{2}$$

But this obviously corresponds not to the average, but only to one half of the maximum thickness of the chip.

The true average value, however, can be determined from Equation [28] as follows: In general if $f(x)$ is a real function of x, the average value f_{avg} of $f(x)$ with respect to x over a given interval $a \leqq x \leqq b$ is defined to be

$$f_{avg} = \frac{1}{b - a}\int_a^b f(x)dx$$

In our case

$$t = f(\alpha) = F_t \sin \alpha$$

and the limits are $0 = \alpha \leqq \alpha_1$; therefore

$$t_{\text{avg}} = \frac{1}{\alpha_1} \int_0^{\alpha_1} F_t \sin \alpha \cdot d\alpha \ldots\ldots\ldots\ldots [29]$$

and carrying out the integration

$$t_{\text{avg}} = \frac{F_t}{\alpha_1} [1 - \cos \alpha_1] \ldots\ldots\ldots\ldots [30]$$

or

$$t_{\text{avg}} = F_t \frac{\sin^2(\alpha_1/2)}{(\alpha_1/2)} \ldots\ldots\ldots\ldots [31]$$

The difference between the values of t_{avg} obtained with the Parsons formula and the more nearly correct Equation [31] of this closure is graphically illustrated in Fig. 45, herewith.

Therefore, it is evident that Professor Boston made a serious mistake in trusting the correctness of the Parsons formula, and in concluding therefrom that the average chip thickness is not a good basis for judging cutting qualities.

It is also interesting to note that, by substituting in Equation [31] for $\cos \alpha$, the expression $(R - d)/R$, this equation becomes identical with Equation [15] of the paper, derived by the author from a direct consideration of the volume of the chip, when L is made equal to the length of the arc of a circle corresponding to the simplified tooth path.

Obviously, for an exact determination of A.T.C., the true trochoidal tooth path must be used as given in Equation [13] of the paper.

Furthermore, it is important to consider that, in milling practice, a variety of combinations of feeds, depths of cut, and speeds are found. Consequently, an approximated formula cannot be used indiscriminately. From Fig. 45 of this closure, it is evident that the Parsons formula will be found satisfactory only for values of α up to 30 deg.

From this it can be concluded that, in using Parsons' formula in the range of depth from 0.01 to 1 in., inconsistencies such as mentioned by Professor Boston will result, and a revision of his conclusions is therefore in order. Professor Boston is still of the opinion that a milling-cutter tooth rubs over the surface of the work at the point of contact until the cutting edge digs into the work. His reference to this action indicates that he has not read the part of the paper in which the author has dwelt at length on the fallacy of this opinion. The author has produced experimental evidence which proves that the cutting edge of a milling-cutter tooth begins to form a chip immediately upon contacting the work.

Professor Boston mentions the works of Salomon[7] and Sawin[8] in connection with the discussion of this paper.

Professor Salomon, without even analyzing the path of a milling-cutter tooth assumes ipso facto that the path is an arc of a circle, and then proceeds by a complicated mathematical procedure to show that the thickness of the undeformed section of the chip, corresponding to a point of the path located at $\alpha/2$ is the true criterion for determining the power required in milling; the angle α being that subtended by the circular arc of the complete tooth path.

His formula for this thickness of the chip is

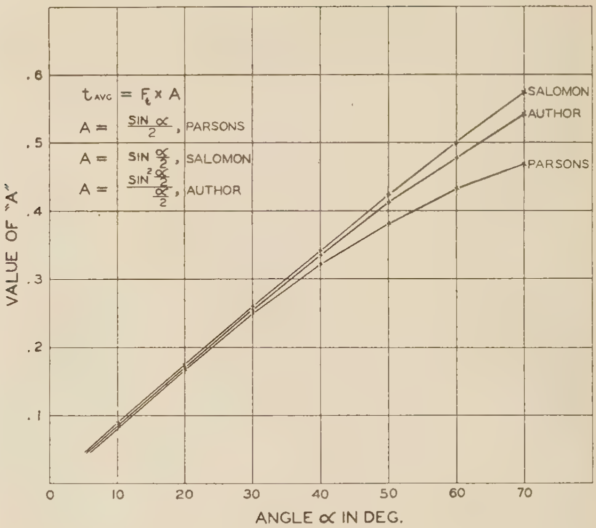


Fig. 45 Comparison of Various Formulas for Determining Average Thickness of Undeformed Section of Chip

$$t = F_t \sin \frac{\alpha}{2}$$

Fig. 45 of this closure shows the variation between the Salomon formula and that derived by the author for the average value of the thickness of the chip for the case (not true in practice) of a circular tooth path.

Mr. Sawin accepts the approximation often made by various investigators that the path described by a tooth can be simulated by an arc of a circle, and then proceeds to produce a number of formulas for calculating the dimensions of a milling cutter.

The author wishes to state that a better understanding of the physical facts involved in any machining process may be obtained by considering (*a*) the principle on which the process is based, to determine the interdependence of the various elements by analytical methods, regardless of the magnitude of the single elements involved; and (*b*) to make the necessary approximation in certain cases if found admissible within the limits of practical application. Evidently this procedure was not followed by Professor Boston.

The author appreciates Mr. Schurig's interpretation of the studies which formed the object of his paper. In the majority of observations made by the author, covering a number of years, he has found that the built-up edge (formed during the passage of a tooth through the work) remains strongly attached to the chip (it is actually a part of it, Fig. 3 of the paper), and usually passes off with it. The entire built-up edge, therefore, will not be found on the edge of the tooth at the completion of its engagement with the work. There may be small particles of it adhering to the face of the tooth, but these particles will usually be displaced by the oncoming new chip, and thereby a clean edge will result.

In those cases, particularly when dry-cutting, in which a strong bond between the face of the tooth and the material of the chip develops, the completely formed chip with the built-up edge may be carried around by the tooth and fall on the finished surface where it might be caught by the next tooth approaching the work. This will result in blemishes on the finished surface, as shown in Fig. 19 of the paper (*A*, *B*, and *C*).

An effective cutting fluid will invariably improve the conditions mentioned.

Calibration of Displacement Meters on Volatile-Liquid-Petroleum Fractions

By E. W. JACOBSON,[1] PITTSBURGH, PA.

The author describes a prover design for use in calibrating displacement meters on volatile-liquid petroleum fractions such as gasoline which will eliminate errors caused by evaporation, change in temperature, and entrainment of gases in the test liquid. Details are also given of a simplified commercial type of prover. This type of prover has been successfully used on an extensive meter-testing program at the Gulf Research Laboratory as a part of the 25-year program of the A.S.M.E. Fluid Meters Committee.

EQUIPMENT for calibration of displacement meters on volatile-liquid-petroleum fractions, such as gasoline, should be such that calibration errors, caused by evaporation of the liquid, change in temperature, and entrainment of gases, are eliminated. Meters for measuring gasoline should be calibrated on gasoline. Accurate calibration of a meter on a petroleum fraction other than gasoline does not insure that measurement on gasoline will be satisfactory; in fact, the contrary can be expected, as the viscosity of the liquid measured has a pronounced effect upon the calibration. The same meter will usually show less variation on the higher-viscosity fractions but will require alteration of adjustment to bring the average error nearer zero. If a meter is to be used on liquids of different viscosities, a separate calibration for each liquid should be made, since a single factor cannot be applied to all the liquids, to different makes of meters, or even to different meters of the same make.

The need for elimination of the errors due to evaporation of the test liquid needs little explanation. The possibility of sizable error in calibration, when evaporation is allowed to take place, is easily demonstrated by pouring gasoline from one measure to another, when a loss of at least 3 to 5 cu in. in 5 gal will be noted. Since 1.15 cu in. in 5 gal is an error of 0.1 per cent, this simple operation involves an error in measurement from evaporation alone of 0.2 to 0.4 per cent. This is a material error when considered from the standpoint of the claims for accurate measurement often made for displacement meters.

Errors introduced by volumetric change, due to change in temperature, can introduce considerable errors in calibration. The coefficient of cubical expansion for liquid hydrocarbons lies between 0.0004 for crude oils and 0.0007 for gasoline at ordinary temperatures. A change in volume of 0.1 per cent is therefore caused by 1.4 F change in temperature of gasoline and by 2.5 F change in crude oil.

Errors from gas entrainment in the test liquid will depend upon the conditions under which the entrainment takes place. Turbulent liquid flow in the presence of a gas will cause the liquid to absorb the gas rapidly and, after the saturation point has been reached, some gas will be entrained in the form of bubbles. If conditions of temperature and pressure change slightly, gas may be thrown out of solution to make more bubbles. The bubbles tend to rise to the surface of the test liquid when it comes to rest and in some petroleum fractions cause foaming, which retards accurate determination of the liquid level. Considerable time is necessary for all the bubbles to rise and leave the surface of the liquid.

Meter-Test Work at Gulf Laboratory

At the beginning of the meter test work at the Gulf Laboratory early in 1934, a survey of the field disclosed no known method of testing meters on gasoline which would remove the errors of measurement due to evaporation of the liquid, temperature change, gas entrainment, and foaming. Therefore, it was necessary to develop an accurate meter prover which would eliminate these errors as far as possible. The basic principle used in the prover was to confine the test liquid over another liquid immiscible with the test liquid, in a gastight container. By suitable design using this principle, it was possible to maintain the test liquid out of contact with gases and thus prevent evaporation, absorption of gases by the test liquid, and foaming. A further result obtained by this method was a decided reduction in the rate of deterioration of the test liquid. Oxidation was prevented and deterioration due to mechanical agitation was found by test to be very slight.

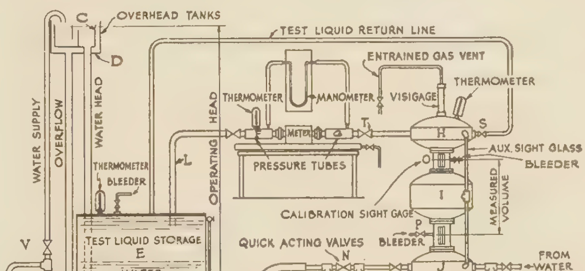


Fig. 1 Schematic Diagram of Prover Used for Testing Meters
(Gulf Research & Development Company; U. S. Patent No. 2,050,800.)

The prover used for testing meters for error of measurement is shown in Fig. 1. Water was the immiscible liquid used to confine the gasoline and the burning oils on which the meters were tested. The prover consists essentially of a circulating pump, overhead tank, storage tank, and measuring tank with the necessary control valves and piping. The method of operation of the prover in testing a meter for error of measurement is as follows: Referring to Fig. 1, tank *A* contains a supply of water which is pumped through water-supply line to overhead tank *C*. Tank *C* discharges into tank *E* which contains test liquid in the portion above water. The head exerted by the water at a height of tank *C* forces gasoline through pipe *L*, pressure tube *F*, meter, pressure tube *G* into the upper bell *H* of the measuring tank. This test liquid displaces water in the measuring tank *I*. The displaced water passes through the bell *J*, flow-regulating valve

[1] Design Engineer, Gulf Research & Development Company. Mem. A.S.M.E.

Contributed by the Petroleum Division and presented at the Semi-Annual Meeting, Kansas City, Mo., June 16–19, 1941, of The American Society of Mechanical Engineers.

M, and quick-opening valve N, then discharges into water-storage tank A.

Flow of the test liquid through the meter on test takes place only when quick-opening valve N is opened. Valve N is opened only long enough to allow a definite amount of test liquid to pass through the meter into the measuring tank. When no test liquid is being displaced, the water being pumped into overhead tank C, overflows tank C into tank D from which it is carried back to storage tank A. At the start of the test, the dividing level between water and test liquid is set at zero on sight gage O, by proper manipulation of valves R, S, and N. Valve T in the meter line is closed during this manipulation. Valve T is then opened after valves R, S, and N are closed. Valve M is adjusted to give the proper rate of flow and the meter-register reading is noted. The quick-opening valve N is opened and the water head at tank C forces the test liquid through the meter into the measuring tank, and the displaced water from the measuring tank discharges into storage tank A. During this procedure, the dividing level in the measuring tank is lowered to some point within lower sight gage P. If the metered volume of the test liquid is exactly the same as the volume of the measuring tank between the zero on gage O and the zero on gage P, the dividing level will be at zero on gage P. If the two volumes are not the same, the reading on gage P at the dividing level will be the error of measurement of the meter. After a test run, the test liquid which has been passed through the measuring tank during the run is returned to the storage tank through the test-liquid return line by closing valve T and pump throttle valve V, and opening valves R and S. This forces the test liquid by means of water under pressure from pump K, into tank E, which in turn displaces water up through tank C into tank D and back into storage tank A.

Air entrainment is prevented by inserting the end of pipe L into tank E so that any air which may get into tank E will collect in the top of the tank where it can be removed through the vent pipe. The curved heads of the measuring-tank bells carry any entrained air up through the liquid in the tank into the visigage on top where it is readily observable to the operator. Some air will be present in the visigage during the first runs after the insertion of a meter for test, until the meter has been worked free of air. This action ordinarily takes two or three test runs, after which, succeeding runs will show the system to be free of air. Air in the visigage is bled off through the vent line. Pressure drop through the meter is determined by the use of manometers connected across standard pressure tubes F and G, which contain piezometer rings and pitot tubes.

A constant operating head on the system is maintained by the overhead spill tank C. The total available head for operating the system is determined by the height of the upper edge of tank C above the level in storage tank A. The bleeders on the sides of sight gages P and O are for the purpose of removing any bubbles, dirt, or contamination which may collect in the dividing level between the water and the test liquid in the measuring tank, which may cause difficulty in gaging the level. The auxiliary sight gage shown on one side of the measuring tank aids in determining the location of the measuring dividing level.

Commercial-Type Prover of Simplified Design

The apparatus just described is of the type to be used in an extensive meter-test program, such as carried on at the Gulf Laboratory, wherein not only the accuracy of measurement under test was desired, but also the variation in accuracy and pressure drop throughout a life test. For a commercial application of this type of prover, a simplified setup, such as shown in Fig. 2, is sufficient, when only the measuring characteristics of the meter are required, as in routine pipe-line measurement or distributing-

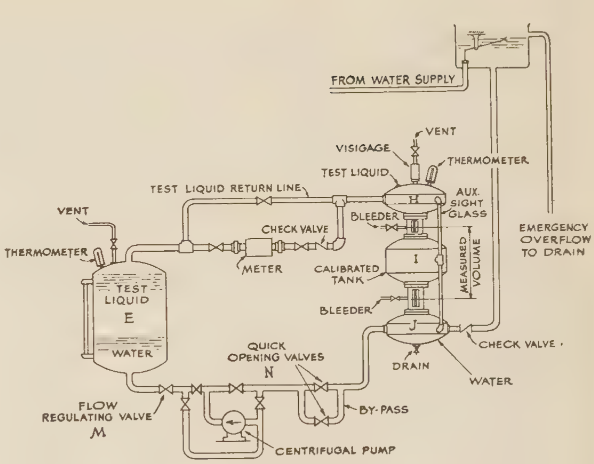


Fig. 2 Commercial Type of Prover of Simplified Form

truck service. The essential elements for a commercial prover would require only two tanks, one the test-liquid storage tank and the second a measuring tank, such as combined tanks H, I, and J, Fig. 2. The discharge from tank J would then be piped through the regulating valves and circulating pump directly into the water side of tank E. Any desired pressure could be maintained on the entire system by connecting an outside high-pressure water supply into tank J through a reducing valve or overhead float tank as shown.

The measuring compartment of a good prover should have a capacity at least equal to the volume corresponding to the full flow of the meter in 1 min; for instance, a prover for testing meters with maximum flow rate of 300 gpm should have a measuring-compartment capacity of at least 300 gal. Careful calibration of the measuring compartment is of the utmost importance.

It should be pointed out that the most commonly used primary standard of measurement, the ordinary 5-gal sealer's measure, is difficult to set closer than 1 cu in. in 5 gal, and even the most accurate measure cannot be depended upon to be more accurate than the nearest 0.5 cu in. This was confirmed by considerable test work at the Gulf Research Laboratory, using both the weighing method with distilled water at constant temperature and a precision balance, and the beaker method. Great care should be taken in handling a measuring can after it has once been accurately calibrated. The best means of avoiding possible damage to the measuring can is to keep it in a case lined with soft material so that it cannot be dented or otherwise damaged when not in use. Where a great deal of calibration work is being done, it would be an excellent practice to have at least two measuring cans, one to be kept in storage. A frequent check should be made against the one in use to insure that it continues to retain its original accuracy.

Several trial fillings of the prover measuring tank, using accurately calibrated measuring cans, should be made to give an average reading for the correct setting. Besides the error in the measuring cans, additional error may be involved in such checking operations by failure to observe due care in maintaining wetted surfaces in the 5-gal measures and prover. There will be some slight further variation due to the different wetting qualities of the test liquid and the liquid to be used in the prover. Even with the utmost care in calibration of a prover measuring tank, differences between provers of at least 0.1 per cent may be expected.

The type of prover described has been used with good success on burning oils as well as gasoline. It should be satisfac-

tory for use in calibrating meters on propane and butane where the test liquid must be maintained under pressure. The curves, in Figs. 3 and 4, are representative of the data obtained on the Gulf Laboratory prover. Repeat tests with this prover at a particular rate of flow gave good repetition of data, demonstrating the suitability of the prover for the service.

Fundamental Characteristics of Displacement Meters

Use of a prover, such as just described, for calibrating displacement meters on volatile-liquid-petroleum fractions may

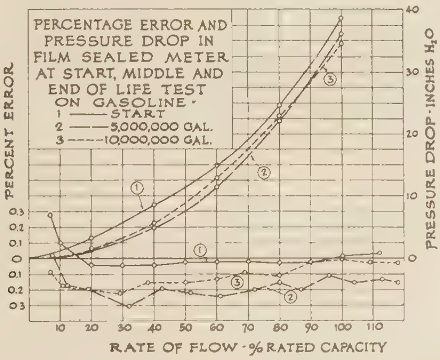


Fig. 3 Percentage Error and Pressure Drop in Film-Sealed Meter at Start, Middle, and End of Life Test on Gasoline

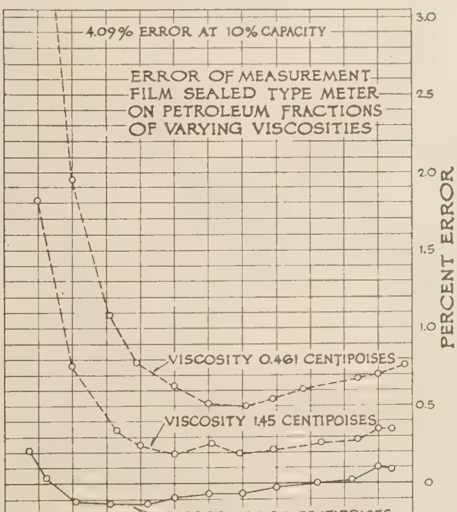


Fig. 4 Error of Measurement, Film-Sealed-Type Meter, on Petroleum Fractions of Varying Viscosities

reveal some surprising measuring characteristics. Before discussion of these characteristics, some fundamentals of displacement meters should be pointed out. A displacement meter is a fluid motor which must generate enough power to overcome the friction of its moving parts as well as drive the registering mechanism. In order to derive power from the fluid stream passing through the meter, there must be a pressure drop through the meter. It is, therefore, evident that, with a difference of pressure and with working clearances in the metering element, there must be some slip, namely, excess of fluid passed through the meter over and above the amount actually displaced in the

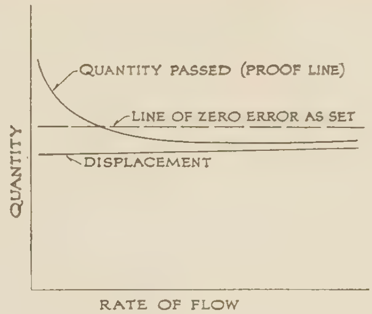


Fig. 5 Displacement Data for Typical Metering Element

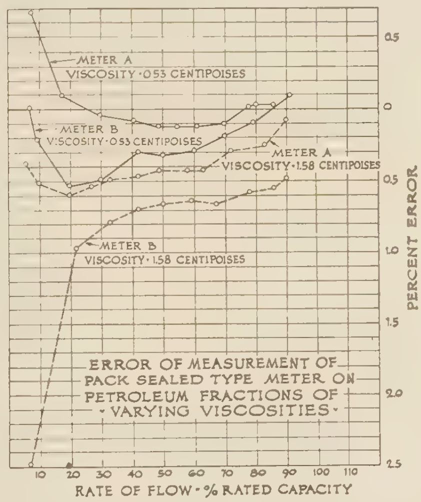


Fig. 6 Error of Measurement, Pack-Sealed-Type Meter, on Petroleum Fractions of Varying Viscosities

metering element. A plot of the displacement data for a typical metering element is shown in Fig. 5, wherein the quantity is plotted against rate of flow. The amount of slip depends upon the viscosity, load on the metering element, and various factors inherent in the design of the metering element, the same as in a fluid motor. Since the slip depends upon these factors, variation of any one or all of them will alter the location of the proof line by an amount depending upon the particular characteristics of the meter. The slip determines the position of the proof line above the displacement line. The position of the line of zero error, as indicated by the register, is determined by alteration of the displacement or by changing the gear ratio of the register drive so that the line of zero error will cross the proof line to give the minimum average difference between the zero-error value and the proof-line values. The line of zero error, since it is set by mechanical means, remains fixed with reference to the displacement line. The proof line changes shape and position, depending upon the viscosity of the liquid, wear in the mechanism, and change in the register and friction load.

Meter Performance

The most important characteristic of meter performance is the effect of viscosity on the shape and position of the proof line. As pointed out in a previous paper[2] by the author, film-sealed-

[2] "Some Fundamental Considerations in the Design and Application of Displacement Meters," by E. W. Jacobson, A.S.M.E. Petroleum Fluid Metering Conference, Norman, Okla., April 11–12, 1940.

type meters are in general affected much more by change in viscosity than pack-sealed-type meters. Fig. 4 shows error data for a film-sealed-type meter which was greatly affected by viscosity of the liquid. This meter gave satisfactory measurement as calibrated on a burning oil of 2 centipoises viscosity. However, its calibration on lighter petroleum fractions showed very unsatisfactory measuring characteristics. Fig. 6 shows data for two pack-sealed meters of identical size and construction. Meter *A* showed wide variation between burning oil (1.58 centipoises) and gasoline (0.53 centipoises). Meter *B* shows the same difference, except that it had large error in deficiency at flow rates below 20 per cent of rated capacity. This difference at the low rate of flow was likely chargeable to faulty valve setting. These data show extreme cases of faulty measurement and are not to be construed as being representative, however, they do illustrate some of the reasons for failure of meters to check one another. Proper testing of each meter would disclose unsatisfactory characteristics and would permit their correction.

In conclusion, a prover design, such as described in this paper, is available for use in calibrating displacement meters on volatile, low-viscosity petroleum fractions which will eliminate calibration errors caused by evaporation of the liquid, change in temperature, and entrainment of gases in the test liquid. This type prover has been successfully used on an extensive meter-testing program and has demonstrated the need for testing each meter on the liquid on which it is to be operated.

Acknowledgment

The writer is indebted to Mr. R. J. S. Pigott, chairman of the Fluid Meters Committee, for suggesting the material for this paper. The apparatus described was used in the fluid-meter tests made at the Gulf Research Laboratory, Pittsburgh, Pa., as a part of the 25-year program of the A.S.M.E. Fluid Meters Committee.

Treatment of Make-Up Water for the Waterside Topping Installation

By C. B. Arnold,[1] R. T. Hanlon,[2] and J. Mindheim,[3] New York, N. Y.

This paper deals with boiler-feedwater treatment for the 1400-psi topping installation at the Waterside Station of the Consolidated Edison Company of New York, Inc. On the basis of thermal economy, degasification, treating costs, and installation convenience, a cold-water carbonaceous-zeolite treating system was deemed preferable. The application of the Zeo-Karb system selected, its operation, and results obtained constitute the principal features of the paper.

The economy of large-scale district heating with exhaust steam from high-pressure turbines has long been appreciated, but lack of experience in the use of raw make-up water for high-pressure units discouraged earlier consideration of such installations without evaporators. However, when the first 1400-psi topping units were installed at Waterside Station, it was felt that the progress in water softening and boiler-feedwater treatment had advanced sufficiently to justify consideration of this scheme with greater assurance as to its ultimate success. Evaporators were also considered but the penalty for the pressure degradation incident to their use, combined with the need for treatment of the water for the evaporators, rendered them unattractive.

Of the two waters supplied to New York City, Catskill water would have been more desirable, due to its lower hardness and silica content, but unfortunately, only the Croton supply could be made available to the station. Although this water is comparatively pure, it could not be used unsoftened in the high-pressure plant, since its bicarbonate hardness content would cause the ultimate formation of scale in stage heaters, piping and economizers, and the problems within the boiler would be aggravated.

Various treating schemes were considered, and the experience and results from many plants were studied in connection with the problem, but the high initial purity of the water limited treating considerations to the few systems suited to the hardness characteristics of Croton water. On the basis of thermal economy, degasification, treating costs and installation convenience, a cold-water carbonaceous-zeolite treating system was deemed preferable. There being no return condensate from the steam used for district heating, the water-softening capacity had to equal or exceed the quantity of steam so used.

The treating plant permits increasing the output of the high-pressure units by the disposal of a large quantity of 200-lb exhaust steam. During light-load periods, the load which would otherwise be carried on less efficient condensing units is transferred to the more efficient high-pressure units. With a heat consumption below 5000 Btu per kwhr on these units, the advantage of increasing their capacity factor and output is obvious.

The present electrical and steam-generating equipment installed at Waterside No. 2 Station consists of two high-pressure topping turbines and four 1400-psi boilers, capable of generating a total of 2,000,000 lb of steam per hr which have been arranged to top the existing 200-psi steam system common to both Waterside Stations No. 1 and No. 2. Two more high-pressure turbines and four more high-pressure boilers, capable of handling 2,460,000 lb of steam per hr are now being installed as additional topping capacity.

The 200-psi steam system for the two interconnected stations is arranged for the dual purpose of conducting steam to the low-pressure condensing turbines in both stations and supplying steam to the New York Steam Corporation's heating mains through a total of twelve desuperheaters having a capacity of about 1,250,000 lb of steam per hr. Although all low-pressure boilers in Waterside Station No. 2 have been removed to make way for the high-pressure boilers, the 200-lb steam system is still capable of being fed by steam generated in the old low-pressure boilers in Waterside Station No. 1, which have an available capacity of 1,836,000 lb of steam per hr.

For a time after the first two topping units were installed, it was deemed advisable that only condensate from the low-pressure turbines and auxiliaries be used as feedwater for the high-pressure boilers. This required that all make-up water for the two stations be taken into the low-pressure boilers where it was evaporated and fed to the 200-lb steam system. Although this arrangement provided the high-pressure boilers with ideal feedwater, it was most uneconomical since every pound of steam delivered to the heating mains had to be made up by the generation and condensation of a pound of steam in the low-pressure system. In other words, this was the same as though the low-pressure boilers were alone feeding the heating mains. The problem, therefore, was to provide a water-treating system by which satisfactory make-up water could be fed into the high-pressure boilers without the necessity of operating the low-pressure boilers, except during emergencies or at times of high load demands.

The requisites of the system were considered to be briefly as follows:

1 Practically complete removal of calcium and magnesium scale-forming salts from the make-up water.

2 A minimum of total solids in the treated water so that the blowdown from the boilers would be as little as possible.

3 A minimum of physical size of the treating system because of the very limited available space in the station.

4 A capacity of 1,200,000 lb per hr of which 600,000 lb per hr should be the initial installation. This ultimate capacity would provide the make-up for a continuous delivery of 1,000,000 lb of heating steam per hr and 200,000 lb of blowdown per hr.

In considering the type of equipment to be installed, the Permutit Zeo-Karb system appeared to best meet these requisites. It also compared favorably with other equipment, considered from the standpoint of installation and evaluated operating costs. In consequence, the Permutit system was purchased and installed. It was placed in operation during September, 1939, and

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Contributed by the Joint Research Committee on Boiler Feedwater Studies and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

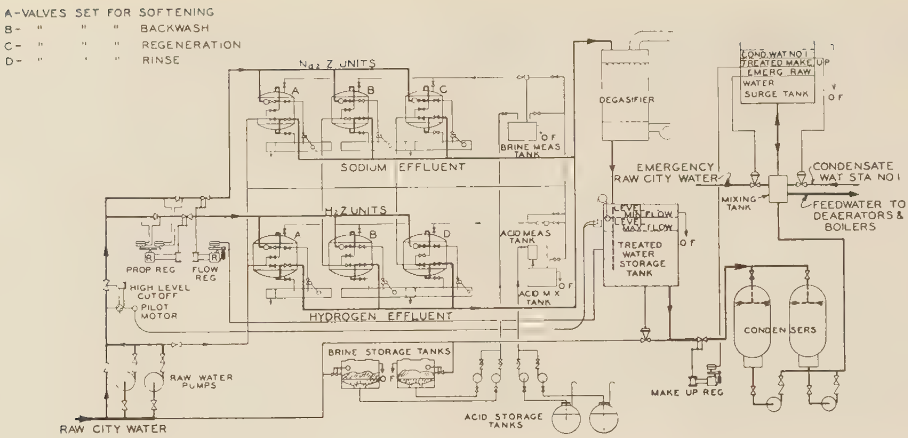

Fig. 1 Water Circuits, Zeo-Karb System; Waterside Station No. 2

is the first installation of the Zeo-Karb system for pretreating the make-up for boilers operating at pressures of this range. To date, it is the largest combination of the hydrogen and sodium Zeo-Karb arrangement ever built.

As the word implies, "Zeo-Karb" is a synthetic, carbonaceous zeolite material. Differing from the natural green sand commonly used as an exchange material, Zeo-Karb is manufactured from coal. The finished product consists of small porous particles of carbonaceous material slightly coarser than granulated sugar. Examination reveals that the Zeo-Karb is practically free of all ash and hydrocarbons, the removal of which by treatment with sulphuric acid accounts for its porous structure. The advantage of Zeo-Karb over green-sand zeolite, aside from its being nonsiliceous, is that it is not attacked or greatly affected by weak acids. This feature permits it to be used as so-called hydrogen Zeo-Karb and sodium Zeo-Karb, here referred to as H_2Z and Na_2Z, respectively. In the sodium units, the hardness is removed and the effluent water is alkaline. In the hydrogen units, the hardness is also removed but the effluent water is slightly acid. Upon mixing, the chemical reactions of the two effluents release carbon dioxide, and the final character of the treated water is contingent upon the proportions of each. These processes are described in detail under "Chemical Performance."

Water Circuits

Fig. 1 illustrates diagrammatically the course of the make-up water from the raw city-water supply to the boilers. Starting at the supply point, the water may be fed directly or pumped into the treatment system, depending upon the pressure drop to be overcome. It first passes through the main inlet valve which is automatically opened wide or closed tightly as required by the demand for make-up. The stream then splits through two automatically controlled proportioning valves which regulate the quantities of water delivered to the hydrogen and sodium units. Following the treating units, the two streams again merge and pass through the degasifier where the major portion of the carbon dioxide is removed. From the degasifier, the water drains by gravity into the treated-water storage tank, wherein the water level is automatically controlled in accordance with demand. All pipe lines, fittings, and valves from the hydrogen units to the degasifier are rubber-lined.

As is common practice in many plants, the make-up water is taken from the storage tank into the low-pressure-turbine condensers, where it is partially deaerated and mixed with the main condensate and then delivered by pumps to the condensate system and surge tank. The demand for make-up is satisfied by controlling a valve in the treated-water make-up line to the condensers in accordance with changes of water level in the surge tank.

In addition to the supply of condensate and treated make-up water in Waterside Station No. 2, as outlined, the feedwater system is supplied by condensate from the low-pressure units in Waterside Station No. 1 and may, during emergencies, be fed by raw city water. The level in the surge tank controls the amount of condensate and raw water thus fed. All the surge tank controls are arranged in the order of preference, (1) condensate from Waterside Station No. 1, (2) treated make-up, and (3) raw city water.

Waters from these sources are caused to flow through a single mixing chamber on their way to the condensate booster pumps. This insures that all the high-pressure boilers will get the same proportion of condensate and treated make-up water and/or any raw make-up water fed during emergencies. From the condensate booster pumps, the water passes through the several deaerators (one for each pair of boilers) to the main feedwater pumps and thence to the boilers.

Proportioning Control

As the constancy of composition of the effluent of the treating system depends largely upon the accuracy of the relative proportions of water treated by the sodium and hydrogen Zeo-Karb units, it was deemed advisable to provide the special regulating equipment, shown diagrammatically in Fig. 1. One air-operated Smoot-type regulator increases or decreases the flow of water to the sodium units as the water level in the treated-water storage tank, respectively, falls or rises. In doing so, it establishes a pressure drop across a control orifice in the supply line to these units. Another Smoot-type regulator, loaded by the pressure drop across this orifice and balanced by the pressure drop across a similar orifice in the supply line to the hydrogen units, controls the flow of water to the hydrogen units in proportion to that delivered to the sodium units. A ratio adjustment on the proportioning regulator affords a means of varying the proportions as required to obtain the proper mixture of the effluents from the two groups of units. The continuously recorded pH value of the mixed effluents is used as a guide in setting the proper proportion. The sizes of the units were selected so that the effluent of the hydrogen units could be varied from 50 to 62 per cent and from the sodium units from 38 to 50 per cent.

The proportioning regulator also serves as a reliable means of preventing any flow of water through the hydrogen units without a corresponding flow through the sodium units. This fea-

ture is very desirable since any unneutralized effluent from the hydrogen units would be extremely corrosive.

The orifice-differential measurements for the proportioning controls become inaccurate when the total flow is reduced to less than 25 per cent of capacity. Means were provided, therefore, to shut off the system when the make-up demand is of this order. This was accomplished by the operation of a float switch in the treated-water storage tank which closes the hydraulically operated main inlet valve through the medium of an electrically operated pilot when the water level rises to a point corresponding to 25 per cent of capacity. After the water level has again dropped to some predetermined point, the float switch will again open the main inlet valve; thus for demands less than 25 per cent of capacity, operation of the treating system is intermittent.

Zeo-Karb Units

With the arrangement of three sodium and three hydrogen units (ultimate plant), as shown in Fig. 1, all or any combinations of pairs of units may be used for treating. Since each pair of units was designed to handle 600,000 lb of water per hr, it will be seen that the system will handle 1,200,000 lb per hr during the regeneration process on any one pair, and could handle up to 1,800,000 lb per hr in emergencies, provided the line capacities would permit.

The main theme of the designs of the sodium and hydrogen units is to provide adequate means for distributing the water evenly through the Zeo-Karb beds so that all exchange material is worked uniformly. This is accomplished by a conventional multiport inlet spray manifold and outlet drain system. The regenerating brine and acid are likewise evenly distributed over the Zeo-Karb beds by separate regeneration spray manifolds.

Consideration had to be given to the choice of materials for the sodium and hydrogen units which handle, respectively, alkaline and acid water. For the sodium units, the shells are made of steel with a bitumastic lining. The underdrain system for the sodium units is laid in concrete. For the hydrogen units, the steel shells and the underdrain systems are rubber-lined.

In addition to softening the water, the Zeo-Karb beds also filter it and, while this is advantageous, it eventually increases the pressure drop through the units and hampers the softening process. It is necessary, therefore, that the beds be backwashed from time to time by flowing water through them in the reverse direction to the backwash sumps. Float-operated backwash rate-of-flow controllers automatically regulate the backwash flow so that the velocity of water up through the units will be sufficient to dispel the foreign matter without carry-over of the Zeo-Karb material.

Regeneration

The equipment for regeneration of the sodium units consists of a galvanized-iron brine-measuring tank and two water ejectors (one a spare). To regenerate a unit, it is necessary only to shut off the valves for the softening process, open a drain valve to the backwash sump, and inject the required amount of brine through the unit. Following regeneration, the unit has to be rinsed by passing water through the unit to waste at the rate of about 1.5 gpm per sq ft of bed area. When analysis of the rinse water leaving a unit shows it to be free from calcium and magnesium salts, the softening process may be resumed.

Brine for regenerating the sodium units is prepared by the storage of rock salt under water in two 20-ton brine-storage tanks which are adequate for bulk delivery. To transport the brine from the storage tanks to the measuring tank, two 40 gpm centrifugal pumps were provided. Each tank is provided with a gravel filter bed about 1 ft deep, over which the salt is stored. Level-operated make-up valves keep the tanks full of water at all times so that the saturated brine is always available. Three large charging holes are provided in the roof of each tank, spaced in such a manner as to permit leveling off the salt bed.

The equipment for regenerating the hydrogen units consists of a totally enclosed concentrated-acid measuring tank, a lead-lined acid-mixing tank, and two dilute-acid ejectors (one a spare). The acid is delivered to the measuring tank by two LaBour self-priming centrifugal pumps. After the required amount of concentrated acid has been pumped into the measuring tank, the acid is then drained into water in the mixing tank and diluted to a concentration of about 7 per cent. Air-jet agitators are provided in the mixing tank to render a uniform solution. In delivering the diluted acid to the hydrogen unit to be regenerated, the water used for injection dilutes the acid still further to about 2 per cent. Following regeneration, the unit is rinsed by flowing water down through the bed to waste at a rate of about 4 gpm per sq ft of bed area until the required free-mineral acidity is obtained.

In order to accommodate bulk delivery of acid, two 15-ton acid-storage tanks were provided and installed in a sidewalk vault just outside the station. These tanks are made of steel and designed in accordance with the A.S.M.E. Unfired Pressure Vessel Code for 25 psi working pressure, even though they will be subjected only to atmospheric pressure.

Degasification

A degasifier is provided to remove the carbon dioxide from the water, after the effluents from the sodium and hydrogen units have been mixed. The principle of this degasifier is based upon the physical phenomenon wherein the relative amounts of gases dissolved in water are proportional to the partial pressures of these gases surrounding the water. Accordingly, the degasifier is designed so that air may be blown through the carbon-dioxide-laden water as it is cascaded over a number of wooden slats on its way to the degasifier outlet. This process does not completely remove the carbon dioxide from the water but reduces it considerably as will be noted from the results which follow.

Leaving the degasifier, the treated water is saturated with oxygen and contains some residual carbon dioxide, which renders it corrosive. By spraying the treated water into the low-pressure unit condensers, partial deaeration is effected and a small amount of heat is recovered.

Chemical Treatment

Although the Permutit system for pretreating the make-up water removes most of the hardness from the feedwater, the following additional equipment is required for maintaining the proper proportions of treatment chemicals in the boiler water:

1 Caustic feed tanks and piping to feed sodium hydroxide into the feedwater line at the deaerator outlet.

2 Phosphate feed tanks and pumps to feed disodium phosphate directly into the boiler drums to maintain a safe excess at all times. This excess will treat any hardness in the water which may enter the system due to condenser leakage or emergency use of city water.

3 A continuous blowdown system to maintain the total solids concentration in the boiler at a sufficiently low point to prevent carry-over.

Continuous-Blowdown System and Flash Tanks

Three blowdown orifices are provided for each high-pressure boiler for the regulation of blowdown. The orifices are designed for 1, 2, and 4 per cent of maximum boiler capacity. Thus, by selection of various combinations of these orifices, amounts of blowdown in steps of from 1 to 7 per cent, inclusive, may be accommodated.

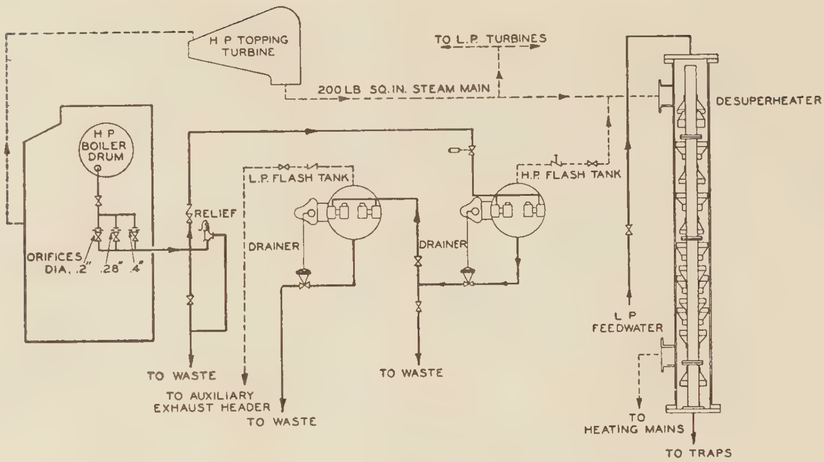

FIG. 2 BLOWDOWN SYSTEM

From these orifices, the blowdown may be discharged to waste or delivered to a blowdown flash system furnished by The Babcock & Wilcox Company, as shown in Fig. 2. The system consists of a high-pressure tank which flashes saturated steam to the 200-lb steam mains and a low-pressure tank which flashes to the exhaust header. These tanks are arranged in series so that the drains from the high-pressure tank flash to the low-pressure tank and thence to waste. Bailey automatic level-control drainer valves are provided in the drain lines from each tank. The flash system is designed to handle 350,000 lb per hr of blowdown from the high-pressure boilers, of which 27 per cent is flashed in the high-pressure tank and 13 per cent in the low-pressure tank.

CHEMICAL PERFORMANCE

The purpose of this installation, to provide make-up water for the 1400-psi boilers which is practically free of scale-forming compounds, has been adequately accomplished. During the process, the pH value of the treated water is maintained at a sufficiently low value before degasification to insure removal of carbon dioxide by aeration and at a sufficiently high value after degasification to preclude the possibility of corrosion of feed lines, valves, and other exposed metal surfaces. As the make-up water is sprayed into the condensers, the concentrations of carbon dioxide and oxygen are further reduced to a sufficiently low value which insures their complete removal by deaerating heaters.

These processes are described as follows:

When raw water, containing calcium and magnesium bicarbonate, calcium and magnesium sulphate and various other compounds of calcium and magnesium, is passed through properly regenerated sodium and hydrogen Zeo-Karb units the following chemical reactions will occur:

$Ca(HCO_3)_2$ Calcium bicarbonate, a scale-forming compound	+	Na_2Z Sodium Zeo-Karb	$\longrightarrow$ Reacts to form	CaZ calcium Zeo-Karb, which remains in the treatment unit	+	$2NaHCO_3$ Sodium bicarbonate, which is a soluble compound and remains in water
$CaSO_4$ Calcium sulphate, a scale-forming compound	+	H_2Z Hydrogen Zeo-Karb	$\longrightarrow$ Reacts to form	CaZ calcium Zeo-Karb, which remains in treatment unit	+	H_2SO_4 Sulphuric acid, which is a soluble compound and remains in water

Upon mixing the products of the foregoing reactions, the mixed effluent will react further as follows, providing the proper ratio of Na_2Z to H_2Z treated water is maintained:

$2Na_2HCO_3$ Sodium bicarbonate	+	H_2SO_4 Sulphuric acid	$\longrightarrow$ Reacts to form	Na_2SO_4 Sodium sulphate	+	$2H_2O$ Water	+	$2CO_2$ Carbon dioxide

Similarly:

$CaSO_4$ Calcium sulphate	+	Na_2Z Sodium Zeo-Karb	$\longrightarrow$ Reacts to form	CaZ Calcium Zeo-Karb	+	Na_2SO_4 Sodium sulphate
$Ca(HCO_3)_2$ Calcium bicarbonate	+	H_2Z Hydrogen Zeo-Karb	$\longrightarrow$ Reacts to form	CaZ Calcium Zeo-Karb	+	$2H_2CO_3$ Carbonic acid

Again with balanced proportions:

Na_2SO_4[a] Sodium sulphate	+	$2H_2CO_3$ Carbonic acid	$\longrightarrow$ Forms	[a]Na_2SO_4 Sodium sulphate	+	$2H_2O$ Water	+	$2CO_2$ Carbon dioxide

[a] Na_2SO_4 does not enter the reaction.
NOTE: In all of the foregoing equations magnesium may be substituted for calcium with the same result.

For these reactions it is assumed that there is no excess of either effluent. In practice, however, the proportions are maintained so that there is a slight excess of effluent from the sodium units which allows a small amount of sodium bicarbonate to pass through the degasifier, unneutralized by the acid effluent.

Thus, at the point where the effluents of the hydrogen and sodium units are mixed, the treated water contains sodium chloride which is not affected by the treatment, sodium sulphate which has been converted from calcium and magnesium sulphate, carbon dioxide which results from the reaction between sulphuric acid and sodium bicarbonate and from the breakdown of the carbonic acid, and a slight excess of sodium bicarbonate. The subsequent aeration and degasification of the mixed sodium and hydrogen effluent results in almost complete removal of the carbon dioxide.

Table 1 is indicative of the performance of these units in the treatment of Croton-supply water which is normal make-up water for Waterside Station No. 2.

Although, as shown in Table 1, the degasification of the mixed effluent results in a reduction in CO_2 concentration from 30 to 2 and a consequent decrease in total solids concentration of from 65 ppm in the untreated water to 38 ppm in the treated water at the degasifier outlet, the current of air used to remove the carbon dioxide saturates the water with oxygen.

Deoxygenation of the treated water, accomplished by spraying it into a turbine condenser where it is exposed to a high vacuum and finally passing it through the deaerating feedwater heaters along with low-pressure condensate, removes the last traces of oxygen and carbon dioxide as shown in Table 2.

TABLE 1 CHEMICAL PERFORMANCE OF ZEO-KARB UNIT

Chemical concentration, expressed as ppm	Raw water, Croton supply	Sodium-Zeo-Karb-unit effluent	Hydrogen-Zeo-Karb-unit effluent	Mixed effluent of degasifier, outlet[a]
Total hardness, as $CaCO_3$[b]...	45	0	0	0
Calcium hardness, as $CaCO_3$..	30	0	0	0
Magnesium hardness, as $CaCO_3$..................	15	0	0	0
M.O. alkalinity, as $CaCO_3$....	34	29	—16[c]	4
Sodium alkalinity, as Na_2CO_3	0	0	0	0
Free carbon dioxide, CO_2.....	3	3	30	2
Chloride, as NaCl...........	7	7	7	7
Sulphate, as Na_2SO_4.........	18	18	18	18
Silica, SiO_2................	6	6	6	6
Total solids................	65	60	31	35
pH........................	7.2	7.0	3.7	6.3

[a] Ratio of hydrogen to sodium effluent 56 : 44.
[b] American Public Health Association soap test.
[c] Free-mineral acidity, expressed as minus alkalinity.

TABLE 2 DEOXYGENATION OF PERMUTIT-TREATED WATER AT CONDENSER-HOTWELL DISCHARGE AND DEAERATING-HEATER OUTLET

Rate of flow of Permutit-treated water, lb/hr × 1000	Rate of flow of turbine condensate, lb/hr × 1000	Oxygen concentration, ppm O_2: Degasifier outlet	Hotwell discharge	Deaerator: Inlet	Deaerator: Outlet	Feedwater temperature at deaerator outlet, deg F
200	300	8.5	0.3	0.3	0.005	220
300	300	9.0	0.3	0.3	0.005	220
400	300	9.0	0.5	0.7	0.006	219
500	300	9.0	1.7	0.9	0.007	220

It will be noted that the oxygen concentration in the deaerated feedwater is negligible. The carbon-dioxide concentrations at the hot-well discharges and the deaerator outlets are also negligible. This has been verified by the condition of the internal surfaces of the economizer tubes which are practically free of corrosion.

Operating Experience

Because of load conditions at the present time it is only necessary to operate one set of treatment tanks. A second set is maintained in a regenerated condition and, when the set in operation is chemically exhausted as indicated by analyses of the treated water, which are performed by the operator, the second set is placed in operation.

Experience has indicated that it is good practice to give a set of units, which have been held in a regenerated condition, a secondary rinse before being placed in operation. Chemical analyses of the rinse and of the treated water, immediately after rinse, are used as an index of rinse period.

Under present conditions of operation, the maximum rate of treated-water flow has been approximately 30 per cent of total evaporation. The present steaming capacity of the boilers is approximately 2,000,000 lb per hr.

Seasonal changes in water composition preclude the possibility of maintaining a constant ratio of hydrogen-to-sodium unit effluent. It has, therefore, been necessary to vary the proportion of the sodium effluent from approximately 44 to 48 per cent. The amount of water passing through each of the units is integrated on individual meters on each treatment tank. These readings are used to ascertain the proportion at approximately -hr intervals. Experience has shown that constant determination of pH value is the best index of treated-water composition. A glass pH electrode cell and recorder have therefore been installed to record the pH value of the treated water at the degasifier outlet. It has been found that, if the pH value of the water at this point is maintained between 5.5 and 6.5, satisfactory performance results.

The operation of the treatment units under ordinary conditions is handled entirely by the operator on watch. This includes regeneration of the units, backwash, primary rinse, and necessary chemical analyses of the treated water. The operator checks the performance of the operating unit by the integrating-water-meter readings and the pH value of the water at the degasifier outlet to determine the proper ratio of each unit effluent. The following chemical analyses are considered adequate for proper control:

Croton water	Sodium-unit effluent	Hydrogen-unit effluent
Alkalinity to M. O.	Alkalinity to M. O.	Free mineral acidity
Hardness	Hardness	Hardness

These analyses are made from three to five times for each softening cycle. Fig. 3 illustrates the typical changes of hardness in the effluent from a sodium unit during one complete cycle between regenerations, and Fig. 4 illustrates the changes of free-mineral acidity in the effluent from a hydrogen unit during one of its cycles.

The operators have been trained to make certain interpretations of the chemical-test results. If any unusual condition arises which the operator cannot correct, a staff chemist is immediately consulted. If the condition arises at a time when a chemist is unavailable, the set of tanks in operation is immediately taken out of service and the second stand-by set is cut in. The units which indicated unusual conditions are then regenerated and rinsed and the rinse water subjected to chemical analysis. Up to the present time, any unusual chemical data have been found to result from faulty regeneration or rinse procedure. The performance of the units is checked daily by a chemist to insure proper operation.

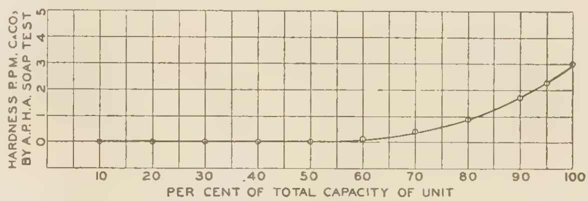

Fig. 3 Approximate Performance of Sodium Unit

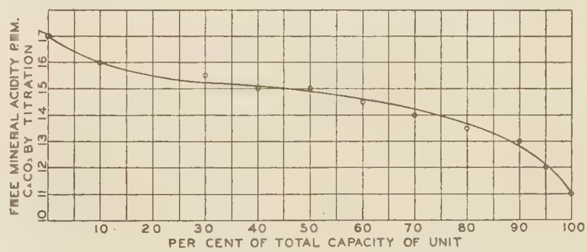

Fig. 4 Approximate Performance of Hydrogen Unit

Effect of Permutit Operation on Internal Boiler Conditions

Since all the Permutit-treated water is fed to the high-pressure boilers, it is important that the concentration and proportions of the various salts present in the high-pressure-boiler water be carefully controlled. As previously stated, the internal feedwater treatment for the high-pressure boilers consists of sodium hydroxide and disodium phosphate in addition to the external treatment given the raw Croton water by the Permutit installation. The sodium hydroxide is introduced at the deaerator outlets in sufficient quantity to confer a pH value of approximately 8.8 upon the feedwater which passes through economizers. The disodium phosphate is admitted directly to the boiler drum.

Because of critical circulation and carry-over conditions, it is necessary to maintain a total dissolved-solids concentration below 450 ppm in the boiler water. In addition, it has been found that high hydroxide concentration has resulted in foaming of boiler water which causes fouling of the turbine blading. The following limits have therefore been prescribed for the high-pressure-boiler water composition:

		Maximum ppm	Minimum ppm
Phosphate	Na_3PO_4	25	15
Sodium hydroxide	NaOH	30	20
Sodium chloride	NaCl	150	..
Sodium sulphate	Na_2SO_4	250	..
Total dissolved solids		450	..

In addition to the operation of the Permutit-treatment plant, the operator performs analyses of the high-pressure-boiler water at 2-hr intervals and advises the boiler operator as to hydroxide and phosphate feed and blowdown procedure.

In order to remove sludge from the lower tubes and headers of the boiler, it has been found advisable to mass-blow water from the sectional header and waterwalls. This is done at approximately 4-day intervals at a pressure of approximately 300 psi. It has been found that this practice eliminates heavy accumulations of sludge in the sectional-header pockets and tubes.

After a year of operation of the Permutit-treatment installation, during which the only water supplied to the high-pressure boilers was condensate from the low-pressure turbines and Permutit-treated Croton water, internal inspections of these boilers show a uniformly good condition. It has also been determined, as a result of recent inspections, that the economizers are free from deposits and corrosion.

Maintenance

The maintenance of the treatment plant has been negligible during its first year of operation. There are no indications at the present time that any increased maintenance will be necessary.

Acid and Salt Handling

The handling of the sulphuric acid and salt has been satisfactory and no problems have arisen to date from an operating point of view. The treatment-plant operators have been trained by competent chemists in the handling of the acid and in the importance of using the protective equipment with which they are supplied. In addition, the acid-handling equipment for making up the dilute solution used for regeneration is so interlocked that it is practically impossible for an operator to add water to concentrated sulphuric acid instead of adding acid to the water. Printed instructions on the operation of the plant are conspicuously posted, and caution signs are prominently displayed at the acid tanks and pumps.

Discussion

L. D. Betz[4] and R. T. Sheen.[5] This paper is an interesting report on the application of a relatively new method of water treatment, namely, the application of zeolite softening in the hydrogen cycle and the advantages that can be obtained by such softening.

While there is no known chemical method of water conditioning today that would give a lower total solids content in the treated water, with the possible exception of a process involving anion exchange, it may be interesting to compare the method described with methods that were available prior to the advent of softening in the hydrogen cycle. Were it not for the relatively low total solids concentration which may be tolerated in the boiler water, this alternate method could possibly have been considered; by alternate method, reference is made to a system of treatment with softening by a carbonaceous zeolite in the sodium cycle, followed by sulphuric-acid treatment and aeration. Considering the requisites of the system as outlined by the authors, such a method of treatment would give a water higher in total solids than that available with the hydrogen zeolite but would involve a smaller treating plant, a desirable feature as outlined in the authors' third requisite. The following items of equipment would be eliminated:

1 The automatically controlled Smoot regulating-proportioning valves, with the possible danger of malfunctioning of these units.

2 The rubber-lined shell and hydrogen-zeolite units.

3 Rubber-lined piping.

4 Sulphuric-acid-dilution system.

5 Sulphuric-acid feed-and-injection system.

To replace these units the following equipment would be required:

1 Additional capacity for softening on the sodium-exchange cycle with carbonaceous zeolite.

2 Concentrated sulphuric-acid-proportioning system to water entering the degasifier.

From the standpoint of handling the sulphuric acid, the necessity of diluting the acid would be eliminated. It is evident that the physical size of the treating plant would be somewhat smaller.

To compare the two systems properly, an evaluation of the chemical performance, which could be expected, should be made. Refer to Table 1, appearing in the paper. In Table 3 of this

TABLE 3 ANTICIPATED RESULTS FROM WATER-SOFTENING PROCESS

	Raw-water Croton supply	Sodium-zeolite effluent	Sodium-zeolite effluent, acid-treated and aerated
Total hardness, as $CaCO_3$	45	0	0
Calcium hardness, as $CaCO_3$	30	0	0
Magnesium hardness, as $CaCO_3$	15	0	0
Methyl-orange alkalinity, as $CaCO_3$	34	34	4
Free carbon dioxide, as CO_2	3	3	2
Chloride, as NaCl	7	7	7
Sulphate, as Na_2SO_4	18	18	60
Silica, as SiO_2	6	6	6
Sodium (by difference), as Na	3	24	24
Total solids (by authors)	65	..	..
Calculated total dissolved solids	61	66	77

discussion, the same analysis is shown for the raw-water Croton supply and, for the sodium-zeolite-unit effluent. We have assumed the theoretical sodium-zeolite-softener effluent which should indicate no difference in the methyl-orange alkalinity and also should indicate some increase in total solids rather than a decrease as shown in the paper. This increase in solids is to be normally expected when the equivalent weights of the substances entering the base exchange are noted to be 12.2 for magnesium, 20 for calcium, and 23 for sodium. In other words, for each 12.2 parts by weight of magnesium removed, 23 parts of sodium are introduced and for each 20 parts of calcium removed, 23 parts per weight of sodium are introduced. On this particular analysis, an increase of about 5 ppm in solids is to be expected in the sodium exchange cycle. In the third column of Table 3, is shown an analysis of sodium-zeolite-softened water treated with sulphuric acid and aerated to give the same residual alkalinity as shown in the paper by the authors. An appreciable increase in sodium sulphate is indicated and the calculated total dissolved solids is shown as 77 ppm. Herein lies the disadvantage of this method of treatment compared to the combination-method soften-

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ing in the hydrogen cycle, namely, 77 ppm of total dissolved solids in the make-up water, compared to 35 ppm, as shown in the mixed effluent of the degasifier outlet by the combination softening system.

The authors indicate that about 70 per cent of the boiler feedwater is condensate and 30 per cent make-up water. While the total dissolved solids shown by the authors is a maximum for boiler-water concentration, approximately 35 ppm in the boiler water will have been added by the introduction of internal-treating chemicals, leaving about 415 ppm of solids which may be present because of concentration of solids in the feedwater.

	Sodium zeolite; hydrogen zeolite	Sodium zeolite treated with sulphuric acid plus aeration
Total solids in make-up water, ppm..........	35	77
With 30 per cent make-up water, total solids in feedwater, ppm........................	10.5	23
Cycles of concentration (415 ppm total solids by concentration)........................	40	18
Blowdown to maintain total solids within limits, per cent........................	2.5	5.5

Because of the increased rate of blowdown required by the method of treatment employing sulphuric acid and aeration, this method suffers materially in comparison with the treatment method by sodium and hydrogen zeolite, as it will be noted that more than twice the rate of blowdown would be required.

No comments are made on the silica concentrations found in the boiler water. On the silica content of the make-up water, as shown by the authors and with the cycles of concentration as indicated, a silica content of boiler water of approximately 72 ppm could be anticipated. The question is raised as to whether it may not be the silica concentration of the boiler water which is the limiting factor on the total solids that are now carried. If this should be the case, a higher total solids could be carried with the sodium-zeolite acid and aerated water used as boiler feed. With the present make-up water, the silica represents 17 per cent of the solids in the make-up water, shown by the mixed effluent of the degasifier outlet. On the acid-treated and aerated water, the silica content is 8 per cent of the total solids. It should be pointed out, however, that this lower percentage of silica by the acid-treated and aerated system is not due to any removal of silica but rather due to the increase in other solids in the water by the introduction of the sulphate ion. However, such a balance of solids might permit the maintenance of boiler-water solids at a somewhat higher level and overcome to some degree the great difference in the rate of blowdown shown to be necessary by the two possible systems.

This discussion is presented not in any attempt to criticize the selection of a combination-treatment system, consisting of the sodium-zeolite and hydrogen-zeolite method of softening in any way, but simply to enlarge upon its possible value, as well as to indicate other possible methods of treatment which might be employed for the solution of such a feedwater-treating problem. It is quite likely that the authors considered such a system, as outlined in this discussion. The results which have been obtained have been very well reported by the authors and this presentation is a valuable contribution to the published literature on this subject.

Authors' Closure

The authors wish to express their appreciation for the helpful discussion presented on this paper.

Consideration had been given to the alternate treating scheme mentioned by Messrs. Betz and Sheen, whereby the water would be softened by carbonaceous-zeolite units with subsequent acid injection and aeration. Despite the fact that this might have provided a cheaper initial installation, it would be far less economical to operate. With regard to the initial cost, since the same amount of softening material would have been required in either case and since some form of acid-proportioning equipment would be required in either case, it is felt that the cost of the rubber linings eliminated by the alternate scheme would constitute the only major saving.

Where hydrogen Zeo-Karb units are employed to produce an acid effluent to neutralize the alkalinity of the sodium effluent, considerable softening (in our case over 50 per cent) is done in the hydrogen or acid units. It is felt that this dual function of the hydrogen units results in sufficiently lower costs of the softening reagents to justify the expense of the rubber linings involved.

As stated, by Messrs. Betz and Sheen, the solids concentration in the water treated by the alternate scheme would be 77 ppm or about twice as much as is obtained in the water treated by the sodium and hydrogen Zeo-Karb combination. They suggested, however, that this would be of little consequence if the concentration of silica in the boiler water were the limiting factor. That is, the rate of blowdown could be held the same as now prevails, provided the total concentration in the boiler water were allowed to double, which, of course, could be done without increasing the silica concentration now prevailing.

In regard to this point, the authors doubt that it would be possible greatly to increase the boiler-water concentration without excessive carry-over from the boilers under present conditions. Hence, in all probability, it would be necessary to double the blowdown loss with the alternate scheme of feedwater treatment. This would amount to a loss equivalent to about 1.5 per cent of boiler efficiency.

Even with the relatively low concentrations now being maintained, in the high-pressure boilers, there is some carry-over of which a portion is silica. This silica, however, has been found only in the feed-heating section of but one of the two high-pressure turbines and in the latter stages of some of the low-pressure turbines. No silica scale has been found in the boilers. If a lower concentration of silica in the boiler water should be advantageous, the quantity of blowdown could be doubled with the present system, which, of course, would halve the silica concentration which now prevails. Under these conditions, there would probably be the additional advantage of a higher-quality steam, caused by correspondingly lower total-solids concentration in the boiler water.

Adsorption Process for Removal of Soluble Silica From Water

By L. D. Betz,[1] C. A. Noll,[2] and J. J. Maguire[3]

Magnesium oxide has been shown to be particularly valuable in the removal of soluble silica from water. This paper deals with an adsorption process for silica removal which has been recently developed. This process is particularly well adapted for use in conjunction with hot-process lime-soda softening, causing no increase in the quantities of either lime or soda ash. Removal of silica from solution is effected, not by a chemical reaction but by adsorption from solution. Adsorption data correspond to Langmuir and Freundlich adsorption isotherms. An outstanding advantage of this process is the fact that the solids content of the treated water is decreased, rather than increased. No increase in the total solids content of the treated water results as would be the case with the use of reagents such as ferric sulphate and magnesium sulphate. In common with other adsorption processes, the last traces of silica are the most difficult to remove. To illustrate what can be accomplished in this respect data are presented from an actual full-scale plant test in which a silica content of 6.3 ppm in the raw water was reduced to an average of 0.6 ppm in the treated water. Experience has also been gained in the treatment of a natural water in which the silica content was reduced from 56 ppm to 1 ppm in the treated water.

The difficulties attendant upon the presence of silica in boiler feedwater have become problems of major importance in the design and operation of high-pressure power plants. No proposed power-plant design is complete without a detailed investigation of the type of feedwater conditioning that may be applied. One of the most important factors to be considered in such an investigation is the silica content of the make-up water and the steps that may be taken for reduction of silica to a tolerable value. In many plants the permissible silica content of the concentrated boiler water is the determining factor limiting the type of feedwater conditioning to be applied.

Such requirements may stipulate that the silica content of the boiler water not exceed 10 ppm. If make-up water is to constitute more than a very minor percentage of the total feedwater, it is usually necessary to design the feedwater-treatment plant with the primary purpose of reducing the silica content. In many cases the stipulation of such low silica content of the boiler water is designed to provide against siliceous deposits on turbine blading.

The deposition of calcium- and magnesium-silicate scales can usually be prevented in low-pressure operation by intelligent control of boiler-feedwater conditioning unless the silica content of the boiler feedwater is in excess of 5 or 10 ppm. In high-pressure operation, however, very minor quantities of silica in boiler feedwater have resulted in the formation of complex silicate scales.

Sodium-alumino-silicate scales are normally referred to as analcite scales. Analcite has the formula $Na_2O \cdot Al_2O_3 \cdot 4SiO_2 \cdot 2H_2O$. Other sodium-alumino-silicate scales have been identified by X-ray-diffraction methods which provide the most reliable means of identifying these complex crystalline formations. Among the formations identified in scale from high-pressure boilers are cancrinite, $Na_8Al_6Si_9CaO_{35} \cdot 2CO_2 \cdot 3H_2O$; noselite, $Na_8Al_6Si_6O_{24} \cdot SO_4$; and sodalite, $Na_8Al_6Si_6O_{24} \cdot Cl_2$.

The presence of aluminum can be noted in all of these deposits. Formation of such complex scales has frequently led to the discontinuance of aluminum coagulants, particularly inasmuch as aluminum becomes soluble to an appreciable extent at pH values above 8 (1).[4] Use of various forms of aluminum as a coagulant in hot- or cold-process lime-soda softening, therefore, may introduce the undesirable aluminum ion into the boiler feedwater.

Silica is conventionally expressed as SiO_2 in a water analysis. Various investigators have made clear the distinction between crystalloidal and colloidal silica (2, 3, 4). They have indicated the major portion of the silica found in natural waters to be in the crystalloidal state. Silica present suspended in water in the form of mud can be removed by coagulation and filtration. The major problem in the conditioning of boiler feedwater is the removal of soluble silica from solution.

Various methods for the removal of silica from boiler feedwater have been presented by different investigators. Among these has been the work of Schwartz on ferric sulphate and hydrous ferric oxide. He clearly outlines the inherent advantages and disadvantages of the use of these materials (4). It has been known that a certain amount of silica removal is effected in hot-process lime-soda softeners where there is an appreciable magnesium content of the raw water. Magnesium sulphate has been used in some instances for the purpose of silica removal. However, until recently, no process was available that would effect any desired degree of silica removal without an increase in the solids content of the treated water, usually to such an extent as to prevent the practical use of this treated water.

An adsorption process for silica removal has recently been developed which permits reduction in soluble silica to practically any desired value with a decrease in solids content of the treated water, rather than an increase. Some laboratory data on this process have already been described by the authors (5, 6).

Various forms of magnesium oxide have been tested from the relatively impure materials such as dolomitic lime, a mixture of calcium oxide and magnesium oxide, and calcined magnesite to the finer materials of U.S.P. grade. The material found most efficient and economically feasible for silica removal is a type of specially prepared adsorptive form of magnesium oxide, prepared from sea-water bitterns by the California Chemical Company, and termed Remosil for identification purposes. Unless specifically stated otherwise results described were secured with the use of this material.

Colorimetric silica was determined with the use of a Taylor

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

analyzer and by a Klett-Summerson photoelectric photometer using the Betz photometric method (7). Gravimetric silica was determined in the conventional manner (8).

Effect of Temperature

A very important factor is the temperature at which silica removal is effected by means of magnesium oxide. The efficiency of silica removal increases with increase in temperature. Fig. 1

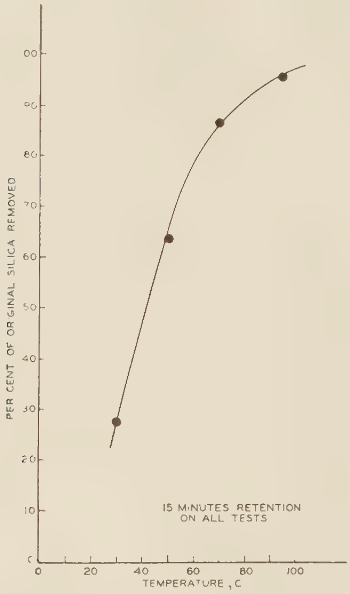

Fig. 1 Effect of Temperature

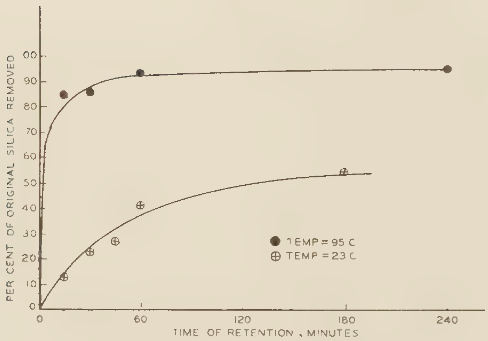

Fig. 2 Effect of Retention

illustrates the effect of temperature, showing the major increase in percentage silica removal resulting with an increase in temperature up to 95 C, the approximate temperature of operation of hot-process softeners.

Effect of Retention Time

Where the silica content of the raw water is to be reduced to the range of 2 to 5 ppm, little is to be gained by increase in time of retention above 1 hr at 95 C. In fact, equilibrium is reached almost completely in 15 min. In the cold, however, equilibrium is reached much more slowly. This is shown in Fig. 2. The data employed in plotting each curve were secured through the treatment of the same water with identical quantities of magnesium oxide. All conditions were held constant in each case with the exception of the fact that one series of tests was conducted at a constant temperature of 95 C and the other series was conducted at a constant temperature of 23 C.

Greater silica removal is effected at 95 C than at 23 C by the same quantity of magnesium oxide, as noted from these curves. The removal of silica is 85 per cent complete in 15 min and retention over 1 hr yields little advantage at the higher temperature. At the lower temperature, however, there is a steady increase in the percentage silica removal with increase in retention time, although this curve too flattens with longer periods of retention.

Effect of pH

Schwartz (4) has shown that the optimum pH value for removal of silica by ferric sulphate was 9 for the waters investigated. An investigation by the authors has indicated that the optimum pH range for silica removal by means of aluminum hydroxide to be 8.3 to 9.1. However, at these pH values the aluminum ion goes into solution making a pH range of 7.6 to 8.0 more suitable from a practical standpoint (1). Investigations of the

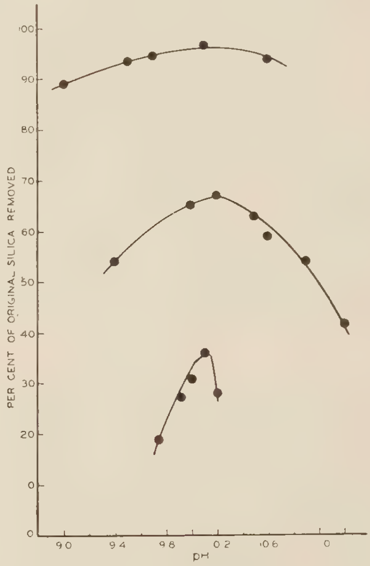

Fig. 3 Effect of pH

effect of pH on removal of silica by means of magnesium oxide have indicated the optimum pH value to be approximately 10.1.

Fig. 3 illustrates the results obtained in the treatment of three different waters for varying degrees of silica removal. For each curve a constant quantity of magnesium oxide was employed, and the only factor varied was the pH. The point of maximum silica removal is observed in each case to occur at a pH of approximately 10.1. It can also be noted from these curves that, with successive increases in the percentage of silica removal, the control of pH becomes less critical.

The optimum pH of 10.1 for maximum silica removal is readily obtainable with either hot- or cold-process lime-soda softening. On the other hand, with a pH of 9 for best results with ferric sulphate and a pH of 7.6 to 8 for best results with aluminum hydroxide, it is evident that neither of these materials can be used with full efficiency in conjunction with lime-soda softening.

Mechanism of Silica Removal by Magnesium Oxide

In the various softening reactions normally encountered in water conditioning there is a stoichiometric relation or a constant reacting value. For example, a definite quantity of soda ash or disodium phosphate will always react with a definite amount

TABLE 1 ADSORPTION DATA

Magnesium oxide, ppm	Residual silica, ppm	Silica removed, ppm	Original silica removed, per cent	Silica removed per part MgO
0	19.9	0.0	0.0	...
10	11.9	8.0	40.0	0.80
30	5.9	14.0	70.2	0.465
60	3.0	16.9	85.0	0.28
100	2.0	17.9	90.0	0.18
130	1.2	18.7	94.0	0.14
0	10.1	0.0	0.0	...
10	7.1	3.0	29.7	0.30
30	4.4	4.7	56.5	0.19
50	3.0	7.1	70.3	0.14
70	2.4	7.7	76.2	0.11

NOTE: Conditions: 30-min stirring-and-retention time; temperature, 95 C.

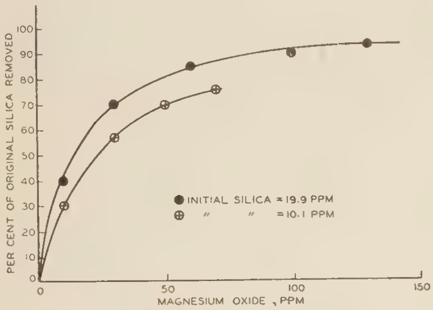

FIG. 4 ADSORPTION OF SILICA BY MAGNESIUM OXIDE

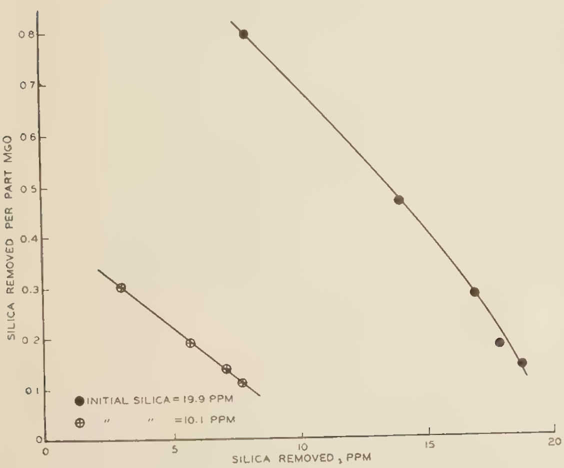

FIG. 5 DECREASE IN ADSORPTION EFFICIENCY

of calcium sulphate in accordance with the law of combining weights. Such a reaction is independent of the initial and final concentration of calcium sulphate. The removal of silica by means of magnesium oxide, however, is not a stoichiometric reaction but proceeds by adsorption from solution. Adsorption is conditioned by the extent of surface exposed, by concentration, by temperature, as well as the specific nature of the adsorbent and adsorbed substance. Adsorption data (9) on two waters of different silica content are presented in Table 1.

The relationship between the percentage silica removal and the quantity of magnesium oxide employed is shown in Fig. 4. A straight line would be characteristic of a stoichiometric chemical reaction, but, as can be noted, the line for each water rises sharply at first and then gradually flattens. The need for greatly increased quantities of magnesium oxide to effect removal of the last 10 to 20 per cent of the original silica content is characteristic of adsorption reactions.

As can be expected with this type of reaction, the efficiency of silica removal (quantity of silica removed per part of magnesium oxide) decreases with increased removal of silica from solution. Fig. 5 illustrates this fact and indicates that, as larger quantities of silica are removed from solution in a given water, each increase in the amount of magnesium oxide removes successively less silica.

This adsorption characteristic should not be confused to mean that the higher the initial silica content of a given water, the less silica will be removed per part of magnesium oxide. On the contrary, as can be seen from Fig. 5, and Table 1, the higher the initial silica content, the higher the quantity of silica removed per part of magnesium oxide. However, for any definite initial silica content of a raw water, as the amount removed from solution increases with increase in quantities of magnesium oxide, the silica removed per part of magnesium oxide decreases.

The empirical equation of Freundlich (10) is generally used to determine if a reaction proceeds by adsorption. This reaction may be expressed as

$$\frac{x}{m} = KC^{1/n}$$

where K and $1/n$ are constants

x = quantity of material adsorbed
m = quantity of adsorbent
C = residual concentration of adsorbed material at equilibrium

When the logarithm of x/m is plotted against the logarithm

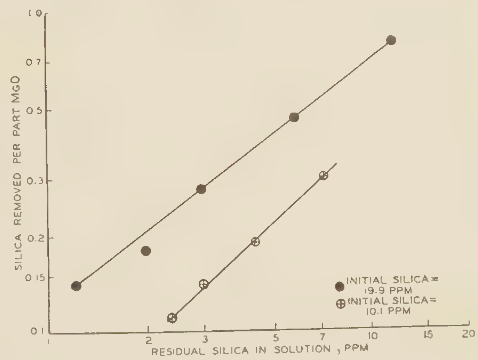

FIG. 6 FREUNDLICH ADSORPTION ISOTHERM

of C, a straight line will result if the reaction involved corresponds to the foregoing equation. Plotting on a double logarithmic scale the silica removed per part of magnesium oxide against residual silica in solution, as shown in Fig. 6, indicates the removal of silica from solution by magnesium oxide to conform with a Freundlich adsorption isotherm.

Although adsorption reactions are normally more efficient at lower temperatures, the adsorption of certain colors and gums (11) like adsorption of silica are more efficient at elevated temperatures.

REMOVAL OF SILICA IN CONJUNCTION WITH LIME-SODA SOFTENING

Removal of silica from solution by means of magnesium oxide can proceed simultaneously with the removal of hardness by lime-soda softening. Both reactions can take place in the same tank and one does not interfere with the other. Since the optimum pH value for silica removal, 10.1, is readily obtained with lime-soda softening, the environment provided by such softening is ideal for securing the most efficient removal of silica from solution.

No increase is occasioned in the amount of either lime or soda

ash used where magnesium oxide is employed in the removal of silica. Requirements of these softening chemicals remain unaffected by the magnesium oxide, whereas, the use of reagents such as magnesium sulphate, ferric sulphate, and dolomitic lime may require the use of large additional quantities of lime and soda ash.

Magnesium oxide acts as an excellent coagulant in conjunction with the lime-soda process and reduces the turbidity of the effluent from the sedimentation tank and filters. Where magnesium oxide is used for silica removal the need of an additional coagulant is eliminated.

The hardness and alkalinity of a properly balanced lime-and-soda effluent are not increased by the use of magnesium oxide. There is no interference with the normal softening reaction, and it is often possible with the use of magnesium oxide to achieve a lower hardness and a lower alkalinity of the softener effluent than if only lime and soda ash were employed.

TABLE 2 SILICA REMOVAL BY MAGNESIUM OXIDE IN CONJUNCTION WITH HOT-PROCESS LIME-SODA SOFTENING

Calcium hydroxide, ppm	Sodium carbonate, ppm	Magnesium oxide, ppm	Retention time, min	Analysis: Hardness as $CaCO_3$, ppm	P alkalinity as $CaCO_3$, ppm	M alkalinity as $CaCO_3$, ppm	Silica as SiO_2, ppm
..	...	...	..	34	0	50	37.6
30	15	0	15	28	36	62	37.1
30	15	200	15	28	34	66	3.6
..	...	...	..	96	0	26	21.8
55	75	0	30	36	20	46	18.4
55	75	100	30	34	18	42	1.2
..	...	...	..	120	0	10	19.9
15	140	0	30	38	34	64	19.0
15	140	100	30	32	26	54	1.0

NOTE: Conditions: Temperature, 95 C.

TABLE 3 SILICA REMOVAL BY MAGNESIUM OXIDE EMPLOYING SODIUM HYDROXIDE

Sodium hydroxide, ppm	Magnesium oxide, ppm	Retention time, min	Analysis: Hardness as $CaCO_3$, ppm	P alkalinity as $CaCO_3$, ppm	M alkalinity as $CaCO_3$, ppm	Silica as SiO_2, ppm
..	...	..	54	0	36	21.3
66	100	30	44	28	68	0.7
..	...	..	60	0	22	31.0
60	150	30	20	34	66	3.0
..	...	..	74	0	70	56
30	300	15	66	20	72	2.5

NOTE: Conditions: Temperature, 95 C.

Table 2 illustrates typical results of the use of magnesium oxide in conjunction with hot lime-and-soda softening on waters of relatively high silica content. Even the higher quantities of magnesium oxide necessary in these cases did not occasion an increase in the quantities of lime or soda ash or interfere with softening efficiency.

Silica Removal by Magnesium Oxide Employing Sodium Hydroxide

On waters of relatively low hardness it may not be desirable to attempt softening by the lime-soda process. In such cases, sodium hydroxide or lime only may be used to establish the proper pH value for most efficient silica removal. In such cases, some degree of softening may be effected.

Table 3 illustrates results obtained on waters of relatively high silica content, employing with magnesium oxide sufficient sodium hydroxide for efficient silica removal.

Silica Removal in Conjunction With Hot-Process Phosphate Softening

Where it is desired to effect practically complete removal of hardness externally, hot-process phosphate softeners are frequently employed. These may be installed taking as feedwater to the phosphate unit, the effluent of a hot-process lime-soda softener which has already effected removal of the major portion of the hardness of the raw supply. Obviously, in such cases, silica removal can be effected in the lime-soda unit, and the partially softened and silica-free effluent passed to the phosphate softener where the hardness removal is completed.

Where a phosphate softener operates directly on the raw supply without preliminary hot-process softening, it is desirable to effect silica removal and hardness reduction in two-stage operation. Silica removal can be first accomplished by magnesium oxide in conjunction with lime or sodium hydroxide and then followed by phosphate softening, after sedimentation of the precipitates from the silica-removal process. Magnesium oxide possesses the property of adsorbing phosphate from solution as well as silica, hence the desirability of two-stage operation. It is not necessary that two sedimentation tanks be employed, but the dual process of silica removal and softening can be accomplished in one tank if properly designed for the purpose.

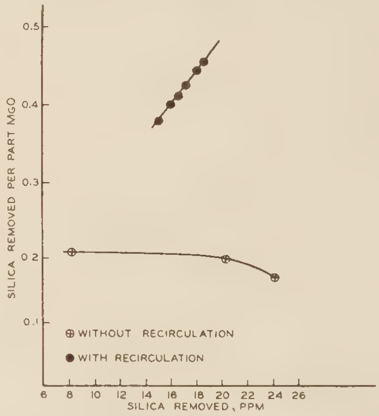

FIG. 7 EFFECT OF SLUDGE RECIRCULATION

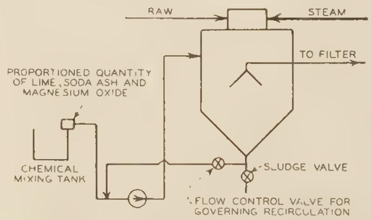

FIG. 8 RECIRCULATION OF MAGNESIUM-OXIDE SLUDGE

Phosphate softening can be applied to the effluent of the silica-removal system as represented by the treated waters in Tables 2 and 3. There will be no increase in the silica content and practically no change in the characteristics of these treated waters with the exception of reduction of hardness to zero and the maintenance of approximately 5 to 10 ppm excess phosphate.

Recirculation of Sludge

It has previously been shown by Table 1, and Fig. 5, that the silica removed per part of magnesium oxide decreases with lower residual silica in solution. Table 1 shows that, on a water of 19.9 ppm initial silica content, the use of 130 ppm magnesium oxide reduced silica to 1.2 ppm, 0.14 part silica being removed per part of magnesium oxide. Where only 30 ppm magnesium oxide were used, however, silica reduction was effected down to 5.9 ppm, and 0.465 part silica was removed per part of magnesium oxide. It is evident, therefore, that in this case the used magnesium oxide for treatment down to 1.2 ppm silica still possesses

a silica-removal capacity of 0.325 (0.465 — 0.14) part silica per part of magnesium oxide. This capacity for silica removal can be employed on the original water to reduce its silica content to 5.9 ppm. In this manner the partially spent magnesium oxide can be used for partial silica removal and thus result in the use of a lesser quantity of fresh magnesium oxide to reduce the silica from 5.9 ppm to 1.2 ppm. Recirculation of sludge provides an opportunity for increasing the efficiency of silica removal in this fashion. By recirculation the quantity of magnesium oxide necessary to accomplish a given silica removal has been reduced by as much as 60 per cent.

The steeper the slope of a Freundlich adsorption isotherm, the greater benefit is to be gained by recirculation of the partially spent magnesium oxide. Reference to Fig. 6 will show relatively steep slopes of the isotherms, indicating that considerable benefit may be derived through recirculation.

Fig. 7 illustrates in a similar manner to Fig. 5 the decrease in the silica removed per part of magnesium oxide with increased removal of silica, without recirculation of sludge. The results obtained with sludge recirculation are also shown. As can be noted the silica removed per part of magnesium oxide is considerably greater with recirculation. A major improvement in the efficiency of silica removal is thus obtained.

An interesting point also noted is that with recirculation, as greater quantities of silica are removed from solution, the silica removed per part of magnesium oxide increases. In this particular case, therefore, it is evident that recirculation has also provided a means of overcoming one of the most troublesome characteristics of adsorption. By recirculation it is actually possible to increase the silica removed per part of magnesium oxide with increasing removal of silica from solution.

The practical application of recirculation is very simple. Fig. 8 illustrates the method of sludge recirculation employed with a lime-and-soda softener. Since it is adsorption of silica by means of the recirculated magnesium precipitates which accomplishes silica removal, there is no change in the chemical balances maintained on the softener effluent. Recirculation affects the characteristics of the softener effluent only with respect to a reduced silica content.

Comparison of Various Forms of Magnesium Oxide

A number of different forms of magnesium oxide have been investigated for use in the removal of silica from water. Many

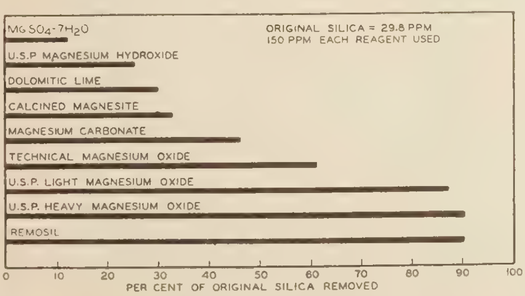

Fig. 9 Comparison of Various Forms of Magnesium Oxide

of these materials, such as, magnesium sulphate and dolomitic lime, may considerably increase the solids content of the treated water. Fig. 9, however, simply illustrates the comparison of the silica-removing capacity of each of the substances listed. Results, on which Fig. 9 are based, were obtained through the treatment of the same raw water with 150 ppm of each reagent. Efficiency of each material is expressed in terms of the percentage of original silica removed. Inasmuch as the removal of the last 10 to 20 per cent of the original silica content of the water, as previously noted, requires the higher quantities of reagent and is most difficult to remove, it is evident that those materials which removed over 80 per cent of the original silica content possess very high silica-removal capacity.

Comparative Tests on Silica Removal

One of the major advantages of the use of magnesium oxide for silica removal is that silica can be removed and the solids content of the water actually decreased by this process, rather than increased. Where magnesium sulphate or ferric sulphate are employed for silica removal, an alkali is necessary for the precipitation of the respective hydroxides. Where sodium hydroxide is employed for this purpose, it is obvious that the sodium sulphate resulting from this reaction will materially add to the solids content of the treated water. The amount of sodium

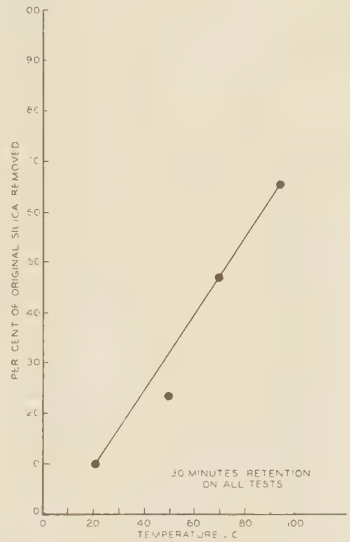

Fig. 10 Effect of Temperature on Silica Removal by Dolomitic Lime

sulphate so introduced, in the case of high-silica waters, may exceed the solids content of the raw supply by several hundred per cent.

Dolomitic lime for use in silica removal was investigated and reported upon by Behrman and Gustafson (12). These investigators concluded that this material could not be successfully used for silica removal. Tests made by the authors have shown that silica removal may be accomplished by means of dolomitic lime but the increase in solids content, either in the form of hardness or alkalinity, occasioned by the use of this material may make such a process impractical for certain water supplies.

Dolomitic lime is prepared by the roasting of dolomite which is a mixture of calcium and magnesium carbonate. Calcium oxide or unslaked lime and magnesium oxide result from this roasting process and the mixture is termed dolomitic lime. The magnesium oxide present in dolomitic lime can, if in the proper form, remove silica but introduces at the same time a large amount of calcium oxide. On a relatively soft water this calcium oxide will materially increase the hardness. The quantity of magnesium oxide, combined in dolomitic lime, that is necessary for silica removal exceeds considerably the amount of calcium hydroxide required for the removal of carbonate hardness. The increased calcium content of the water must then be removed by the use of soda ash, precipitating calcium carbonate but re-

TABLE 4 COMPARATIVE TESTS ON SILICA REMOVAL

	Original water	Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
Hardness as $CaCO_3$, ppm......	56	48	24	148	242	8	48
Sulphate as SO_4, ppm.........	14	14	484	478	14	14	276
Chloride as Cl, ppm..........	19	19	19	19	19	19	19
P alkalinity as $CaCO_3$, ppm....	0	20	68	18	132	568	12
M alkalinity as $CaCO_3$, ppm..	60	68	140	102	168	602	74
pH........................	7.1	9.3	9.8	8.9	11.1	11.7	9.0
Silica as SiO_2, ppm...........	48	2.5	3.0	3.8	2.8	18.4	14.6
Silica removed per part reagent used......................	..	0.175	0.034	0.033	0.062	0.041	0.089
Silica removed per part MgO in reagent used............	..	0.180	0.209	0.205	0.176	0.115	...
Magnesium oxide (Remosil), ppm.....................	..	260	....	....	...	...	...
Sodium hydroxide, ppm.......	..	50	325	160	...	...	200
Magnesium sulphate ($MgSO_4 \cdot 7H_2O$), ppm..............	..	...	1320	1320	...	...	...
Dolomitic lime (35.5 per cent MgO), ppm................	..	...	....	....	725	725	...
Sodium carbonate, ppm.......	..	...	....	....	...	670	...
Ferric sulphate (Ferrisul), ppm	..	...	....	....	...	...	375

NOTE: Conditions: Tests 1 to 5 inclusive; 15-min stirring-and-retention time; temperature, 95 C. Test 6; 60-min stirring-and-retention time; temperature 25 C.

TABLE 5 SUMMARY OF FIELD TESTS ON SILICA REMOVAL

Date	Raw-water silica as SiO_2, ppm	Softener-effluent silica as SiO_2, ppm	Magnesium oxide (Remosil), ppm	Silica removed per part magnesium oxide
12/7/39	6.2	3.4	15	0.19
12/5/39 and 12/6/39	6.4	2.3	25	0.16
12/11/39 to 12/14/39	6.3	0.6	40	0.14
12/4/39	6.3	0.2	60	0.10

leasing an equivalent amount of objectionable sodium hydroxide. This results in high solids and alkalinity of the treated water.

Fig. 10 illustrates the fact that increase in the temperature at which silica removal takes place materially aids in the efficiency of silica removal by means of dolomitic lime in the same manner as increase in temperature aids removal of silica by magnesium oxide, as was shown in Fig. 1.

Table 4 illustrates the results obtained in the treatment of a raw water with the use of magnesium oxide, magnesium sulphate, dolomitic lime, and ferric sulphate. On a raw water of the type employed, it is evident from the analyses of the treated waters that removal of silica by magnesium sulphate, dolomitic lime, and ferric sulphate introduces exceedingly large solids content and the resulting characteristics of the treated water will present difficulties if used as boiler feedwater.

Field Tests on Silica Removal

The use of magnesium oxide for silica removal, in combination with hot-process lime-and-soda softening, has been proved under actual field conditions. The following data, secured from the first field tests conducted at the Lester, Pa., plant of the Westinghouse Electric and Manufacturing Company, are typical of those secured under plant conditions without recirculation of sludge.

Table 5 summarizes the results obtained in the removal of silica employing magnesium oxide. The values for the silica content of the raw and softener effluent are averages of tests made on samples composited from the softener effluent at 2-hr intervals and from the raw water at 4-hr intervals of the dates noted. It can be observed that the silica removed per part of magnesium oxide decreases with increased removal of silica from solution.

Fig. 11, based on the results given in Table 5, shows the plot of percentage of original silica removal against quantities of magnesium oxide employed. Like Fig. 4, which was based on laboratory data, the curve flattens with increased quantities of magnesium oxide as is characteristic of adsorption.

Fig. 12 illustrates a Freundlich adsorption isotherm plotted from the data of Table 5. It can be noted that the slope of the line is considerably less than was secured in the case of the laboratory data shown in Fig. 6. Although recirculation of sludge was not employed in these tests, the accumulation of sludge taking place in the sludge cone of the sedimentation tank will yield effects similar to partial recirculation. Experiments in laboratory beakers without recirculation result in the water coming in contact with only the quantity of magnesium oxide used in the treatment of the individual 1-liter sample.

Under field conditions, however, if the sedimentation tank is desludged only once in 8 hr and the softener is operating at its rated capacity, there is the possibility of the treated water coming into contact with 8 times the quantity of magnesium-oxide sludge as was used in the treatment of an individual quantity of the water. Although this sludge will not, of course, come into the desired intimate contact with the raw water, there will be some effect observed as if a very small amount of recirculation were applied. This effect, coupled with the fact that a lesser quantity of adsorbent is always required under field conditions than under laboratory conditions, results in decreasing the slope of the Freundlich isotherm.

Field Tests on Coagulation With Magnesium Oxide

It has been mentioned previously that, with the use of magnesium oxide for silica removal in hot-process lime-soda softeners,

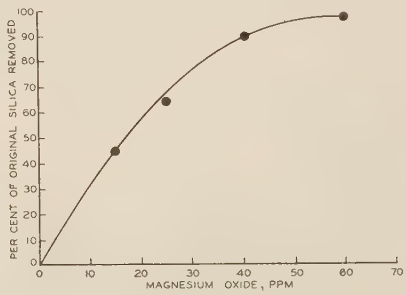

FIG. 11 FIELD TESTS ON ADSORPTION OF SILICA

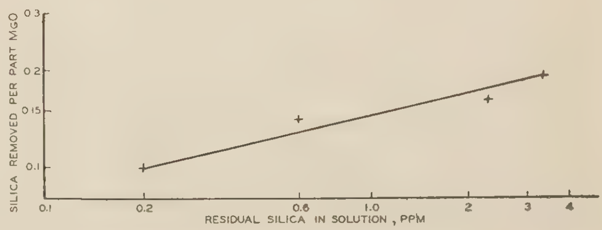

FIG. 12 FREUNDLICH ADSORPTION ISOTHERM BASED ON FIELD TESTS

excellent coagulation is secured and the need for additional coagulants is eliminated. Fig. 13 illustrates the turbidity determined on samples of effluent from the sedimentation tank and filter of the Cochrane hot-process unit at the Lester plant under comparable load conditions. Two series of tests were made using, in one case, 10 ppm magnesium oxide, and in the other case, 20 ppm

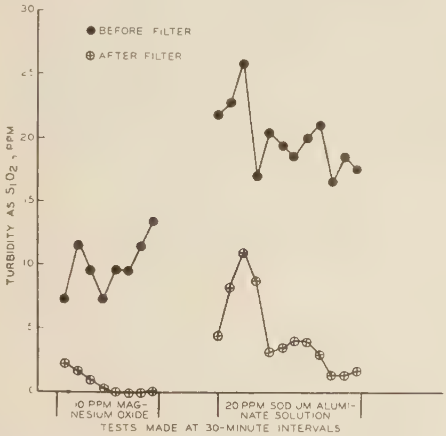

Fig. 13 Field Tests on Coagulation With Magnesium Oxide in a Hot-Process Lime-Soda Softener

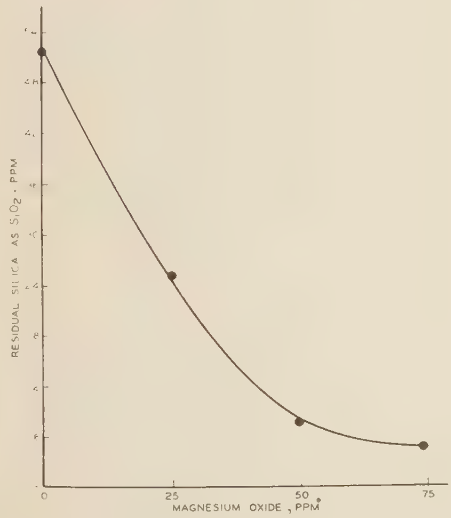

Fig. 14 Field Tests on Silica Removal With Sludge Recirculation

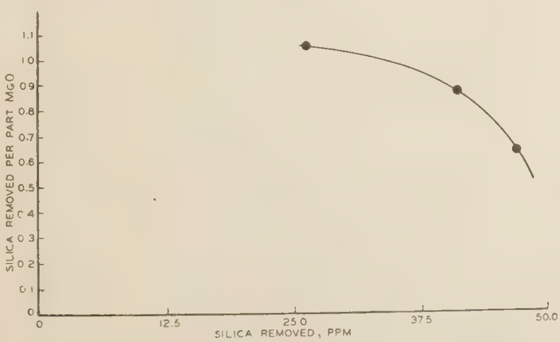

Fig. 15 Adsorption Efficiencies; Field Tests With Sludge Recirculation

of a sodium-aluminate solution of 21.6 per cent, Al_2O_3 content. Turbidities were determined with the use of a Hellige turbidimeter, immediately after collecting and cooling the samples.

Field Tests on Silica Removal With Recirculation

The tests to be cited are typical of the silica removal which has been accomplished with recirculation of the partially spent magnesium-oxide sludge. These results are particularly interesting because of the high silica content of the raw water of 51.5 ppm as SiO_2.

Fig. 14 illustrates the silica removal accomplished in a hot-process lime-soda softener with three different rates of feed of magnesium oxide at 25, 50, and 75 ppm. Reduction of silica to 4.5 ppm was obtained by the use of 75 ppm magnesium oxide. In these tests the magnesium oxide was simply added to the chemical mixing tank of the softener along with lime and soda ash. Recirculation of sludge was employed in the manner shown in Fig. 8.

Fig. 15 illustrates the typical reduction in adsorption efficiency (quantity of silica removed per part of magnesium oxide) with increased removal of silica from solution. The effect of sludge recirculation can be noted in the relatively high quantity of silica removed per part of magnesium oxide, corresponding during the first part of the curve to better than 1 part of silica removed per part of magnesium oxide employed.

Conclusion

The efficient removal of silica from solution by means of magnesium oxide has been shown by laboratory and field tests. Comparative tests have indicated the superiority of magnesium oxide over the other reagents employed, not only in the quantity of silica removed per part of reagent employed but also with respect to the solids content of the treated water. It should be borne in mind, however, that the laboratory results presented have been secured under certain definite conditions and usually at retention times less than 1 hr. Before the requirements of magnesium oxide necessary for silica removal for any certain water can be calculated, it is necessary to know the temperature of operation, retention time permitted, equipment available, whether or not softening is also to be effected in the same tank, and whether recirculation of the partially spent magnesium-oxide sludge is to be employed. In translating laboratory requirements into terms of practical operation, it should be remembered that a lesser quantity of magnesium oxide is required under field conditions than in the laboratory and also that recirculation of sludge provides an opportunity for securing as much as 60 per cent reduction in magnesium-oxide requirements.

Acknowledgment

The authors wish to acknowledge with appreciation the courtesy of the Westinghouse Electric and Manufacturing Company in cooperating in the extension of the laboratory data to embrace field conditions. The first field tests, described in this paper, were conducted at the Lester, Pa., plant of that company. Naturally, laboratory data are subject to confirmation or revision by plant tests on a large scale, and a fine spirit of cooperation was extended by the Westinghouse Electric and Manufacturing Company in collaborating in the first practical tests of this silica-removal process. We are particularly indebted to Mr. F. W. Simon, chief engineer of the Lester Plant, for his many valuable suggestions and aid in securing and interpreting these data.

BIBLIOGRAPHY

1 "Removal of Silica From Water by Cold Process," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1320–1323.

2 "Investigation on the Removal of Silica From Water," by R. Stumper, *Die Wärme*, vol. 55, 1932, pp. 272–278.

3 "Complex Silicate Scales in High-Pressure Boilers," by S. T. Powell, L. V. Carpenter, and J. J. Coates, *Combustion*, vol. 11, no. 2, August, 1939, pp. 18–24.

4 "The Removal of Silica From Water for Boiler Feed Purposes," by M. C. Schwartz, *Journal*, American Water Works Association, vol. 30, 1938, pp. 659–678.

5 "Removal of Silica From Water by Hot Process," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 1323–1329.

6 "Adsorption of Soluble Silica From Water," by L. D. Betz, C. A. Noll, and J. J. Maguire, *Industrial and Engineering Chemistry*, vol. 33, 1941, pp. 814–820.

7 "Determination of Soluble Silica in Water. A Photoelectric Method," by H. L. Kahler, presented at the 100th Meeting of the American Chemical Society, September 11, 1940.

8 "Standard Methods for the Examination of Water and Sewage," eighth edition, American Public Health Association, New York, N. Y., 1936, p. 77.

9 "Outlines of Theoretical Chemistry," by F. H. Getman and F. Daniels, fifth edition, John Wiley & Sons, Inc., New York, N. Y., 1931, p. 238.

10 "Colloid Chemistry," by Jerome Alexander, section by H. Freundlich, Chemical Catalog Company, New York, N. Y., vol. 1, 1926, pp. 575–599.

11 "Adsorption as a Means of Separation," by J. W. Hassler, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 640–644.

12 "Removal of Silica From Water," by A. S. Behrman and H. Gustafson, *Industrial and Engineering Chemistry*, vol. 32, 1940, pp. 468–472.

Silica in High-Pressure-Boiler Water

By HAROLD FARMER,[1] PHILADELPHIA, PA.

The advent of high steam pressure has resulted in many boiler-tube failures in which silica appears to have played a very significant part. Attention is called in this paper to the fact that silica trouble appears to be more prevalent in high-pressure boilers having high rates of heat input. Trouble has occurred irrespective of high or low silica content in the boiler water, although silica scale does not form in all boilers irrespective of the amount present. The author points out the need for more basic knowledge regarding the behavior of silica in boiler water at high pressures and temperatures, as well as the development of satisfactory methods for determining the form in which silica exists in the boiler water.

DURING the last two years, it has become increasingly evident that recognition is being given to the fact that the presence of silica is a factor of major importance in controlling satisfactorily the water conditions in high-pressure boilers. The author has had some experience with this problem and has also had an opportunity to observe the results obtained by other investigators.

Generally speaking, the presence of silica in boiler water at steam pressures below 600 psi has not given serious trouble or else it could be satisfactorily controlled.

The advent of steam pressures of the order of 1200 psi has resulted in many tube failures in which silica appears to have played a very significant part. In some instances, failures have occurred when the silica content of the boiler water has been lower than 20 ppm. This has occurred even when other water conditions with respect to alkalinity, pH, and the freedom from scale-forming salts, such as calcium and magnesium, were considered ideal.

Where Silica Trouble Is Found

It has been observed, in high-pressure boilers of low heat input, that the trouble from silica deposits has not been especially serious. Most of the silica trouble has occurred in high-pressure boilers having high rates of heat input. However, this does not imply that all high-pressure boilers, having high rates of heat input, will experience difficulty from silica. It may be, however, that the high rate of heat input at high rates of evaporation will upset the proper circulation and result in starvation and dry areas, under which condition, soluble salts including silica may deposit on the dry-area surfaces. When these surfaces are again wetted the soluble salts dissolve again but, due to the increased tube temperature during the starvation period, the silica present may undergo a change in structure and, in this form, may not now be soluble. Repeated cycles of starvation will ultimately build up a deposit of silica which will cause overheating of the tube with ultimate failure. Some of these failures have manifested themselves by characteristic blistering.

In contrast to this type of failure, some do not have the blister, the failure being characterized by a small hole in the tube, which, when examined from the inside, gives the appearance of corrosion and overheated metal. Study of the tube metal on the inner surfaces shows that the portion which failed is often surrounded by a hard silica scale. This type of failure has been attributed by some investigators to the same cause as that mentioned, i.e., starvation with formation of silica deposits. The ultimate failure, however, is attributed to some condition which might cause a small piece of the scale to break away or flake off from the tube surface, resulting in this small area (relieved of material which tended to reduce the heat transfer) trying to absorb too much heat, causing dissociation of water and consequent failure of the tube.

The theory advanced to explain this type of failure, if correct, is of more than passing interest, because a similar condition could exist on the furnace side of the tube as on the water side. Imagine a powdered-fuel furnace with a tube completely slagged when, suddenly, a small piece of the slag falls away leaving a clean surface; this would be similar in all respects to the breaking away of scale on the inside of the tube. While we have no proof to support this theory, we have evidence of tube failures in the furnace when there is no sign of scale on the inside of the tube.

There is no doubt but that the silica problem is serious. As in the case of many other problems of the past, we must go through a period of failures in order to gather sufficient data, first to develop theories and then to substantiate them until they are generally accepted. Ultimately, evidence must be developed to show that they follow definite laws. However, the number of data now available might be correlated by some research group with a thorough knowledge of the problem, coupled with operating experience, at least to establish an intelligent starting point for attacking this problem.

Silica Should Be at a Minimum in Feedwater

Information now available definitely indicates that silica in any form should be maintained at a minimum value in the boiler water. Even with good condensate feedwater, assuming the total dissolved solids to be 1 ppm, the water may contain 0.1 ppm of silica. This seems like a very insignificant amount but, when we consider that many of the present-day high-pressure boilers evaporate their own volume of water 5 times in 1 hr, this insignificant figure represents 12 ppm of silica in the boiler in 24 hr. Many operators set this figure as the limit and resort to blowing down the boilers when this value is approached. If for any reason a slight increase in condenser leakage takes place, the silica content of the water is yet further increased, necessitating more frequent blowing down of the boiler. In many high-pressure plants, it has been necessary to maintain a 24-hr constant supervision of the chemical control in order to assure satisfactory water conditions in the boilers.

Plants which operate with a large amount of make-up water, which is not evaporated make-up, must resort to external chemical treatment for removal of silica. The present effectiveness of such treatment appears to have a limitation of about 2 ppm of silica residual remaining in the water. Waters of this character may necessitate continuous blowing down of the boilers in order to maintain silica not in excess of 12 ppm. Frequent blowing down of high-pressure boilers is not looked upon with much favor by the operators.

Methods of Preventing Silica Scale

The use of internal chemical treatment for the prevention of

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Contributed by the Joint Research Committee on Boiler Feedwater Studies, and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

silica scale in high-pressure boilers has in some instances been given consideration but, as yet, this treatment has not proved very satisfactory. Some instances have been reported where the cure was worse than the disease. One reason why some internal treatments may not have been satisfactory is because of the lack of the extremely careful supervision and control of the variables present in the water at all times, which is vitally necessary to uccess.

Many who have made chemical analyses of boiler deposits and scale know that such analyses have limitations but that we can rely on them to determine the quantities of certain elements present. However, when we wish to know just how these elements exist in combination with each other, the information is lacking or at the best it is only hypothetical. A knowledge of the combination and form in which the elements exist is definitely of more importance because it throws light on the physical conditions which played a part in the production of the scale. For example, nearly every chemical analysis of boiler scale shows or reports silica as silicon dioxide (SiO_2). In reality it is total silica reported as SiO_2. When it is realized that this reported silica probably could have existed in five different forms, all having different physical properties, we begin to realize that the quantitative determination of the silica present is not the complete story.

Quite recently, several investigators have employed the X-ray diffraction method of analysis for boiler scale and deposits. This method has made it possible to identify compounds of the elements which, from the knowledge of the physical properties of the compounds, has enabled the investigator to indicate, for example, the temperature at which certain compounds would or could have been formed. Information of this character correlated with tube-temperature measurements may furnish reliable data with respect to poor circulation or to the possible formation of scale which may be troublesome. The author takes this opportunity to state that the absence of scale, as shown by boring, is not reliable since many of these silica scales are so hard and tough that some of the best cutters ride over the surface of the scale.

Turbine-Blade Deposits

While this paper is supposed to be confined to silica in high-pressure-boiler waters, it would be an omission if some reference were not made to the aftereffect of silica, namely, the turbine-blade deposits. If there were no carry-over with steam, we could forget about the turbine, but most boilers have some carry-over. Examination of any turbine-blade-deposit analysis always shows the presence of some silica and, in most cases, entirely out of proportion to the silica and other constituents of the boiler water. This of course can be explained but it might appear, at first glance, that silica carry-over was selective. While this is not generally accepted, it does appear that the form in which silica exists in the boiler water may have a bearing on the nature of the silica deposits formed on the turbine blades. For example, in some instances, we find the deposit to be entirely silica (SiO_2), deposited like an enamel, not water-soluble, and hard to remove. In other instances, we find the deposit to be in various stages of dehydrated sodium silicate, some of it is water-soluble and some of it is not, depending upon the degree of dehydration. The temperature of the superheated steam of course plays a large part in the form in which the silica may exist on the blades, but there is also some evidence that the form in which the silica exists in the boiler water is related to the form of silica found on turbine blades.

Conclusion

In conclusion the writer wishes to point out certain significant observations:

1 The presence of silica in high-pressure-boiler waters has been blamed for certain types of tube failures.

2 The experience of many plant operators indicates the necessity of maintaining the silica content of the boiler water at a minimum (not in excess of 15 ppm).

3 When poor circulation exists, under certain conditions, the presence of small quantities of silica in the boiler water may cause serious trouble by deposition of silica scale which reduces the heat transfer and results in overheating the tube.

4 There is evidence to indicate that boilers having a high rate of heat input have experienced more trouble from silica scale than boilers having a low rate of heat input.

5 Silica scale does not form in all boilers irrespective of the amount present. The condition in which the silica exists in the boiler water is an important factor and should be evaluated, together with the operating conditions of the boiler.

6 There is a lack of definite information regarding the behavior of silica and its compounds in boiler water at high pressures and temperatures.

7 Some operators have blamed poor boiler circulation for the deposition of silica scale which has been the direct cause of tube failure. It is possible that the boiler designer can make further improvements in the circulation of high-pressure boilers, but this does not relieve the boiler-water chemist from the obligation to find ways and means for controlling satisfactorily the silica in high-pressure-boiler waters.

8 The control of water conditions in high-pressure boilers requires a careful and continuous supervision in order to insure satisfactory conditions in the boiler at all times.

9 It would seem that the Joint Research Committee on Boiler Feedwater Studies is a logical body to sponsor an investigation of the silica problem in high-pressure-boiler waters.

Discussion

E. P. Partridge[2] and R. E. Hall.[3] The author has presented a general indictment of silica; let us consider briefly the individual charges which may be drawn against it:

Silica Accelerates Embrittlement. Resolving some earlier discrepancies between their experimental results, Schroeder and Berk,[4] and Straub and Bradbury[5] each found 5 years ago that as little as 3 or 4 parts of silica for each 1000 parts of caustic soda greatly stimulated intergranular attack by concentrated solutions of the latter on stressed steel. A normal boiler water, containing 100 ppm of NaOH might accordingly, when concentrated in a riveted seam, be fully capable of producing embrittlement even if it contained less than 0.5 ppm of silica.

Silica Causes Deposits in Boilers. As noted in the paper, silica may appear in various forms in boiler deposits. Studies with the polarizing microscope and with X-ray diffraction have revealed a variety of compounds definitely present in various boilers.

Calcium silicate, like calcium sulphate, forms a hard dense scale on heat-transfer surfaces. Clark and Bunn[6] report that four hard calcium-silicate scales consisted of zonotlite, $5CaO \cdot 5SiO_2 \cdot H_2O$. In one soft scale, they found a sodium-calcium silicate, corresponding to the mineral pectolite, $Na_2O \cdot 4CaO \cdot 6SiO_2 \cdot$

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[2] Hall Laboratories, Inc., Pittsburgh, Pa.

[3] Director, Hall Laboratories, Inc., Pittsburgh, Pa. Mem. A.S.M.E.

[4] "Action of Solutions of Sodium Silicate and Sodium Hydroxide at 250°C on Steel Under Stress," by W. C. Schroeder and A. A. Berk, Trans. American Institute of Mining and Metallurgical Engineers, vol. 120, 1936, pp. 387–400; *Combustion*, vol. 7, no. 8, 1936, pp. 29–33.

[5] "New Laboratory Data Relative to Embrittlement," by F. G. Straub and T. A. Bradbury, *Power Plant Engineering*, vol. 40, 1936, pp. 104–105.

[6] "The Scaling of Boilers. Part IV, Identification of Phases in Calcium Silicate Scales," by L. M. Clark and C. W. Bunn, *Journal of the Society of Chemical Industry*, vol. 59, 1940, pp. 155–158.

H_2O. In our own work,[7] we have observed, in addition to zonotlite, a calcium silicate with refractive indexes corresponding to hillebrandite, $2CaO \cdot SiO_2 \cdot H_2O$.

Sodium-aluminum silicate in the form of analcite $Na_2O \cdot Al_2O_3 \cdot 4SiO_2 \cdot 2H_2O$, was first reported as a boiler scale by Powell.[8] A similar complex silicate with a lower ratio of SiO_2 to Na_2O and Al_2O_3 has also been identified more recently in the course of our own work. Scales comprising these constituents develop on those heat-transfer surfaces where, because of either an extremely high rate of heat input or limited circulation, localized evaporation to an excessive degree takes place.

Another product of this same type of action is the sodium-iron silicate, acmite, $Na_2O \cdot Fe_2O_3 \cdot 4SiO_2$, found occasionally in overheated slag-screen or waterwall tubes, or along the ceiling of top-row tubes in straight-tube cross-drum boilers.

Silica itself, in the crystalline form of quartz, occasionally appears as a major constituent of a scale along with analcite.

All of the substances noted, other than the rarely occurring pectolite, generally form compact scales so hard that mechanical removal is difficult. In contrast, magnesium silicate typically occurs in a boiler as a soft, sludgy deposit. The individual particles appear amorphous, like the mineral meerschaum; the X-ray diffraction pattern corresponds to chrysotile, $3MgO \cdot 2SiO_2 \cdot 2H_2O$.

Silica Forms Deposits in Turbines. Carry-over to turbines has produced not only deposits of readily water-soluble salts, among which sodium silicate may be a prominent constituent, but also insoluble hard deposits of silica which can be removed only with difficulty. The constitution of these deposits of silica changes from the crystal form of quartz in the higher temperature stages to amorphous silica in the subsequent stages.

These charges preferred against silica are admittedly grave, particularly at a time such as the present, when an unscheduled outage of a boiler or turbine may be a real calamity, and scheduled outages are to be postponed as long as possible.

As a saboteur of power production, silica now gains the attention held by calcium sulphate a quarter of a century ago during the first World War. Since that time, controlled conditioning with phosphate has liquidated scales of calcium sulphate and all other calcium compounds, including calcium silicate. What measures can be taken against silica in its other aspects?

To exclude silica from the boiler water would require the evaporation of make-up, with no carry-over from the evaporators and no contamination of condensate returns by condenser leakage, storage in concrete tanks, or otherwise. Even under the best practical conditions, it would be difficult to prevent the concentration of silica in the boiler water from reaching the fractional part per million capable of accelerating embrittlement. Protection against this type of metal cracking must be sought by other means: (1) From the mechanical viewpoint, by the use of one-piece drums; (2) from the chemical viewpoint, by testing the response of each boiler water to the various inhibitors which have been suggested. In this connection, our thanks are due Straub and Bradbury[9] on the one hand, and Schroeder, Berk, and O'Brien,[10] on the other, for the development of testing equipment which is putting an end to the "Dark Ages" of blind faith, or heretical lack of faith, in the sulphate-alkalinity ratios. Today, instead of arguing as advocates, we can test as engineers, then modify the conditioning of the boiler water and test again, until the tendency of the water to cause intergranular cracking has been reduced or eliminated.

Even though extreme efforts to exclude silica from boiler waters would have no significance from the standpoint of embrittlement, to the casual observer a case for partial removal of silica might be made out with respect to boiler scales and turbine deposits. Some clear and hardheaded thinking is, however, needed here. It has been pointed out previously that the silica-containing substances which actually form scales in boilers are predominantly calcium silicate, on the one hand, and sodium-aluminum silicate, on the other. That calcium-silicate scale, like calcium-sulphate scale, is prevented from forming by the maintenance in the boiler water of a slight excess of phosphate ion, has been abundantly proved in plant after plant. Scales of sodium-aluminum silicate and of sodium-iron silicate in the boiler and deposits of silica in the turbine remain as the troubles which might drive an operator to consider reduction of the silica content of his boiler feedwater.

Along with the possibility of silica removal in primary treatment, the operator will want to keep in mind the concept that the complex silicate scales are products of mechanical as well as chemical factors, developing only where the normal silica content of the boiler water is multiplied many times by localized evaporation to an exceptionally high degree. In many cases, indeed, excessive evaporation is indicated by the presence of relatively soluble salts, such as sodium sulphate and sodium phosphate, trapped in the deposits of sodium-aluminum silicate. While the solubility studies of Straub[11] indicate that this scale might be produced at a concentration of 120 ppm of SiO_2, it must not be forgotten that his solutions at the same time contained much more alumina than would be present in any boiler water derived from properly treated feedwater. With the low concentrations of alumina in the boiler water corresponding to good practice, much more than 120 ppm of silica would probably be required to start the deposition of sodium-aluminum-silicate scale.

But why belabor the point that most boiler waters can deposit sodium-aluminum-silicate scale only when concentrated locally in regions of very high heat input or limited circulation? The operator confronted with the current necessity of getting more and more steam from his existing equipment may well ask, "so what?" He cannot wait for new boilers to share the load or for his old boilers to be remodeled. While these solutions might ultimately be less expensive, he may be forced to choose between increase in blowdown or reduction of silica in feedwater to minimize his difficulties.

That he will only minimize rather than eliminate trouble is a pessimistic prophecy, but one that the writers feel is fully justified. Sodium-aluminum silicate or sodium-iron silicate is essentially a symptom of a condition. Chemical treatment directed at the symptom will not make the "hot spots" in a boiler any less hot or move more water where it is needed to prevent failure of tubes due to continuous steam blanketing or to repeated overheating and quenching.[12] On the other hand, proper mechanical treatment of the fundamental source of trouble will eliminate the symptom.

Silica deposits on turbine blades seem a strong argument for

[7] "Some Applications of the Polarizing Microscope to Water-Conditioning Problems," by E. P. Partridge, Proceedings American Society for Testing Materials, vol. 37, part 2, 1937, pp. 600–608.

[8] "A Critical Study of Boiler Scales and Advanced Methods of Analysis and Identification," by S. T. Powell, *Combustion*, vol. 5, no. 3, 1933, pp. 15–19.

[9] "Boiler Water Treatment. New Methods for Preventing Embrittlement," by F. G. Straub and T. A. Bradbury, *Mechanical Engineering*, vol. 60, 1938, pp. 371–376.

[10] "Intercrystalline Cracks in Locomotive Boilers," by W. C. Schroeder, A. A., Berk, and R. A. O'Brien, Association of American Railroads, Circular DV-989, 1940; "Embrittlement Detector," *Combustion*, vol. 12, no. 2, 1940, pp. 19–21.

[11] "Analcite: Preparation and Solubility Between 182° and 282° C," by F. G. Straub, *Industrial and Engineering Chemistry*, vol. 28, 1936, pp. 113–114; correction, p. 361.

[12] "Attack on Steel in High-Capacity Boilers as a Result of Overheating Due to Steam Blanketing," by E. P. Partridge and R. E. Hall. Trans. A.S.M.E., vol. 61, 1939, pp. 597–622; discussion, vol. 62, 1940, pp. 711–717.

keeping silica in the boiler water at a low value. Whether silica is carried over physically to the turbine in droplets of boiler water, suspended in the steam and concentrated to a high degree on passing through the superheater, or whether it is transported in the vapor phase, as suggested by the experiments of Grieg,[13] decreasing the concentration of silica in the boiler water should decrease the rate at which deposits accumulate on the turbine blades. Other approaches to the problem may be the reduction of carbon dioxide in the steam, or even the apparently crazy expedient of allowing controlled carry-over. Although the evidence is by no means complete, experience in a number of plants suggests that hard deposits of insoluble silica develop when the steam condenses to give an acid solution, as would be the case when the concentration of carbon dioxide was large relative to the carry-over of alkali from thé boiler water. When this relation is reversed, any deposits formed are water-soluble and can be removed readily by systematic washing at prearranged intervals. While this may seem like compromise with, rather than conquest of a difficulty, the long-range economic balance sometimes favors compromise.

Processes and equipment for the removal of silica from boiler feedwater have the center of the stage today. Their relative merits should be, and undoubtedly will be, debated widely during the next few years. Lest this debate lead to uncritical acceptance of silica removal for its own sake, the specific objective in each individual case should be clearly defined. Is the problem calcium-silicate scale, or is it silica on the turbine blades? How much will it cost to get the silica in the feedwater down to 5 ppm or 2 ppm, in comparison with the cost of increased blowdown, and how much will maintenance costs and outage be reduced? These are some of the questions which must be asked and answered. In the meantime, we need more knowledge all along the line, knowledge which might well be secured by a cooperative effort through the Joint Research Committee on Boiler Feedwater Studies, as suggested by the author.

[13] "Notes on the Volatile Transport of Silica," by J. W. Greig, H. E. Merwin, and E. S. Shepherd, *American Journal of Science*, vol. 25, 1933, pp. 61–73.

The Flexible-Sleeve Multiple-Oil-Film Radial Bearing

By GUSTAVE FAST,[1] ANNAPOLIS, MD.

Prof. Osborne Reynolds' theoretical analysis of Beauchamp Tower's researches on journal bearings proved the necessity for a convergent wedge-shaped oil film between journal and bearing surfaces in order to create useful load-carrying pressures in the oil film. Michell and Kingsbury working independently applied Reynolds' theory to the development, notably of thrust bearings, and incidentally of radial bearings, with surfaces divided into tilting pads which created a multiplicity of convergent oil films instead of the single convergent film naturally present in an ordinary plain journal bearing. Later Wallgren brought out another form of pivoted-pad radial bearing, in which the pads rotate with the shaft. The multiple-oil-film radial bearing discussed in this paper accomplishes the foregoing without recourse to the use of pivoted shoes or pads. The flexing of the continuous block-sleeve-bearing member under load provides a multiplicity of wedge-shaped oil-film boundaries which create shearing stresses in the circulating oil and produce useful load-carrying pressures in the oil films. Careful tests have been conducted on the multiple-oil-film radial bearing under different conditions of speed, load, and oil viscosity, in order to determine its operating characteristics. The paper summarizes and discusses these test data.

A BRIEF analysis of the history of bearings would seem to be a logical introduction to an exposition of the multiple-oil-film radial bearing, which is the most recent bearing development. Osborne Reynolds' theoretical analysis in 1886 of Beauchamp Tower's researches on journal bearings (1)[2] proved beyond doubt that a convergent wedge-shaped oil film between the journal and the bearing surfaces was necessary to create shearing stresses in the viscous lubricant which result in useful load-carrying pressures in the oil film.

Any full journal bearing must obviously have some clearance and the resultant eccentric displacement of the journal under load automatically creates convergent oil-film boundaries (4, 10, 11). Thrust bearings, however, present a much more difficult problem. They transmit their load between two flat parallel rigid members, and much grief is experienced with such bearings because of the impossible task of establishing wedge-shaped oil films between their coacting surfaces.

About 20 years elapsed after Reynolds advanced his classical hydrodynamic theory of lubrication before practical application of these principles was attempted.

Michell (6) and Kingsbury both perceived, independently of each other, the inherent differences in geometrical relationship between the coacting surfaces of journal and thrust bearings, and devoted their respective efforts toward a rational solution by applying Reynolds' theory to thrust bearings. By dividing the bearing surface into tilting pads, they succeeded in establishing multiple convergent wedge-shaped oil films between thrust-bearing surfaces, in contrast to the single convergent film naturally present in the plain journal bearing with clearance.

Early journal-bearing designs were noted for their great length-to-diameter or L/D ratio which was sometimes as high as 6 or 8. They were little affected by side leakage but suffered much from shaft deflection, which necessitated a low unit pressure with consequent great film thickness to accommodate the bent journal. As time went on, the bearings were shortened until today the conventional design shows an L/D ratio ranging from 1 to 1.5 with no serious increase in side leakage resulting from the reduction. The earlier long bearings were limited to very low unit bearing pressures of the order of 25 to 50 psi, whereas, the present-day short bearings usually sustain pressures of 100 to 200 psi.

Problem of Shaft Deflection

Shaft deflection is unavoidable and becomes a serious matter in heavily loaded bearings, due to concentration of the load at the bearing edges. Self-alignment of the bearings, although highly desirable if not absolutely necessary, does not entirely compensate for shaft deflection, as the shaft is actually bent to a definite radius of curvature, depending upon the bending moment, the geometrical form, and the elastic properties of the shaft.

There are really only two ways in which to limit the harmful effect of shaft deflection

1 By making the shaft heavier and consequently stiffer.

2 By making the bearing shorter.

The first method calls for a more expensive shaft, besides greater frictional losses in the bearing. The second method has its limitation in the ordinary journal bearing. With an L/D ratio much below 1, there is serious side leakage and excessive friction coupled with low carrying capacity. This has been thoroughly demonstrated by McKee and McKee (3).

The object of making a very short bearing is therefore not so much the saving in space as the limiting of shaft deflection within the compass of the bearing. When we consider that most bearings operate with a film thickness of the order of 0.001 in. or less the importance of keeping the journal deflections to a still lesser order of magnitude will be readily appreciated. By minimizing the effect of shaft deflection through the use of a very narrow bearing, considerably higher unit pressures can be used with a corresponding reduction in frictional losses if the side leakage is no greater than in normally proportioned bearings.

It was with this criterion in mind that the development of the multiple-oil-film radial bearing was undertaken.

Details of Multiple-Oil-Film Radial Bearing

Several types have been developed and thoroughly tested and a great deal of useful data have been collected. The final form which is now in commercial production is shown in Figs. 1 and 2. The bearing normally consists of but three parts as a complete unit, but may also be furnished in only two parts for bearing directly on the journal portion of a shaft.

A is a hardened, ground, and lapped journal hub, mounted directly on the shaft with a shrink fit. B is a flexible continuous

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[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

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block sleeve of polygonal shape with spherical abutments mounted in C, a keeper sleeve having an internal spherical surface to provide self-alignment for sleeve B. The bearing sleeve B is made of steel and lined with a plastic metal applied by the centrifugal casting method. The nature of this plastic metal can be varied to suit any special application, but a high-tin-base babbitt is generally employed. The bearing surface is finished by a diamond tool in a precision boring machine. Transverse oil grooves d divide the bearing surface into a number of blocks, deflected by the oil-film pressure to form wedge-shaped films. The grooves d communicate at each side of the bearing with circumferential oil-distribution grooves e. The oil supply is drawn up through holes f in sleeve B by the viscous pump action of the rotating journal hub. An abundance of oil is thus circulated, assuring an ample supply for any speed requirement without excessive splashing or churning of the lubricant. Because of this lubrication arrangement, a radial load may be taken in any direction with equal efficiency.

The L/D ratio of these bearings may be taken as 0.25 for an approximate value. Due to the shortness of the bearing, there is very little deflection across the journal surface. Consequently, high unit pressures may be carried with safety.

In so far as affecting side leakage, friction, or load capacity, the L/D ratio is not the decisive factor when dealing with multiple-oil-film radial bearings. In these bearings, each block should be considered as an individual bearing whose circumferential length-to-width ratio should have a suitable value for the control of side leakage.

It is obvious, from the nature of bearing sleeve B, that each loaded block cannot have a continuous convergent film for its entire length, although such a condition would be highly desirable from a theoretical point of view. Instead, the loaded blocks have a convergent and a divergent portion of film space, each portion being approximately one half of the geometrical length of the block.

The effective length of the wedge-shaped oil film is approximately 65 per cent of the geometrical length of the block.

If l = effective film length
w = block width
then l/w = 0.6 to 1 (according to the design)

As has been clearly demonstrated by Needs (5), the smaller this ratio is made the less is the effect of side leakage.

In order to utilize the entire geometrical length of each block for a pressure producing convergent film, it is necessary to use loose blocks pivotally mounted. This leads to a complicated design that is hardly practical from a commercial point of view. The number of blocks would have to be greater than in the form of bearing shown, in order to maintain a desirable l/w ratio. One particularly serious problem incident to the use of loose blocks is that of providing a supporting pivot or fulcrum which will not break down under the enormous loads that the oil film itself is able to sustain. The real limitation to the load-carrying capacity of all bearings of this type is not the carrying capacity of the oil film, but the pivotal support of the block members as pointed out by Michell (13, 17).

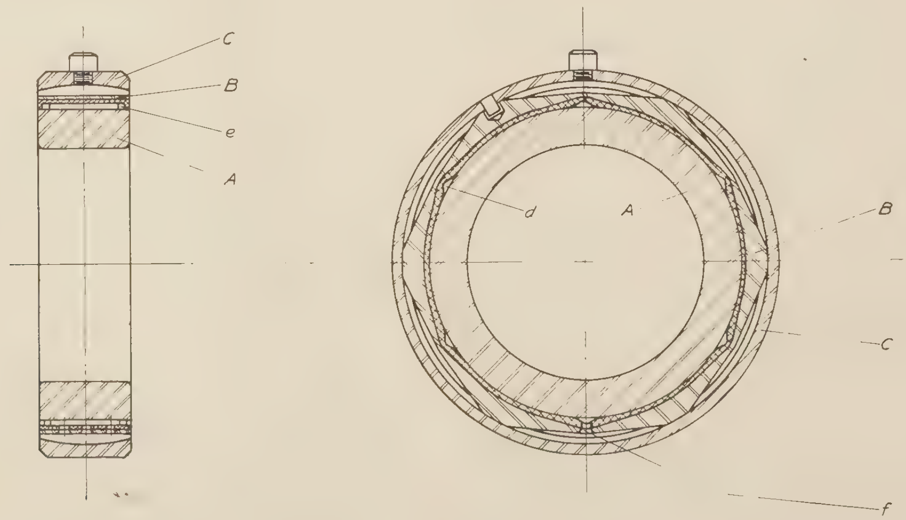
Tests Conducted to Determine Performance Laws

In order to determine the performance laws of the multiple-oil-film radial bearing, a series of tests under varying conditions of speed, load, and oil viscosity was conducted, and the results so obtained are herein reported and briefly discussed.

These tests were carried out on a ball-bearing testing machine, using four multiple-oil-film bearings, shown in Figs. 1 and 2. The arrangement of the four bearings in this machine is shown in Fig. 3.

The two outer bearings were loaded upward and the two inner ones were loaded downward. A small quantity of oil was circulated at the higher speeds for cooling purposes, but the supply was erratic and therefore the oil temperature varied greatly from time to time. Five speeds were used during this test, namely, 900, 1200, 1800, 3000, and 3600 rpm.

The friction torque was measured by the usual dynamometer-type electric motor, equipped with a scale beam and weights. The duration of the test for each speed was 31 days continuous run (744 hrs). The total length of time for the entire series of tests therefore was 3720 hr. The time allowed between each increment in load until the maximum for a given speed was reached was about 4 hr. The load was then maintained at this maximum for the remainder of the test. The maximum load for each speed was fixed arbitrarily at a value much in excess of that which might be expected in actual normal practice for that speed. Preceding the test, all the bearing parts were measured by a Zeiss optical micrometer to 5 decimal places of an inch in an air-conditioned room at constant temperature. After the completion of the test the same procedure was duplicated.



Figs. 1 and 2 Commercial Form of Flexible-Sleeve Multiple-Oil-Film Bearing

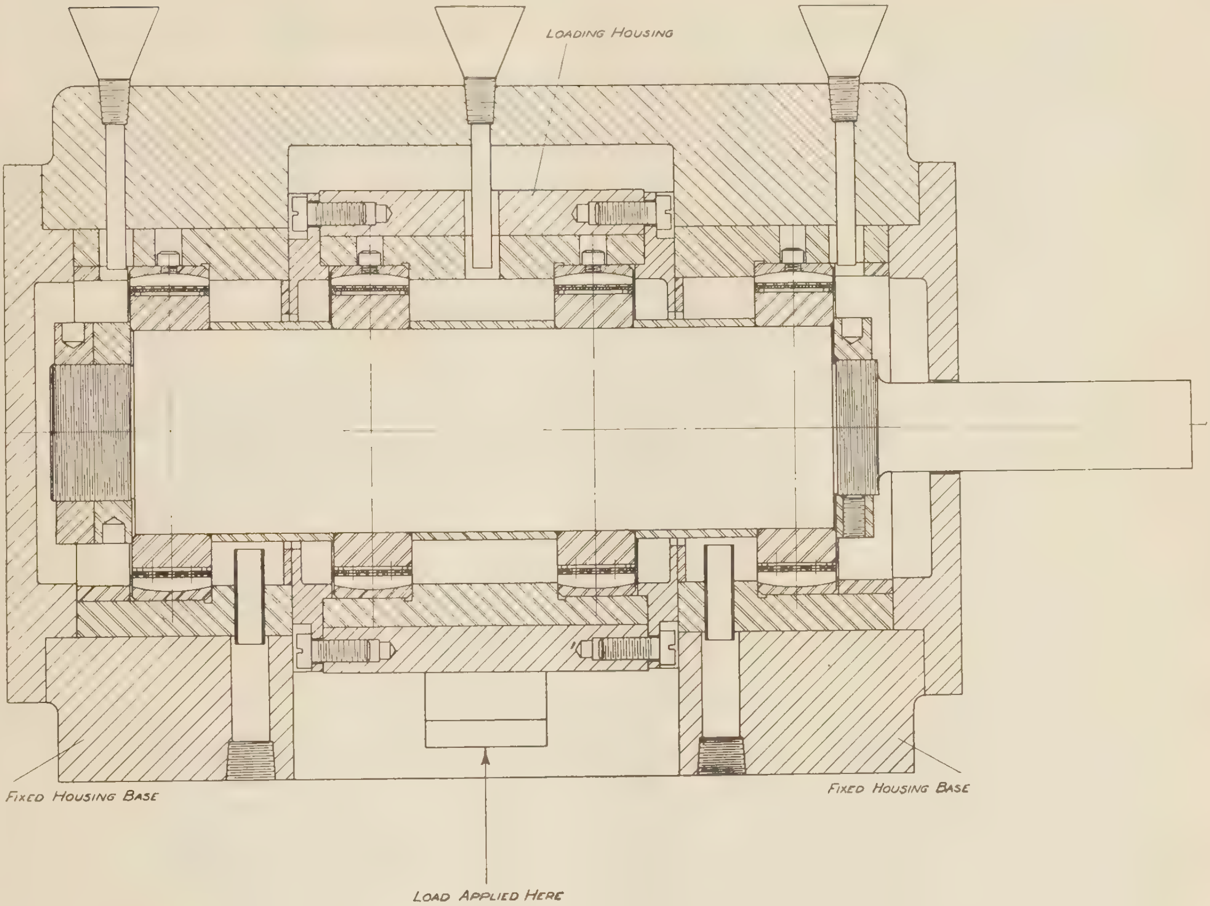


FIG. 3 ARRANGEMENT OF FOUR BEARINGS IN BALL-BEARING TESTING MACHINE

The result of this search for signs of wear disclosed the fact that no wear was detectable within the limit of sensitiveness of the micrometer which was 0.00001 in.

Two grades of straight mineral oils were used, namely, No. 2110 for the speeds of 900, 1200, and 1800 rpm; No. 2075 for the speeds of 3000 and 3600 rpm.

The No. 2110 oil had the following characteristics

Viscosity S.U. at 100 F = 180 Sec
Viscosity S.U. at 130 F = 97 Sec
Viscosity S.U. at 210 F = 45 Sec
Specific gravity = 0.87

The No. 2075 oil had the following characteristics

Viscosity S.U. at 100 F = 150 Sec
Viscosity S.U. at 130 F = 84 Sec
Viscosity S.U. at 210 F = 42 Sec
Specific gravity = 0.899

The important bearing dimensions pertinent to this investigation were as follows

Diameter of bearing = 4.28755 in.
Diameter of journal = 4.27960 in.
Bearing clearance = 0.00795 in.
Clearance ratio = 0.00185 in.
Length of bearing = 1.2 in.

Tables 1 to 5 give the log of the various tests.

The following nomenclature is used in the ensuing discussion of test data

NOMENCLATURE

L = length of bearing along axis of rotation, in.
D = journal diameter, in.
D' = bearing bore, in.
C = bearing clearance = $(D' - D)$
l = effective oil-film length in direction of motion
w = width of block = L
P = unit bearing pressure on projected area $(D \times L)$, psi
μ = coefficient of friction
N = speed, rpm
Z = absolute viscosity, centipoises
n = number of blocks

All of the test results of $\mu \sim P$ at various speeds were plotted, as shown in Fig. 4.

Next all the values of μ were plotted as ordinates against abscissas representing ZN/P as shown in Fig. 5.

The curve shown in Fig. 4 represents a graph of the equation

$$\mu = \frac{0.191}{\sqrt[3]{P^2}} \qquad [1]$$

which represents fairly well the average empirical relationship between μ and P without taking viscosity and speed into consideration. This is not incompatible with theory, as Z decreases with an increase in N, making the product of $(Z \times N)$ almost a con-

TABLES 1 TO 5 LOG OF TESTS ON MULTIPLE-OIL-FILM RADIAL BEARINGS

(The values given for total load and friction torque apply to one test bearing only having a diameter of 4.2796 in. and a length of 1.2 in.)

Table No. 1 – 900 R.P.M.

Total Load in Pounds W	Load per Square Inch of Projected Area P	Friction Torque in Inch-Pounds	Actual Coefficient of Friction	Oil Film Temp. Rise above Ambient – °F	Mean Oil Film Temperature °F T	Absolute Viscosity in Centipoises at Temperature T	$\frac{ZN}{P}$
1000	194.6	11.46	.00535	32	127	18.27	84.5
2000	389.2	16.00	.003737	41	130	17.4	40.2
3000	583.8	17.37	.002705	47	141	13.92	21.5
4000	778.4	18.90	.002208	50	147	12.62	14.6
5000	973.0	20.33	.001900	55	151	11.75	10.87
6000	1167.6	21.77	.001697	69	155	10.96	8.45
7000	1362.2	23.80	.001588	68	154	11.05	7.3
8000	1556.8	25.00	.001460	69	162	9.74	5.63
9000	1751.4	27.00	.001392	74	167	9.05	4.65
Table No. 2 – 1200 R.P.M.							
1000	194.6	11.67	.00545	45	141	13.92	85.8
2000	389.2	15.75	.00368	59	145	13.05	40.3
3000	583.8	17.25	.002686	66	156	10.70	22
4000	778.4	18.625	.002176	72	164	9.44	14.55
5000	973.0	19.875	.001857	72	168	8.84	10.9
6000	1167.6	21.15	.001646	73	169	8.70	8.94
7000	1362.2	22.9	.001528	78	170	8.62	7.6
Table No. 3 – 1800 R.P.M.							
1000	194.6	12.00	.005600	69	160	10.14	94
2000	389.2	14.625	.003415	74	172	8.27	38.25
3000	583.8	16.95	.002640	90	179	7.57	23.4
4000	778.4	17.875	.002087	96	186	6.79	15.7
5000	973.0	19.65	.001835	106	197	5.88	10.88
Table No. 4 – 3000 R.P.M.							
500	97.3	8.45	.0079	94	178	6.52	201
750	146	11.1	.00692	87	173	7.01	144
1000	194.6	11.5	.00537	101	184	6.02	92.8
1250	243	12.5	.00466	104	187	5.8	71.6
1500	292	13.5	.0042	108	193	5.3	54.5
1750	340.6	14	.00373	114	197	5.08	44.75
2000	389.2	14.68	.00343	128	199	4.94	38.1
2500	486	16.5	.00308	132	204	4.68	28.9
Table No 5 – 3600 R.P.M.							
500	97.3	9.125	.00852	102	183	6.11	226
750	146	10.375	.00646	115	192	5.39	133
1000	194.6	11.800	.00552	115	195	5.19	96
1250	243	12.250	.00458	129	206	4.53	67
1500	292	14.000	.00436	121	201	4.85	59.8

stant within certain limits. The formula must be considered however as purely empirical.

According to McKee (3), in an ordinary journal bearing of certain fixed dimensions and clearance ratio

$$\mu = \phi\left(\frac{ZN}{P}\right) + \Delta\mu \quad [2]$$

It is well known that, in the ordinary journal bearing, the clearance ratio greatly affects the coefficient of friction as well as the load-carrying capacity. A small clearance ratio means high load-carrying capacity with a high coefficient of friction and vice versa. For this reason, heavily loaded, slow- or moderate-speed journal bearings have a clearance ratio of 0.001 or less and lightly loaded high-speed bearings a clearance ratio of 0.002 or more.

In view of the test results, the coefficient of friction for the multiple-oil-film radial bearing can be written $\mu = \phi\left(\frac{ZN}{P}\right)^x$. This results in the equation

$$\mu = 567(10^{-6})\sqrt{\frac{ZN}{P}} \quad [3]$$

which fits all the test results within the limits of reasonable errors of observation. The graph of Equation [3] is shown in Fig. 5.

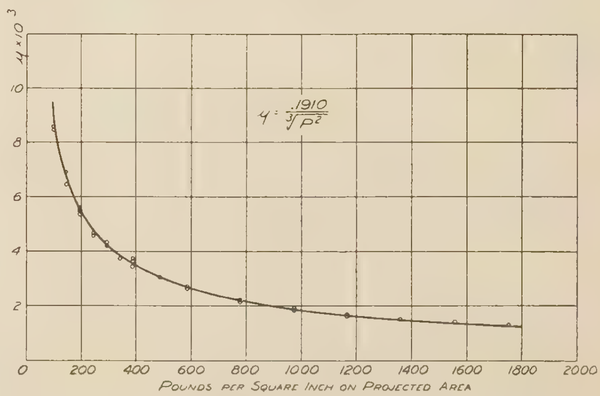


Fig. 4 Curve of Coefficient of Friction Versus Unit Bearing Pressure From Tests

It is unlike in form the equation for the coefficient of friction in ordinary journal bearings, according to McKee and McKee (3)

$$\mu = K\left(\frac{ZN}{P}\right)\left(\frac{D}{C}\right) + \Delta\mu \quad [4]$$

where

$$\Delta \mu = \phi \left(\frac{L}{D}\right)$$

It is, however, similar in form to the generally accepted equation for the coefficient of friction of tilting-block thrust bearings which is (7)

$$\mu = 152(10^{-6}) \sqrt{\frac{ZN}{P}\,(n)} \quad\ldots\ldots\ldots\ldots\ldots [5]$$

where n = number of blocks.

The usual thrust bearing has six blocks, as has also the test bearing under consideration. If we insert this value in Equation [5], we have

$$\mu = 372(10^{-6}) \sqrt{\frac{ZN}{P}} \quad\ldots\ldots\ldots\ldots\ldots [6]$$

This shows that the conventional type of tilting-block thrust bearing has about 34 per cent less frictional loss than the multiple-oil-film radial bearing, a comparison which is not unreasonable to expect. In a six-block thrust bearing, all blocks are effective as load-carrying members, whereas, in the multiple-oil-film radial bearing only two blocks out of six carry the load. The

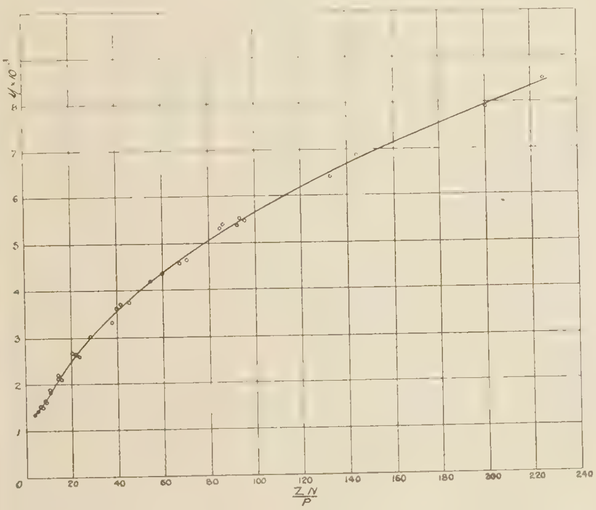


FIG. 5 CURVE OF COEFFICIENT OF FRICTION μ PLOTTED AGAINST ZN/P

other four blocks serve no useful purpose for a unidirectional load but, on the contrary, act as brakes due to their hydraulic drag. This is also true in the ordinary journal bearing where the pressure producing oil-film length occupies only an arc of from 90 to 120 deg.

The clearance ratio does not affect the frictional losses in multiple-oil-film radial bearings to the extent that it does in the ordinary journal bearing. Clearance ratios, corresponding to usual practice with ordinary journal bearings for different applications and speeds, were first used. Ratios of from 0.0005 to 0.002 had apparently little or no effect on frictional losses, but lately further experiments with larger clearance ratios indicate a critical point where the friction becomes a minimum, after which it increases again to values almost corresponding to clearance ratios of 0.002 or less. This critical zone seems to have clearance ratios around 0.003. Tests show a reduction of friction of about 20 per cent.

In examining the coefficient of friction, shown in Fig. 5, it is

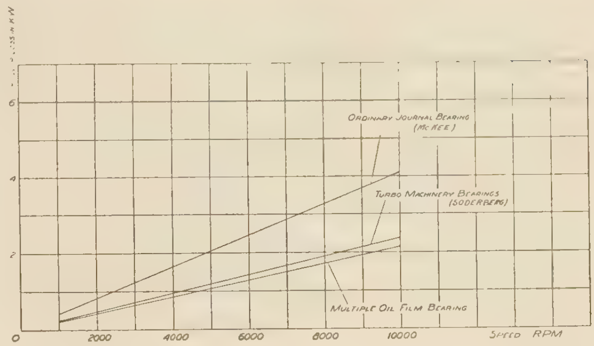


FIG. 6 COMPARISON BETWEEN POWER LOSSES IN MULTIPLE-OIL-FILM RADIAL BEARING AND JOURNAL BEARING OF CORRESPONDING SIZE

Specifications

Ordinary journal bearing	Multiple-oil-film bearing	Turbomachinery bearings
D = 4.2796 in.	D = 4.2796 in.	D = 4.2796 in.
L = 4.2796 in.	L = 1.2 in.	L = 4.2796 in.
L/D = 1	L/D = 0.28	L/D = 1
C/D = 0.002	C/D = 0.00185	C/D = 0.002
P = 150 psi	P = 536 psi	P = 150 psi
ZN/P = 170	ZN/P = 30	ZN/P = 170

$\mu = 473 \times 10^{-10} \left(\frac{ZN}{P}\right)\left(\frac{D}{C}\right) + \Delta\mu = 0.0059$ $\quad \mu = 567 \times 10^{-6} \times \sqrt{\frac{ZN}{P}} = 0.0031$ $\quad \mu = 1268 \times 10^{-8}\left(\frac{ZN}{P} + 100\right) = 0.0034$ (Soderberg)

$\Delta\mu$ = 0.0018 when
L/D = 1 (McKee)

well to bear in mind that these values might be reduced by 20 per cent through the use of a larger clearance ratio than that used in these tests.

UNIT PRESSURES PERMISSIBLE ON BEARING

As pointed out earlier in this paper, the development of the multiple-oil-film radial bearing was based on the ideal of a bearing so short as to be virtually free from alignment errors due to shaft deflection within the scope of the bearing and, consequently, capable of sustaining much greater unit pressures. The tests will speak for themselves as to the extent to which this ideal has been attained, but it should be clearly realized that, in order to take full advantage of this development, it is essential to utilize the high bearing pressures now permissible, together with low but conservative values of ZN/P. It is suggested that for bearings starting under full load, unit pressures should range from 400 to 500 psi, and for bearings that do not receive their maximum loading until rotation has started, pressures of the order of 1000 psi may be safely used.

For special applications, such as rolling-mill roll-neck bearings, bearing pressures as high as 5000 psi may be used. The values of ZN/P should preferably not exceed 50, whereas tests have been run with values as low as 2.

If these recommendations are followed, it is possible to effect a substantial decrease in the power losses of radial bearings. Fig. 6 shows a comparison between the power losses in the multiple-oil-film radial bearing and an ordinary journal bearing of corresponding size, having a coefficient of friction according to McKee and McKee and a similar one according to turbine practice as reported by Soderberg (7).

The coefficient of friction as reported by Soderberg is very low

$$\mu = 1268(10^{-8})(ZN/P + 100) \quad\ldots\ldots\ldots\ldots [8]$$

From Equation [8], the power loss is

$$\text{Kw} = 0.075Z \left(\frac{D^2L}{1000}\right)\left(\frac{N}{1000}\right)^2\left(1 + \frac{100}{ZN/P}\right) \quad\ldots\ldots [9]$$

The power loss in the multiple-oil-film radial bearing is

$$\text{Kw} = 3.36(10^{-9})(D^2L)(Z^{1/2})(P^{1/2})(N^{3/2}) \ldots\ldots\ [10]$$

It is obvious that the reason for the substantial power saving with the multiple-oil-film radial bearing is the small ZN/P selected, as compared with the corresponding values commonly used for the ordinary journal bearing.

Since the multiple-oil-film radial bearing usually has a slightly greater journal diameter for a given shaft size than that of an ordinary plain bearing, using the shaft itself as its journal, it might be assumed on first thought that the plain bearing should show an inherently lower power loss. The relative lengths of the two types of bearings must also be taken into consideration however.

Once the coefficient of friction has been determined for a certain type of bearing, the only physical dimensions that will affect the power loss are D and L.

If D = journal diameter of a multiple-oil-film bearing
L = journal length in a multiple-oil-film bearing
D_1 = journal diameter in an ordinary bearing
L_1 = journal length in an ordinary bearing

then if $(D^2 \times L) = (D_1^2 \times L_1)$, the power loss is the same as far as the physical dimensions are concerned.

If

$$L = 0.25D$$

and

$$D_1 = L_1$$

then

$$0.25 \times D^3 = D_1^3$$

and

$$D = \sqrt[3]{4} \times D_1 = 1.5874D_1 \ldots\ldots\ldots\ [11]$$

It is therefore apparent that the journal diameter of a multiple-oil-film radial bearing, having the foregoing relationship between L and D can be 59 per cent greater than the shaft diameter and still have no greater power loss than an ordinary plain bearing using the shaft as its journal. In actual practice, the journal-hub diameter is about 23 per cent greater than the shaft bore so that the multiple-oil-film radial bearing shows an advantage over the ordinary plain bearing from the standpoint of purely physical dimensions. The multiple-oil-film radial bearing can, of course, as has been previously stated, be mounted directly on the shaft, but it is preferable to use a special journal member, as ordinary shaft material is not as suitable for a good journal surface.

It is believed that the foregoing presentation indicates a marked advance toward the ideal bearing having maximum life and load-carrying capacity, together with minimum frictional losses.

BIBLIOGRAPHY

1 "On the Theory of Lubrication and Its Application to Mr. Beauchamp Tower's Experiments," by Osborne Reynolds, Royal Society Philosophical Transactions, vol. 177, pt. 1, 1886, pp. 157–234.

2 "First and Second Reports on Friction Experiments," by Beauchamp Tower, Proceedings of The Institution of Mechanical Engineers, 1883, pp. 632–659; 1884, pp. 29–35; and 1885, pp. 58–70.

3 "Friction of Journal Bearings as Influenced by Clearance and Length," by S. A. McKee and T. R. McKee, Trans. A.S.M.E., vol. 51, 1929, paper APM-51-15, pp. 161–171.

4 "The Laws of Lubrication of Horizontal Journal Bearings," by M. D. Hersey, *Journal*, Washington Academy of Sciences, vol. 4, 1914, pp. 542–552.

5 "Effects of Side Leakage in 120-Degree Centrally Supported Journal Bearings," by S. J. Needs, Trans. A.S.M.E., vol. 56, 1934, pp. 721–732; discussion, vol. 57, 1935, pp. 135–138.

6 "The Lubrication of Plane Surfaces," by A. G. M. Michell, *Zeit. für Mathematik und Physik*, vol. 52, 1905, pp. 123–138.

7 "Bearing Problems of Large Steam Turbines and Generators," by C. R. Soderberg, The Institution of Mechanical Engineers; general discussion on lubrication, vol. 1, 1937, pp. 285–296.

8 "A Graphical Study of Journal Lubrication, Part III," by H. A. S. Howarth, Trans. A.S.M.E., vol. 47, 1925, pp. 1073–1099.

9 "Optimum Conditions in Journal Bearings," by A. Kingsbury, Trans. A.S.M.E., vol. 54, 1932, paper RP-54-7, pp. 123–148.

10 "Zur hydrodynamischen Theorie der Schmiermittelreibung," by A. Sommerfeld, *Zeit. für Mathematik und Physik*, vol. 50, 1904, pp. 97–153.

11 "The Theory of Film Lubrication," by R. O. Boswall, Longmans, Green, & Company, New York, N. Y., 1928, pp. 206 and 227.

12 "Oil Film Bearings With Perfect Lubrication," by H. A. S. Howarth, Kent's Mechanical Engineer's Handbook (Wiley Engineering Handbook Series, vol. III), 1938, pp. 13–15, Fig. 9.

13 "Progress of Fluid-Film Lubrication," by A. G. M. Michell, Trans. A.S.M.E., vol. 51, 1929, paper MSP-51-21, pp. 153–163.

14 "The Wedge-Shaped Oil Film Does the Trick," by J. Haydock, *American Machinist*, vol. 83, 1939, pp. 823–825.

15 "The Nomy Pad Bearing: Development and Design," by F. K. G. Odqvist, The Institution of Mechanical Engineers; general discussion on lubricaton, vol. 1, 1937, pp. 227–233.

16 "Pivoted Pad Bearings," by J. Hamilton Gibson, The Institution of Mechanical Engineers; general discussion on lubrication, vol. 1, 1937, pp. 107–114.

17 "Tilting-Pad Bearings and Their Practical Limitations," by A. G. M. Michell, The Institution of Mechanical Engineers; general discussion on lubrication, vol. 1, 1937, pp. 196–203.

Discussion

E. B. Etchells.[3] The author's explanation of the 20 per cent increase in friction coefficient of the multiple-oil-film bearing over the conventional six-block thrust bearing is interesting. The statement of the braking action of the unloaded shoes can be confirmed by some of our experiments, in which we found as much as 20 per cent of the loss in a unidirectional loaded plain bearing to be due to an excess supply of oil.

However, our calculations on the two types of bearings, under the same conditions, show the multiple-film bearing to have an increased friction coefficient of 92 per cent over the six-block thrust bearing, instead of 20 per cent. The increase in power loss is 124 per cent. The differential of friction coefficient and power-loss increase, when the total load is the same on both bearing types, may be explained by the angularity of the two shoes in respect to the vertically imposed load. The braking action of the unloaded shoes is not included. The method employed in connection with the multiple-oil-film bearing investigation checked the curve of Fig. 6 of the paper very well, using the physical data of Tables 1, 2, 3, 4, and 5, and the bearing specifications in connection with Fig. 6. The method used for the six-block thrust bearing was that as presented by Howarth.[4]

The author shows that the multiple-oil-film bearing may have a 59 per cent greater diameter than an ordinary plain bearing with the same power loss. The basis of this is the assumption that, if the coefficient of friction remains constant in both bearing types, the power-loss differential is due to a change in diameter and length. This is odd, in view of Needs's work on optimum 120-deg bearings.[5] This study shows how the l/w ratio may be varied as much as 130 per cent with only a 6.5 per cent change in friction coefficient. The author's analogy attributes to the ordinary bearing, a disadvantage which does not appear to be entirely substantiated by prior bearing studies.

[3] Assistant Head, Mechanical Engineering Department, Research Laboratories Division, General Motors Corporation, Detroit, Mich.

[4] "The Loading and Friction of Thrust and Journal Bearings With Perfect Lubrication," by H. A. S. Howarth, Trans. A.S.M.E., vol. 57, 1935, pp. 177–178.

[5] "Effects of Side Leakage in 120-Degree Centrally Supported Journal Bearings," by S. J. Needs, Trans. A.S.M.E., vol. 56, 1934, pp. 721–732.

R. A. BAUDRY.[6] The flexible-sleeve bearing developed by the author is a very ingenious application of the hydrodynamic theory of lubrication, and of the modern manufacturing methods which have already made possible the high unit pressure, low friction, flood-lubricated sleeve-type, roll-neck bearing.

For more than 10 years the Westinghouse Electric & Manufacturing Company has used very successfully, on its large umbrella-type vertical waterwheel generators, a narrow large-diameter guide bearing made of pivoted pads.[7] This pivoted-pad or shoe-type guide bearing is usually placed at the periphery of the runner of a Kingsbury bearing and run in the same oil bath. This guide bearing, having a width of 3 to 8 in. and a diameter of 30 to more than 100 in., has replaced guide bearings of smaller diameter and much larger area placed on the shaft. However, on extremely high-speed machines, guide bearings of the latter type are still used because of their lower friction losses, due to their smaller diameter.

In recent years the Westinghouse Electric & Manufacturing Company has also been experimenting with pivoted-pad journal bearings. Some have already been used on machines where the load can act in any radial direction and they have proved very successful at very high unit loading.

The flexible-sleeve multiple-oil-film radial bearing should be applied very successfully where a large load has to be carried in a very limited space. However, its use will probably be limited to small sizes, for in large sizes it is believed that the pivoted-pad bearing is more efficient, easier to adjust, and cheaper to build than the flexible-sleeve bearing.

Because of the complexity of the problem involved, bearings of different types can be compared only if they are tested under absolutely identical conditions. The average temperature of the oil film, which should preferably be used to determine the viscosity of the oil in the factor ZN/p, is very difficult to measure or evaluate. Usually the temperature in the bearing, which is believed to be the nearest to the average temperature, is used. In the references cited by the author, McKee and McKee use the temperature near the babbitt under the load, and Soderberg uses the temperature of the oil at the outlet of the bearing which is kept below 140 F.

It would be very interesting to know how the author determined the mean oil-film temperature in the test log of his paper. Some of the temperatures recorded in that test log seem to be somewhat higher than values accepted in practice. When operating at a lower temperature, the friction losses of the multiple-oil-film bearing will of course increase.

According to the experience of the Westinghouse Electric & Manufacturing Company, friction losses increase with the diameter of a bearing, even if allowance is made for the increase in bearing pressure. This is shown in Table 6 of this discussion,

TABLE 6 RESULTS OF TESTS ON VARIOUS BEARING TYPES BY WESTINGHOUSE ELECTRIC & MANUFACTURING COMPANY

Type of bearing	L/D	Diam, in.	Length in.	Unit pressure, psi	Friction, 900 rpm, $Z = 14$	Losses kw 3000 rpm, $Z = 7.5$
Pivoted-pad bearing	1.5/1	2.4	3.6	300	0.105	0.507
Journal bearing[8]	2/1	2.92	5.85	150	0.13	0.59
Turbomachinery[9]	1.5/1	3.38	5.06	150	0.168	0.7
Multiple-oil-film	1/3.5	4.287	1.2	500	0.194	0.61 ($Z = 6.3$)
Turbomachinery[9]	1/1	4.13	4.13	150	0.204	0.85

[6] Mechanical Engineer, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Mem. A.S.M.E.

[7] "Shoe-Type Guide Bearings for Hydro-Generators," by R. Baudry, *Power*, vol. 79, 1935, p. 541.

[8] "Journal Bearing Performance," by R. Baudry and L. M. Tichvinsky, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 57, 1935, p. A-121.

[9] "Tests of a 7 × 10½-In. Bearing at 3600 Rpm," by L. M. Tichvinsky, Trans. A.S.M.E., vol. 60, 1938, pp. 393–397.

where the friction losses in different types of bearings used by the company are compared with the loss of a multiple-oil-film bearing. All these bearings carry the same load of 2500 lb.

On many types of machines having to start under full load, it is also very important to keep the diameter of the bearing to a minimum in order to limit the starting torque.

The Westinghouse Electric & Manufacturing Company is using journal bearings having a ratio $\left(\frac{\text{length}}{\text{diameter}}\right) = 2$ on most of its large rotating equipment. This bearing appears to be the best compromise from the standpoint of low starting torque, low friction losses, liberal oil-film thickness, and low manufacturing costs.

W. F. OBERHUBER.[10] As a user of the author's bearing, the writer can report very satisfactory results. We have had some failures but in most cases they could not be attributed to the bearing itself, but rather to circumstances surrounding the installation.

Today, we have six bearings of this type in operation, the original installation having been in service approximately 3 years. These bearings, because of their size (diameter and length), can be mounted as replacements of ball bearings, or in any place where a limited-life bearing is used, with satisfactory results. However, it should be remembered that these bearings must be mounted where it is possible to remove the runner from the shaft in case of failure. It must also be remembered that unless the bearing is made with the shells split, *B* and *C*, Fig. 2 of the paper, as has been done on our bearings, experimentally this bearing cannot be used in a practical way on some mountings since it is not always practical to remove the rotor to replace bearings. It is the writer's experience that, even with the split shell, it would be questionable whether the runner *A*, Fig. 2, would be in a condition for replacement of the split bearing. Since our original installation, we have changed to the solid type, as shown by the author, with excellent results. This bearing is a very timely development and very practical when used where limited-life bearings have not the endurance and life expectancy desired.

A. G. M. MICHELL.[11] This paper is welcome as bringing before American engineers the economy of space and reduction of frictional losses which can be effected by the use of segmental journal bearings. As the author points out, the condition essential for attaining these results is the adoption of intensities of loading much higher than those sanctioned by earlier practice, and in order that this may be practicable it is necessary that the axial length of the bearing surfaces shall be relatively small and that the accuracy of finish of the bearing surfaces shall be of a high order.

Hitherto the development of segmental, or multiple-film, journal bearings complying with these conditions has been almost entirely confined to Europe. Besides the Michell journal bearing dating from 1911, and the "Nomy" bearing, mentioned by the author, reference may be made to Table 7 which presents in skeleton form a history of the development of multiple-film journal bearings of all the three known types, viz., pivoted, flexible, and floating. In the last-named type the pads automatically form wedge-shaped lubricating films on both their inner and outer faces, and the series of multiple pads rotates as a whole relatively both to the journal and to the stationary member of the bearing.

Recent developments in America, which up to the present ap-

[10] Superintendent, Operating Maintenance, Philadelphia Electric Company, Philadelphia, Pa. Mem. A.S.M.E.

[11] Melbourne, Australia.

TABLE 7

Pivoted journal bearings

Name	Country	Patent no.	Date
Michell	Gt. Britain	23496/1911	24/10/11
Kingsbury	U. S. A.	1,117,505	3/ 3/13
Brown, Boveri & Co.	Gt. Britain	19801/14	26/ 1/14
Vickers Ltd.	Gt. Britain	121422	17/ 4/18
Haniel & Lueg	Germany	368608	14/ 3/22
Briggs	Gt. Britain	211707	26/ 2/23

Multiple-pad flexible journal bearings

Name	Country	Patent no.	Date
Brown, Boveri & Co.	Germany	312489	21/ 1/17
H. G. Reist	Australia (U. S. A., Serial No. 159976)	5858/17	5/ 4/17
Michell	Australia	3870/17	5/ 5/17

Floating-pad journal bearings

Name	Country	Patent no.	Date
Bostock & Moore	Gt. Britain	187,497	2/12/21
Hulsebos	Norway	43,243	28/10/25
Shebat	U. S. A.	2,076,254	9/ 9/32
Michell & Seggel	Australia	101,857	10/ 9/36

NOTE: For a discussion of the advantages and disadvantages of the flexible type as compared with the other two types see "The Mechanical Properties of Fluids," Blackie & Son, Second edition, 1936, p. 155 (First edition, 1923).
Tests of the Michell form of these bearings are reported by A. Tenot in the "Discussion on Lubrication," The Institution of Mechanical Engineers, London, 1937.

pear to be of a more or less tentative nature, include the adoption of the early Michell pivoted type of journal pad by some of the prominent makers of high-speed grinding tools.

While the author's graphs of his experimental results show, by the close adherence of the individual results to smooth curves, convincing evidence of effective film action of some kind, it seems doubtful to what extent the performance of the bearings depended on the elastic deformation of the pads as suggested by the author.

In view of the relatively large diametral clearance (8/1000 in. for a journal of 4.28 in. diam), it is clear that it was only for a very limited arc of the bearing that the oil films could be sufficiently thin to support loads of the intensities recorded. Within this small distance the thickness of the film can only have been affected to a relatively small extent by changes of shape of the blocks due to their elastic deformation. Such being the case, it is not clear that the author's form of bearing presents any material advantage over a plain cylindrical bearing of similar proportions and of equal refinement of workmanship.

The experiments of Stanton[12] have shown that cylindrical bearings are capable of carrying loads greater than those employed in the author's experiments, and with even lower coefficients of friction. In Stanton's experiments the clearance ratios were still greater than in the author's bearing and the arcs of effective film action were estimated to extend not more than 10 degrees on each side of the line of closest approach of journal and sleeve.

Bearings with such large diametrical clearances unavoidably allow radial movements of the journal, thus debarring them from use in many important applications (machine tools, electric motors, high-speed engine crankshafts), for which they might otherwise be acceptable. The advantages in this respect of the floating-pad type of bearing are discussed in the present writer's paper listed as reference (6) in the bibliography of the author's paper.

Author's Closure

The author is grateful to have had his paper discussed by such eminent authorities. Apparently, lack of clarity at some point in the paper has created a certain confusion of thought which he will attempt to correct.

Mr. Etchells' reference to the author's alleged statement that the multiple-oil-film bearing showed an increase of 20 per cent in the coefficient of friction over the conventional tilting-block thrust bearing begs correction. The author stated: "The conventional type of tilting-block thrust bearing has about 34 per cent less frictional loss than the multiple-oil-film radial bearing."

Using the same syntax as used by Mr. Etchells, this statement would read: "The multiple-oil-film bearing showed an increase of 52 (not 20) per cent in the coefficient of friction over the six-block thrust bearing."

The author based his comparison on information of the coefficient of friction of thrust bearings contained in reference (7) for the correctness of which he naturally cannot assume responsibility. It is believed, however, to be entirely reliable as it emanates from an excellent and experienced authority.

In computing the coefficient of friction as well as the power loss in a six-block thrust bearing, according to Howarth,[4] is it not possible that Mr. Etchells has overlooked the fact that, in a double thrust bearing, the inactive end of the bearing exerts a considerable hydraulic drag, as does also the cylindrical portion of the periphery of the runner and the shaft? These "additional" losses are considerable, especially so in high-speed machinery with a flooded bearing housing, and reflect themselves in a higher coefficient of friction than that computed on purely theoretical grounds for a single set of thrust blocks.

Equation [5] from reference (7) was used by the author not so much for bringing out a comparison between the numerical values of the coefficients of friction of the two different types of bearings as for its general mathematical form showing the similarity in characteristic behavior of the multiple-oil-film radial bearing to that of the conventional tilting-block thrust bearing.

In commenting on the author's exposition of power losses in bearings having the same coefficient of friction and the same pressure intensities but differing in length and diameter, Mr. Etchells has apparently confused Needs's[5] l/w ratio with the author's L/D ratio. In Needs's[5] paper, l = actual length of the film in the direction of motion, and w is the width of the film in a direction transverse thereto. In the author's paper L = the axial length of the bearing = w in Needs's paper, and D is the diameter of the journal. In the multiple-oil-film bearing the l/w ratio is maintained constant irrespective of how L/D may vary within certain practical limits. In the two bearings being compared as an example, it is assumed that the l/w ratio of the film or films has approximately the same optimum value.

Under such conditions, it is obvious that the power losses must vary, with close approximation, as the square of the diameter and directly as the length. The product of the two represents a numerical value of the physical dimensions of the bearing to which the power loss is proportional.

It was far from the author's intention to discredit the ordinary journal bearing, but to show an advance in the art, as represented by the multiple-oil-film bearing, described in his paper, based on facts substantiated by extensive tests and a great number of highly successful installations.

Mr. Baudry has given a very interesting exposition of the bearing practice of The Westinghouse Electric & Manufacturing Company. It is only natural that, in such a large concern, manufacturing a wide variety of machinery, the bearings are especially designed to fit the application. Thus, where a small starting torque is paramount to any other consideration, the diameter of the journal is made as small as possible.

As pointed out by Mr. Baudry, bearings of different types cannot be compared, unless they are tested under absolutely identical conditions. Of the examples shown in his Table 6, the last one designated "Turbomachinery,"[9] having a diameter of 4.13 in. and a length of 4.13 in., comes closest in dimensions to the

[12] "Some Recent Researches on Lubrication," by T. E. Stanton, Proceedings of The Institution of Mechanical Engineers, 1922, p. 1117.

multiple-oil-film bearing under discussion, which has a diameter of 4.28 in. This turbomachinery bearing shows a power loss, according to Mr. Baudry's Table 6, of 0.85 kw, as compared to 0.61 kw for the multiple-oil-film radial bearing under the same load of 2500 lb and the same speed of 3000 rpm. This proves a somewhat greater saving in power consumption by the multiple-oil-film radial bearing than indicated by the graph for turbomachinery bearings in Fig. 6 of the paper.

The oil-film temperature was determined by means of a potentiometer and a copper-constantan thermocouple, placed in the center of the transverse oil groove *d*, immediately following a loaded block. The thermocouple was exposed to the oil flowing over it from the loaded block. Due to the divergence of the oil-film space at the trailing end of the block, oil is pulled in from the circumferential grooves *e*. It is believed that the temperature of the mixture fairly well represents the effective mean temperature of the pressure film.

Mr. Oberhuber has given an interesting account of his experiences with commercial installations of the multiple-oil-film radial bearing under discussion, and points out the advantages of a split bearing. Whereas it is possible to build split bearings of this type, such construction adds greatly to the cost. To meet the requirements specified by Mr. Oberhuber, one-half bearing sleeve *B*, which can readily be replaced, may be used for a unidirectional load. However, when it is remembered that ball and roller bearings are not split and that the life of the multiple-oil-film bearing is virtually unlimited, the author is of the opinion that, as a standardized product for universal application, it is better to retain the construction shown in Figs. 1 and 2. The author wishes to express his appreciation to Mr. Oberhuber and his company for having been among the first of the public-service corporations to give this new bearing development a trial on important installations. It is just such an open attitude, by large public-service corporations, toward new developments, which promotes progress in engineering.

Mr. Michell, F.R.S., has very kindly furnished Table 7, showing the history, in outline form, of the European progress in the art of multiple-oil-film bearings of various types. In this connection, it may be mentioned that of late years many United States patents have been issued on the subject, indicating a marked interest in and appreciation of the virtues possessed by this type of bearing. No doubt we will see great developments along these lines in the near future.

Tests, carried out with sensitive micrometer dial gages, prove that elastic deformation takes place, confirming theoretical computations. If the sleeve *B* lacks in sufficient flexibility, or if no flexibility at all is provided, the bearing will carry only a relatively small load safely and with greatly increased friction.

The pressure intensities recorded in the test log are based on the projected area of the bearing ($D \times L$), corresponding to an arc of 180 deg, and not an arc limited to the active or pressure-producing film length, such as was the case in Sir Thomas Stanton's[12] experiments. In Stanton's experiments at the National Physical Laboratories, Teddington, England, the clearance ratios were purposely made so abnormally large (0.020 and 0.060) as to limit the arc of the pressure-producing film length to but a few degrees (30 deg and 15 deg, respectively) for the purpose of throwing light "on the cause of the high efficiency of worm gears."[13] Under such abnormal conditions, one would expect a higher pressure intensity to be reached on the active film area with a lower coefficient of friction than in a commercially built 360-deg bearing, designed for actual all-around service.

However, if we examine the data of the experiments made by Stanton, we find that the pressure intensity on the projected area ($D \times L$) was only 277 psi, and the pressure intensity on the active film area was only 1055 psi for the 30-deg arc, and 2110 psi for the 15-deg-arc bearing. Of the four tests reported, only one showed a coefficient of friction lower than the author's test, namely, 0.00072 for sperm oil. The other three tests showed higher coefficients of friction, namely, 0.0017, 0.0023, and 0.0035.

In the author's opinion it was not Dr. Stanton's intention that bearings having such abnormal clearance ratios would have any practical use. It is obvious that such bearings would have the excessive radial movements, mentioned by Mr. Michell, which would render them unfit for the applications cited.

The highest pressure intensity on the projected area ($D \times L$) recorded in the author's test log (1751 psi) is by no means the maximum pressure which this bearing will carry. Numerous tests have been made with pressure intensities on the projected area ($D \times L$) of more than twice this amount, and a somewhat lower coefficient of friction. The lowest ZN/P in the test log is 4.65, and tests have been run with less than one half this value.

The clearance ratio of 0.00185, used in the author's test bearings, is quite normal for average conditions in a general power-transmission bearing. For special applications mentioned by Mr. Michell, this would be modified to suit the occasion. The use of an extremely small clearance is fraught with danger of seizure during the starting-up period, due to the temperature gradient between the journal and the bearing sleeve with its surrounding housing, and especially so if the housing is of rather substantial construction. Clearance ratios less than 0.0005 are not considered safe from an operating point of view unless forced oil circulation for cooling purposes is employed to limit this temperature gradient to but a few degrees.

The multiple-oil-film bearing units under discussion have been so designed and rated commercially that, without artificial cooling, and depending only upon natural convection, the maximum temperature rise will not exceed 40 F above the ambient.

[13] "Friction," by T. E. Stanton, Longmans, Green & Company, New York, N. Y., 1923, pp. 101–110.

Progress Report on Tubular Creep Tests

By F. H. Norton,[1] Cambridge, Mass.

This paper is a progress report describing the results which have been obtained on the creep of tubular specimens. The first part of this work was described in a previous paper.[2] In this investigation similar apparatus and similar specimens are employed, but an attempt is made to correlate the stress between the tensile specimens and the tubular specimens more accurately. Also, tubes with thinner walls have been used. The results obtained to date indicate that the longitudinal creep of the tubes is substantially zero, and that the circumferential creep is approximately the same as the creep in the tensile specimen with a stress equivalent to the circumferential stress in the tube. In a companion paper,[3] Professor Soderberg derives the relations in two-dimensional creep, and applies them to the experimental values.

The data presented in this paper are of the nature of a progress report, concerning the results obtained on tubular specimens tested under internal pressure for circumferential and longitudinal creep. This work follows closely a previous investigation[2] and is intended to complete the data given there. The apparatus and the method of testing are exactly the same as previously described.[2]

Specimens

Seven tubular specimens were made up for this test by The Babcock and Wilcox Company, consisting of tubes 4 in. outside diam, with either a 3/8-in. or a 1/8-in. wall, having hemispherical ends welded in place. The specimens themselves were of the same carbon-molybdenum steel previously reported, and, in fact, were made from the same stock.

In Table 1 is given the schedule of the seven tests proposed

TABLE 1 SCHEDULE OF TESTS

	Tensile coupons		Tubular specimens			
Test temp, F	Loading of T.S., psi	Creep rate T.S., per cent per 100,000 hr	Wall thickness of tube, in.	Internal pressure in tube, psi	$\frac{PR^*}{t}$	Circumferential creep rate of tube, per cent per 100,000 hr
900	13000	0.3—	1/8	..	...	Not started
			3/8	..	...	Not started
900	20000	1.2—	1/8	1490	23000	1.5
			3/8	4780	23000	0.9
1050	5000	1.2	1/8	372	5750	Now running
1050	8000	8.5	1/8	596	9200	8.7
			3/8	1911	9200	6.5

* Based on mean radius.

for the tubular specimens, together with the four equivalent tests on the tensile specimens. Four of the seven tests have been practically completed, the fifth one has been running a short time, and the sixth and seventh have not yet been started.

Results on Tensile Test Specimens

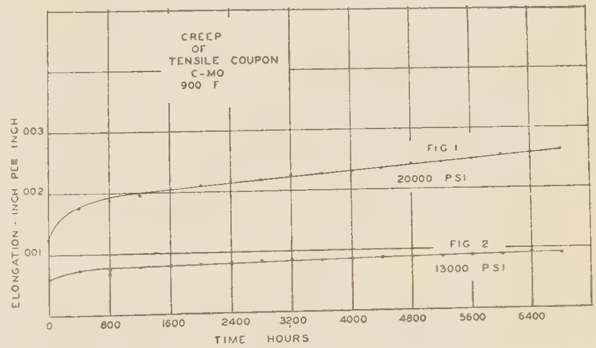
These specimens, cut from the 3/4-in. wall of the tubular stock before it was turned and bored, were run on regular tensile-creep apparatus, using a 10-in. gage length. The rate of creep of these specimens under specified conditions is shown in Figs. 1 to 4, inclusive. It will be noted that the tests have run for a period of around 7000 hr and that the original curve, wherein daily readings are carefully plotted, is sufficiently close to a straight line to insure that a stable condition has been reached.

Creep of Tubular Test Specimens

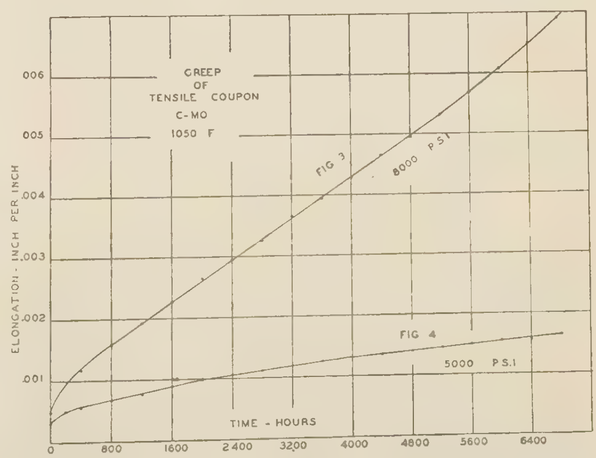
The creep of these specimens is shown in Figs. 5 to 8, inclusive. In all cases, the longitudinal creep is substantially zero, while the circumferential creep is definite. It should be noted that the specimen at 1050 F and 1911 lb internal pressure developed a leak and had to be shut off at 2200 hr. The specimen at 900 F and 4780 lb internal pressure had to be shut off for the same reason at 2700 hr. A discussion of the cause of these leaks will be described later in this paper, because they seem of considerable practical interest in the design of tubular pressure vessels.

Failure of Hemispherical Ends

The hemispherical ends, as in previous tubular specimens, were designed to give the same dilation on the basis of elastic formula



Figs. 1 and 2 Creep of Carbon-Molybdenum Tensile Coupon at 900 F



Figs. 3 and 4 Creep of Carbon-Molybdenum Tensile Coupon at 1050 F

[1] Massachusetts Institute of Technology.

[2] "Creep in Tubular Pressure Vessels," by F. H. Norton, Trans. A.S.M.E., vol. 61, no. 3, 1939, pp. 239–245.

[3] "Interpretation of Creep Tests on Tubes," by C. R. Soderberg, Trans. A.S.M.E., vol. 63, 1941, pp. 737–740.

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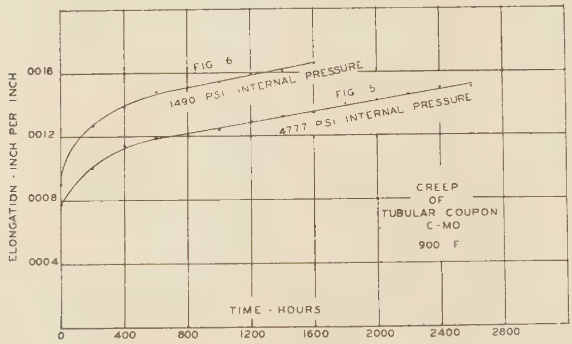
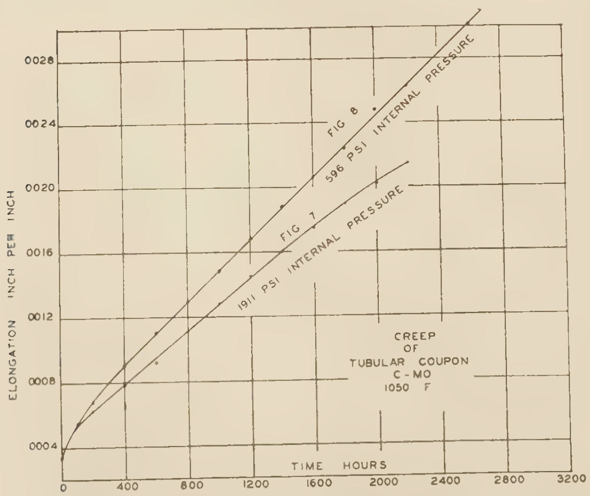
as the cylinder and were, therefore, one half of the thickness of the tubular section. The thickness of the hemispherical ends at the time of failure was 0.14 in. on the specimen at 1050 F, in comparison with a thickness of 0.375 in. for the tubular section. The stress, therefore, in two directions on the hemispherical ends at failure was approximately 12,400 psi from calculations by Professor Soderberg. The 900 F specimen had an end thickness of 0.165 in. at failure, which, in the same way, gives a stress of 26,300 psi.

While temperature measurements were not made on the hemispherical ends during the first test, measurements made since show that the hemispherical heads may be from 5 to 15 deg higher than the temperature of the cylinders themselves. This higher temperature might result in approximately double the creep rate that would exist in the cylinder. In the case of the first end failure, there was a final stress of 8000 psi on the cylinder at 1050 F, giving a creep rate of 6.5 per cent in 100,000 hr. On the hemispherical end we could then expect a creep rate of 3 times the amount, previously given, due to the higher stress to which the hemispherical end was designed. Due to the higher temperature, we might expect a maximum of double that creep rate. The combination of these two would, therefore, give us a rate of about 6 times the creep rate in the tube itself, or 39 per cent. It would therefore appear that at such a rate failure in carbon-moly steel at 1065 F occurred in 2240 hr, the stress being approximately equal in two directions.

In the second case, with 4780 lb internal presure on the cylinder



Figs. 5 and 6 Creep of Carbon-Molybdenum Tubular Coupon at 900 F



Figs. 7 and 8 Creep of Carbon-Molybdenum Tubular Coupon at 1050 F

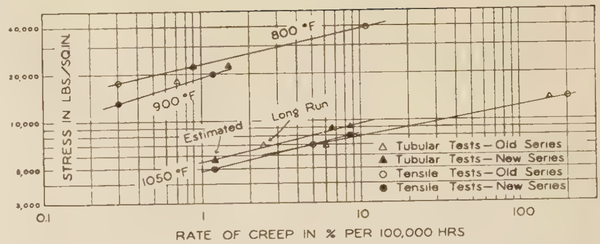


Fig. 9 Summary: Results on Old[2] and New Series of Creep Tests

at 900 F, the creep rate was 0.9 per cent per 100,000 hr in the cylinder. Therefore by the same method of calculations we might expect a creep rate in the hemispherical ends of approximately 6 per cent. It seems hard to understand the failure of this head under these conditions; further investigation is being made of this.

It is possible that, with the higher creep rate on the hemispherical ends, a stress-concentration effect may have existed close to the cylinder proper. At any rate both failures took place close to the junction of the hemispherical ends with the main cylinders, and the cracks were circumferential in relation to the cylinder.

Metallurgical examination has shown that for the 1050 F end the material has spheroidized. In the case of the 900 F end apparently no changes have taken place in the material.

It is contemplated on the next $^3/_8$-in-thick cylinder to weld additional thickness to the hemispherical ends so that on this sample we shall be able to conduct the test for a longer period than 2000 hr. In the case of the $^1/_8$-in-thick cylinders, one of which is now under test, the hemispherical ends were made $^3/_{32}$ in. thick so that there will not be any excessive stress in these ends, and every effort will be made to hold the temperature of the ends close to that of the cylinders themselves.

In considering why we did not obtain failures on the first series of tests in the hemispherical ends, it would appear that this was due to the fact that most of these ends were welded over due to porosity of the metal, and the two which were not so welded over operated only 1100 hr.

The diagram, in Fig. 9, gives a summary of the results obtained here and in the first part of the work by plotting rate of flow against stress. It will be seen that the results at 1050 F are in excellent agreement with our previous figures, but that the results at 900 F are closer to the 800 F values than we would have anticipated. It is shown in general that the rate of flow in the tubes is slightly less than that in the tensile specimens, although it should be remembered that the numbers of the values are still tentative as the tests have not been completed.

Summary

It is not attempted here to discuss the agreement between the tensile and tubular specimens, as this is being taken care of in a separate paper[3] by Professor Soderberg.

Acknowledgment

The author wishes to express his thanks for great assistance received from The Babcock & Wilcox Company in loaning the furnaces for the tensile tests and making up the tubular specimens. Mr. H. J. Kerr and Mr. J. B. Romer, of that company, have been particularly helpful with suggestions for the tests. Thanks are also due to Professor Soderberg for calculating the pressures used.

[Note: A joint discussion of this paper and the paper, "Interpretation of Creep Tests on Tubes," by C. R. Soderberg, appears on pages 740–748 of this issue of the Transactions.—Editor.]

Interpretation of Creep Tests on Tubes

By C. R. SODERBERG,[1] CAMBRIDGE, MASS.

In a companion paper,[2] Professor Norton has presented the results of creep tests on tubes, carried out under Project 10, of the Joint Research Committee on Effect of Temperature on the Properties of Metals. The present paper contains a discussion of the phenomena involved in the light of the accepted theory of yielding, the theoretical basis for the present research program, and the significance of the results thus far obtained.

The Theory of Yielding

In this study of creep phenomena, it is assumed that there is available a set of creep curves for the material in question, covering the appropriate temperature range of stress. In general, these creep curves will give the total strain ϵ as a function of the time t. It is further assumed that these creep curves approach straight lines, Fig. 1, with minimum slope u, which depends in some manner upon the stress σ, Fig. 2. This relation is the basic part of the experimental information available from the tensile creep test. The actual form of this relation is not important in the present connection.

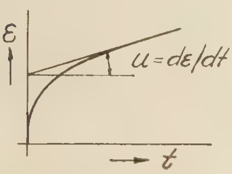


Fig. 1 Typical Creep Curve

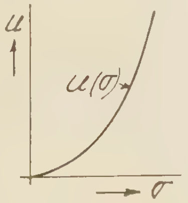


Fig. 2 Creep Rate-Stress Relation

It is next assumed that the same material, at the same temperature, is subjected to a three-dimensional state of stress, defined by the principal stresses, σ_1, σ_2, and σ_3. The resulting creep will take the form of an increase with time of the principal strains, ϵ_1, ϵ_2, and ϵ_3. Based on the experience from the tensile test, it may be expected that this creep will settle down to three minimum creep rates, u_1, u_2, and u_3. The theory of creep is based on the assumption that the relation between these creep rates and the corresponding state of stress is contained in the σ-u relation obtained from the tensile test.

Such a theory was presented a few years ago by R. W. Bailey.[3] In a discussion[4] of that paper, the author pointed out that the Bailey theory appears needlessly complicated, and that the necessary elements of a satisfactory theory are already contained in the premises for the Mises-Hencky criterion of yielding, which has been generally verified for the case of ductile materials at room temperature. There is nothing in the situation at elevated temperature which would indicate the need for additional assumptions.

The basic assumptions of the Mises-Hencky criterion of yielding are isotropy and constancy of volume, although in the development of the theory this is not always apparent. The former is one of those ideals which is never quite fulfilled, but which forms a suitable background against which our experimental results may be projected. The latter assumption appears reasonably well fulfilled for the plastic deformations, but volume changes under elastic deformations are unavoidable. However, if the application is confined to stresses which are constant in time, this objection is removed. This is an important restriction, which applies to the tensile creep test as well.

In general, the three-dimensional state of stress σ_1, σ_2, σ_3 may be resolved into a "hydrostatic tension"

$$p = \frac{\sigma_1 + \sigma_2 + \sigma_3}{3} \qquad [1]$$

and a "remainder" represented by the principal stresses

$$\left.\begin{aligned} s_1 &= \sigma_1 - p = \frac{2}{3}\left[\sigma_1 - \frac{1}{2}(\sigma_2 + \sigma_3)\right] \\ s_2 &= \sigma_2 - p = \frac{2}{3}\left[\sigma_2 - \frac{1}{2}(\sigma_3 + \sigma_1)\right] \\ s_3 &= \sigma_3 - p = \frac{2}{3}\left[\sigma_3 - \frac{1}{2}(\sigma_1 + \sigma_2)\right] \end{aligned}\right\} \qquad [2]$$

Similarly, the state of strain ϵ_1, ϵ_2, ϵ_3 may be resolved into the "dilatation"

$$q = \frac{\epsilon_1 + \epsilon_2 + \epsilon_3}{3} \qquad [3]$$

and a remainder represented by the principal strains

$$\left.\begin{aligned} e_1 &= \epsilon_1 - q = \frac{2}{3}\left[\epsilon_1 - \frac{1}{2}(\epsilon_2 + \epsilon_3)\right] \\ e_2 &= \epsilon_2 - q = \frac{2}{3}\left[\epsilon_2 - \frac{1}{2}(\epsilon_3 + \epsilon_1)\right] \\ e_3 &= \epsilon_3 - q = \frac{2}{3}\left[\epsilon_3 - \frac{1}{2}(\epsilon_1 + \epsilon_2)\right] \end{aligned}\right\} \qquad [4]$$

The stress Equation [1] and the strain Equation [3] relate to the change of volume; the stress Equation [2] and the strain Equation [4] relate to the change of shape. The latter may be called "deformation stresses" and "deformation strains," respectively.

If the volume remains constant, the phenomenon of yielding is reduced to a change of shape, produced by the deformation stresses. The Mises-Hencky theory expresses this observation by defining "the amount of yielding" as the resultant of the deformation strains $\sqrt{e_1^2 + e_2^2 + e_3^2}$ and "the cause of the yield-

[1] Professor of Applied Mechanics, Massachusetts Institute of Technology. Mem. A.S.M.E.

[2] "Progress Report of Tubular Creep Tests," by F. H. Norton, Trans. A.S.M.E., vol. 63, 1941, pp. 735–737.

[3] "Design Aspect of Creep," by R. W. Bailey, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 58, 1936, p. A-1. This paper is an abridgment of a more comprehensive paper, "The Utilization of Creep-Test Data in Engineering Design," by R. W. Bailey, Proceedings of The Institution of Mechanical Engineers, vol. 131, 1935, pp. 131–349.

[4] Discussion by C. R. Soderberg of "Design Aspect of Creep," by R. W. Bailey, *Journal of Applied Mechanics*, Trans. A.S.M.E., vol. 58, 1936, p. A-150.

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ing" as the resultant of the deformation stresses $\sqrt{s_1^2 + s_2^2 + s_3^2}$. The stress-strain relation of the tensile test may thus be looked upon as the evidence of a more fundamental relation between these resultants. This is the basis for the introduction of the "intensity of strain"

$$e = \sqrt{\frac{2}{3}} \sqrt{e_1^2 + e_2^2 + e_3^2}$$

$$= \frac{\sqrt{2}}{3} \sqrt{(\epsilon_1 - \epsilon_2)^2 + (\epsilon_2 - \epsilon_3)^2 + (\epsilon_3 - \epsilon_1)^2} \quad \ldots\ldots\ldots [5]$$

and the "intensity of stress"

$$s = \sqrt{\frac{3}{2}} \sqrt{s_1^2 + s_2^2 + s_3^2}$$

$$= \frac{1}{\sqrt{2}} \sqrt{(\sigma_1 - \sigma_2)^2 + (\sigma_3 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2} \quad \ldots\ldots\ldots [6]$$

These terms were introduced by Hencky[5] for the quantities $\sqrt{e_1^2 + e_2^2 + e_3^2}$ and $\sqrt{s_1^2 + s_2^2 + s_3^2}$, respectively. The reason for departing from this convention is that the quantities s and e as defined become identical with σ and ϵ for the conditions of the tensile test ($\sigma_1 = \sigma$, $\sigma_2 = \sigma_3 = 0$; $\epsilon_1 = \epsilon$, $\epsilon_2 = \epsilon_3 = -\epsilon/_2$). The rate of change of e may thus be looked upon as an "intensity of creep," depending directly upon the intensity of stress in accordance with the σ-u relation obtained from the tensile test.

Since the volume remains constant, the deformation components e_1, e_2, and e_3, Equation [4], are identical with the strain components ϵ_1, ϵ_2, and ϵ_3. The intensity of creep may thus be defined as the quantity

$$u = \frac{de}{dt} = \sqrt{\frac{2}{3}} \sqrt{u_1^2 + u_2^2 + u_3^2} \quad \ldots\ldots\ldots [7]$$

In the development of the theory, it is assumed that this intensity of creep distributes itself along the principal axes in the same manner as the intensity of stress is resolved into the deformation stresses. The mathematical formulation of this assumption is

$$u_1 : s_1 = u_2 : s_2 = u_3 : s_3$$

$$= \sqrt{u_1^2 + u_2^2 + u_3^2} : \sqrt{s_1^2 + s_2^2 + s_3^2} = \sqrt{\frac{3}{2}}\, u : \sqrt{\frac{2}{3}} \cdot s. \quad [8]$$

Note that in this expression u_1, u_2, and u_3 are the principal creep rates; s_1, s_2, and s_3 the deformation stresses; and u the creep rate which is measured in a tensile test at the stress s. To keep this relation in mind we will write $u(s)$ instead of u. By introducing the expressions Equation [2] for s_1, s_2, and s_3 this gives

$$\left.\begin{aligned} u_1 &= \frac{\sigma_1 - \frac{1}{2}(\sigma_2 + \sigma_3)}{s} u(s) \\ u_2 &= \frac{\sigma_2 - \frac{1}{2}(\sigma_3 + \sigma_1)}{s} u(s) \\ u_3 &= \frac{\sigma_3 - \frac{1}{2}(\sigma_1 + \sigma_2)}{s} u(s) \end{aligned}\right\} \quad \ldots\ldots\ldots [9]$$

which is the proposed form of the theory of yielding.

Application to Tube Under Internal Pressure

Consider next the case of a long tube with closed ends, subjected to internal pressure. The state of stress of such a tube may be defined by the tangential stress σ_1, the radial stress σ_2, and the axial stress σ_3. In the elastic state of stress, the tangential and radial stresses will vary throughout the tube thickness. As creep is started, these stresses are certain to alter, and the premise of constant stress is violated. Eventually, a steady state of stress will be reached, and from then on the stresses will remain constant. The determination of this state of stress is a matter of considerable difficulty, however, and for this reason the thick cylinder is not suitable for this type of experiment.

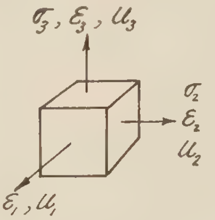

Fig. 3 State of Stress and Strain

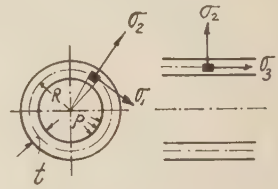

Fig. 4 Cylinder Under Internal Pressure

When the wall thickness of the tube is small in comparison with the radius, the change in stress will be small. The ideal of a truly constant stress may then be approximated by considering the stresses at the middle of the tube wall. With the notations of Fig. 4, the magnitude of these stresses may be determined in the following manner: The tangential stress σ_1 and the axial stress σ_3 are determined directly from the conditions of equilibrium with the internal pressure p. The radial stress σ_2 has a value $-p$ at the inner wall and zero at the outer wall, with a mean value of $-p/2$. This reasoning gives

$$\left.\begin{aligned} \sigma_1 &= \frac{pR}{t} - \frac{p}{2} \\ \sigma_2 &= -\frac{p}{2} \\ \sigma_3 &= \frac{pR}{2t} - \frac{p}{2} \end{aligned}\right\} \quad \ldots\ldots\ldots [10]$$

Introducing these stresses into Equation [6], we have for the intensity of stress

$$s = \frac{\sqrt{3}}{2} \cdot \frac{pR}{t} \quad \ldots\ldots\ldots [11]$$

Furthermore, the different factors of Equation [9] have the values

$$\left.\begin{aligned} \frac{\sigma_1 - \frac{1}{2}(\sigma_2 + \sigma_3)}{s} &= \frac{\sqrt{3}}{2} \\ \frac{\sigma_2 - \frac{1}{2}(\sigma_3 + \sigma_1)}{s} &= -\frac{\sqrt{3}}{2} \\ \frac{\sigma_3 - \frac{1}{2}(\sigma_1 + \sigma_2)}{s} &= 0 \end{aligned}\right\} \quad \ldots\ldots\ldots [12]$$

This gives for the three creep rates of the cylinder under internal pressure

$$\left.\begin{aligned} u_1 &= \frac{\sqrt{3}}{2} u \left(\frac{\sqrt{3}}{2} \frac{pR}{t}\right) \\ u_2 &= -u_1 \\ u_3 &= 0 \end{aligned}\right\} \quad \ldots\ldots\ldots [13]$$

[5] "The New Theory of Plasticity, Strain Hardening, and Creep, and the Testing of the Inelastic Behavior of Metals," by H. Hencky, Trans. A.S.M.E., vol. 55, 1933, paper APM-55-18, pp. 151–155.

The theory shows that the axial creep rate is zero. The tangential creep rate is $\sqrt{3}/2$ times the creep rate of a tensile test performed at the stress $\sqrt{3}/2 \times pR/t$. The radial creep rate (change in thickness) has the same value, but the opposite sign.

The tangential creep rate is obtained from the σ-u relation of the tensile test by entering with the stress s and multiplying the corresponding creep rate by $\sqrt{3}/2$. It is not necessary, however, to have access to the complete σ-u curve in order to check the theory. If there is available a single tensile creep test, with a minimum creep rate u at the stress σ, the tube test should be run at a pressure which will produce an intensity of stress equal to σ; that is, the pressure should be, from Equation [11]

$$p = \frac{2}{\sqrt{3}} \frac{\sigma t}{R} \quad \text{[14]}$$

At this pressure, the theory predicts a tangential creep rate of $\sqrt{3}/2 \times u$ and an axial creep rate of zero. The tests described by Professor Norton[2] were all planned with the object of proving or disproving this prediction.

In tests of this kind, it is never possible to make the ratio of thickness to radius strictly small, so that an exact check with the theory cannot be expected. The measurements of the deformations are of necessity made at the outside radius, so that a correction is required in order to obtain the creep rate at the mean radius.

If the creep rate recorded by Professor Norton is denoted by u_1' the rate of increase of the outside radius is $u_1'\left(R + \frac{t}{2}\right)$. It is reasonable to assume that the creep phenomena take place in such a manner as to preserve every element of the volume constant. This requires that the tangential creep be inversely proportional to the radius.[6] The rate of change of the mean radius is, therefore, $u_1'\left(R + \frac{t}{2}\right)\frac{R + \frac{t}{2}}{R}$ and, hence, the creep rate at the mean radius

$$u_1 = u_1'\left(1 + \frac{t}{2R}\right)^2 \quad \text{[15]}$$

An approximation of the same result is obtained by stating that the rate of increase of the outside radius is equal to the rate of increase of the mean radius, minus the rate of decrease of one half of the thickness. This gives

$$u_1 = u_1' \frac{1 + \frac{t}{2R}}{1 - \frac{t}{2R}} \quad \text{[16]}$$

When $t/2R$ is small, Equations [15] and [16] are identical. In the following discussion, Equation [15] will be used.

Comparison of Results With Theory

At the present time there are four tube tests which may be used as a check of the theory. It is probable that they have not yet reached the steady state, but the present results may be used to indicate the trend. These results are given in Table 1.

All experiments were made on tubes with an outside diameter of 4 in. The creep rate u_1' was measured from the curves; from this the creep rate u_1 was calculated by Equation [15]. At 900 F

[6] Note that a radial displacement ρ produces a tangential strain ρ/r and a radial strain $d\rho/dr$ at the radius r. Since there is no axial strain the sum $\rho/r + d\rho/dr$ must be zero to keep the volume constant. This gives, after integration, $\rho = \text{constant}/r$.

TABLE 1 RESULTS OF TUBE TESTS

Tensile tests: Temp, F	Tensile tests: σ stress, psi	Tensile tests: u creep rate, per cent in 100,000 hr	Tube tests: Dimensions, in. t	R	$t/2R$	p pressure, psi	Creep rates, per cent in 100,000 hr u_1'	u_1	Ratio u_1/u
900	20000	1.2 (7000)[a]	1/8	31/16	1/31	1490	1.5 (1400)[a]	1.6	1.33
			3/8	29/16	3/29	4780	0.9 (2600)	1.1	.92
1050	8000	8.5 (7000)	1/8	31/16	1/31	596	8.7 (2600)	9.3	1.09
			3/8	29/16	3/29	1911	6.5 (2200)	7.9	.93

[a] Indicates approximate duration of test in hours.

the $^1/_8$-in. tube gave 1.6 per cent and the $^3/_8$-in. tube 1.1 per cent, against a predicted value of $\sqrt{3}/2 \times 1.2 = 1.04$ per cent. At 1050 F the $^1/_8$-in. tube gave 9.3 per cent and the $^3/_8$-in. tube 8 per cent against a predicted value of $\sqrt{3}/2 \times 8.5 = 7.4$ per cent. The last column gives the ratio of tangential creep rate to creep rate of the tensile test, which theoretically should be $\sqrt{3}/2 = 0.867$.

All tube tests so far show higher creep rates than predicted by the theory, but the difference for the $^3/_8$-in. tubes is small. It is probable that the figures for the $^1/_8$-in. tubes will improve as the tests are continued.

While the final check on the tangential creep rates must await the completion of the project, it is significant that the prediction of zero axial creep was fulfilled in all cases.

At first sight the results obtained so far are not impressive but, when the circumstances of the tests are taken into account, they are probably as close as can be expected. Imperfections in isotropy, slight variations in temperature and dimensions, and other factors, not to mention the difficulty of measurements, may easily cause divergencies greater than those now obtained. For example, an error in thickness of 0.01 in. in the $^1/_8$-in-thick tube may be expected to alter the tangential creep rate by more than 50 per cent. This particular phase will be investigated more closely at the end of the tests and covered by a later report.

In conclusion it may be said that these creep tests have not revealed any serious discrepancy with the theory of yielding, but that the present program must be completed before a positive correlation may be obtained.

Conditions of the Hemispherical Ends

The problem of the failure of the hemispherical ends of some of these specimens is an instructive application of the theory of

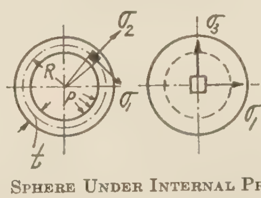

Fig. 5 Sphere Under Internal Pressure

yielding. Considering a thin sphere under internal pressure, it is not difficult to determine the principal stresses at the middle of the wall. With the notation of Fig. 5 and the same reasoning as that developed for the thin cylinder, it is found that

$$\left.\begin{aligned} \sigma_1 = \sigma_3 &= \frac{pR}{2t} - \frac{p}{2} \\ \sigma_2 &= -\frac{p}{2} \end{aligned}\right\} \quad \text{[17]}$$

Here σ_1 and σ_3 are the tangential stresses, now alike, while σ_2 is the radial stress.

Introducing these stresses into Equation [6] we have for the intensity of stress

$$s = pR/2t \quad \text{[18]}$$

The factors of Equation [9] have the values

$$\left.\begin{aligned} \frac{\sigma_1 - \frac{1}{2}(\sigma_2 + \sigma_3)}{s} &= \frac{1}{2} \\ \frac{\sigma_2 - \frac{1}{2}(\sigma_3 + \sigma_1)}{s} &= -1 \\ \frac{\sigma_3 - \frac{1}{2}(\sigma_1 + \sigma_2)}{s} &= \frac{1}{2} \end{aligned}\right\} \quad \text{[19]}$$

and the creep rates of the sphere are

$$\left.\begin{aligned} u_1 &= \frac{1}{2} u \left(\frac{pR}{2t}\right) \\ u_2 &= -u \left(\frac{pR}{2t}\right) \\ u_3 &= \frac{1}{2} u \left(\frac{pR}{2t}\right) \end{aligned}\right\} \quad \text{[20]}$$

If creep tests were conducted on a thin sphere, therefore, the tangential creep rate to be expected is one half of the tensile creep rate under the stress $pR/2t$. Such experiments might be of interest as a check on the theory of yielding.

In the present instance, it is proposed to apply these results to the hemispherical ends of the tube specimens. These were originally designed to give the same elastic tangential strain as the cylinder; this required a wall thickness of a little less than one half that of the cylinder. For various manufacturing reasons, however, the wall thickness became smaller than intended. Thus, the $^3/_8$-in-thick tube at 1050 F had a hemispherical end with a measured thickness of about 0.14 in. The mean radius was the same as that of the tube, namely, 29/16 inch. This gives for the intensity of stress in the hemisphere $\frac{1911 \times 29}{16 \times 2 \times 0.14} =$ 12,400 psi, which is considerably higher than the 8000 psi to which the tube was subjected, and which gave a tangential creep rate of 6.5 per cent. Assuming that the creep rate varies with the fifth power of the stress, the tangential creep rate of the hemispherical end would be, from Equation [20], $\frac{1}{2} \times \left(\frac{12{,}400}{8000}\right)^5 \times 8.5$ $\cong$ 38 per cent in 100,000 hr. This high rate is further increased by the rise of temperature at the end. The failure is quite well explained by the discrepancy between the rate in the hemispherical portion and the adjoining cylindrical portion. Fig. 6, which is reproduced from a photograph of the longitudinal section of this tube after the failure,[7] confirms this explanation.

The 900 F specimen at 4780 psi had a measured thickness of about 0.165 in.; the resulting stress is $\frac{4780 \times 29}{16 \times 2 \times 0.165} = 26{,}300$ psi, as compared with the stress of 20,000 psi in the cylindrical portion, and which produced a tangential creep rate of 0.9 per cent in 100,000 hr. In accordance with the theory cited this gives a creep rate of $\frac{1}{2} \times \left(\frac{26{,}300}{20{,}000}\right)^5 \times 1.2 = 2.35$ per cent in 100,000 hr. This rate is not particularly high, but it is probable that the stress intensity of 26,300 psi may be high enough to cause intercrystalline fractures, particularly in view of the higher temperature at the ends.

The failures are not surprising, particularly that of the tube at 1050 F. The theory of yielding indicates that in an application of this kind the stress intensity should be about the same in the end as in the cylindrical portion. This requires a wall end thickness of about 60 per cent of that of the cylinder. If there should be appreciable differences of temperature, the thickness must be increased further. The safest arrangement would probably be to maintain the thickness the same throughout the specimen.

This aspect of the experiments illustrates a practical application of the theory of yielding, which deserves attention in the design of structures of this kind.

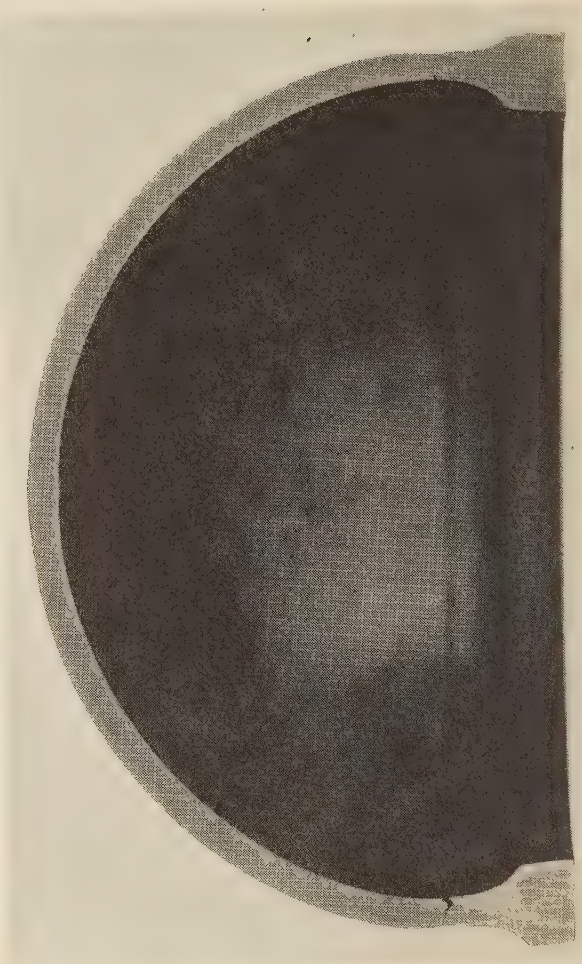

Fig. 6 Fracture of Tube Specimen

Discussion[8]

W. O. Clinedinst.[9] Professor Soderberg has aptly shown,

[7] "Failure of End Cap," by J. B. Romer, The Babcock & Wilcox Company, New York, N. Y., Babcock & Wilcox Engineering Reprint, June 21, 1940. (Not published.)

[8] This discussion applies jointly to the present paper and the paper, "Progress Report on Tubular Creep Tests," by F. H. Norton, which appears on page 735 of this issue of the Transactions.

[9] Engineer, Operating Department, National Tube Company, Pittsburgh, Pa. Jun. A.S.M.E.

both in his present paper and in his discussion[4] of a paper[3] by Bailey in 1936, that the postulates of creep do not conflict with those of stationary plastic flow. He has obtained from these postulates Equations [9], relating the principal creep rates in a three-dimensional stress system with the creep rate of a tensile creep specimen.

In applying Equations [9] of the paper to a tube under internal pressure, the author states that analysis of creep of a thick tube presents difficulties. The observation might be made that, if creep or yielding of the outer radius of a tube is obtained, the entire tube has yielded. If such is the case, then the theory of yielding of thick tubes is readily employed. Only the laws of stationary plastic flow which the author employed in developing Equations [9] are necessary.

Using the nomenclature of the author's paper for principal creep rates of the outer radius of the tube u'_1, u'_2, u'_3, and defining the tangential, radial, and axial stresses at the outer surface as σ'_1, σ'_2, σ'_3, the following expressions may be developed from the laws of stationary plastic flow[10]

$$\left.\begin{aligned} \sigma_1' &= \frac{2s}{\sqrt{3}} \\ \sigma_2' &= 0 \\ \sigma_3' &= \frac{s}{\sqrt{3}} \end{aligned}\right\} \qquad [21]$$

The following expressions relating the strain rate of the outer radius of the tube with that of a tensile creep specimen are obtained by substituting Equation [21] of this discussion into the author's Equation [9]

$$\left.\begin{aligned} u_1' &= \frac{\sqrt{3}}{2}\,u(s) \\ u_2' &= -\,u_1' \\ u_3' &= 0 \end{aligned}\right\} \qquad [22]$$

The condition for yielding of the outer surface of a thick-walled tube defines the "intensity of stress" as[10]

$$s = \frac{\sqrt{3}}{2}\,\frac{p}{\ln \cdot \dfrac{R + t/2}{R - t/2}} \qquad [23]$$

When t is small in comparison to R, Equation [23] may be expressed as

$$s \approx \frac{\sqrt{3}}{2}\,p\,\frac{R}{t} \qquad [24]$$

It is seen that Equation [24] corresponds to Equation [11] of the paper.

Combining Equations [13] and [15] of the paper, the following expression is obtained relating the creep rate of the outer radius of a thick tube to the creep rate of a tensile creep specimen, where the relationship of intensity of stress of the tube is equal to the stress on the tensile creep specimen

$$u_1' = \frac{\sqrt{3}}{2}\,\frac{u(s)}{\left(1 + \dfrac{t}{2R}\right)^2} \qquad [25]$$

Equation [25], based on the assumption of a thin tube, should be compared with Equation [22] of this discussion, obtained from the plastic yielding of a thick-walled tube. It is seen that the latter, Equation [22], results in a higher estimate of the creep rate for the outer radius of a thick tube relative to the creep rate of a tensile creep test than does Equation [25].

[10] "Plasticity," by A. Nádai, Engineering Societies Monographs, McGraw-Hill Book Company, Inc., New York, N. Y., 1931, p. 188.

D. S. Jacobus.[11] The Joint Research Committee sponsoring the present investigation was responsible for publishing data[12] on creep characteristics in 1938. The results of laboratory tests given in that compilation have been used for setting working stresses both in this country and abroad.

The Joint Research Committee has been responsible for securing contributions from the industry for conducting its work. The committee has been particularly fortunate in securing the services of as able men as the authors of the two papers, and it is hoped that they will follow up the work until it is completed.

In 1929 the writer presented a paper[13] before the Society, in which the working stresses at the higher temperatures were based upon the rate of creep, together with the results secured in practice. Since that time, many questions have come up regarding the dependence which can be placed upon creep tests for establishing the working stresses. Fundamental data such as secured in the tests will serve as a valuable contribution to the subject.

There is a broad field for perfecting the design of pressure vessels through avoiding abrupt changes in section which act as stress raisers. Important improvements are bound to come which will result in a saving of material without sacrificing safety. Fundamental data such as covered by the papers will be of inestimable value in making advances of the sort and the Joint Research Committee should be thanked for sponsoring the investigations. The authors of the papers should be commended for the able way in which they have conducted the work.

J. J. Kanter.[14] The interpretation which Professor Soderberg advances for Professor Norton's carefully conducted tests on steel tubes creeping under internal pressure is a promising approach to the problem. Whether or not the Mises-Hencky theory, as applied, affords an adequate analysis of the problem of creep under combined stress is open to some discussion.

At the temperatures at which Professor Norton conducted his tube tests, creep-strain rates seem to persist at any stress sensibly greater than zero. In previous discussions of creep phenomena,[15, 16] the writer has taken the view that the relationship between stress and creep-strain rate may be treated as viscous flow and that, upon the application of stresses, metals change their intrinsic viscosities. As stress is applied, the viscosity of the metal continuously decreases from an initial value at zero stress, so we may write that the creep rate u is the product of the stress σ and a "flowability" factor ϕ, which in turn is a function of the resolved stress in a system

$$u = \phi\sigma = \sigma f(\sigma) \qquad [26]$$

The stress function $f(\sigma)$, is generally found to be an exponential and can be represented graphically by plotting $\log\left(\dfrac{u}{\sigma}\right)$ versus σ.

[11] Advisory Engineer, The Babcock & Wilcox Company, New York, N. Y. Past-President A.S.M.E.

[12] "Compilation of Available Creep Characteristics of Metals and Alloys," Joint A.S.M.E.-A.S.T.M. Research Committee on the Effect of Temperature on the Properties of Metals, Creep Section, 1938.

[13] "Working Stresses for Steel at High Temperatures," by D. S. Jacobus, Trans. A.S.M.E., vol. 52, 1930, paper FSP-52-35, pp. 295–299.

[14] Research Metallurgist, Crane Company, Chicago, Ill. Mem. A.S.M.E.

[15] "Interpretation and Use of Creep Results," by J. J. Kanter, Trans. American Society for Metals, vol. 24, 1936, pp. 870–918.

[16] "The Problem of Temperature Coefficients of Tensile Creep Rate," by J. J. Kanter, Trans. American Institute of Mining & Metallurgical Engineers, vol. 131, 1938, pp. 385–418.

For much data such a plot approximates linear. Creep rates are frequently represented in the empirical relation

$$\frac{u}{u_0} = \left(\frac{\sigma}{\sigma_0}\right)^n \ldots\ldots\ldots\ldots [27]$$

where u_0, σ_0, and n are empirical constants for Equation [27] Writing the expression as

$$u = \sigma\left(\frac{u_0}{\sigma_0^n}\,\sigma^{n-1}\right) \ldots\ldots\ldots\ldots [28]$$

we find that the quantity $\left(\frac{u_0}{\sigma_0^n}\,\sigma^{n-1}\right)$ expresses the "flowability" of the material at the stress σ. Analytical expression of the relationship appearing on the semilog plot of $\frac{u}{\sigma}$ versus σ takes the form

$$\frac{u}{u_0} = \frac{\sigma}{\sigma_0}\,e^{f\left(\frac{\sigma}{\sigma_0}\right)} \ldots\ldots\ldots\ldots [29]$$

where u_0, σ_0 are empirical constants for Equation [29], and $\frac{u_0}{\sigma_0}\,e^{f\left(\frac{\sigma}{\sigma_0}\right)}$ is the flowability at σ.

These considerations focus attention upon the manner in which Professor Soderberg relates "stress intensity" and "creep intensity." Presumably the function $u(s)$ of the author's Equations [9] is intended as a measure of the flowability associated with the stress intensity of his Equation [6]. If such be the case, this is equivalent to the assumption that the submerged "hydrostatic stress" p makes no contribution whatsoever in affecting the viscosity of the material. It is furthermore an assumption that a given stress intensity, resulting from a set of combined stresses, is associated with the same identical viscosity value as a similar stress intensity, resulting from simple tension. Although the hydrostatic stress may play no part in propelling creep strain, it does elastically dilate the material. Likewise, the stress intensity elastically alters the structure of the material in various fashions, depending upon the stress composition. These elastic responses in the material are unquestionably accompanied by alterations of the viscosity and flowability. Therefore, in order to rectify the physical significance of Professor Soderberg's Equations [9], it would seem that $u(s)$ should be replaced by a function depending upon the total principal stresses σ_1, σ_2, and σ_3 defined, perhaps, in terms of the state of elastic dilation.

If the assumption be granted that flowability is a function of the state of elastic dilation, it becomes of interest to derive the corrections, which from the "viscosity" point of view should be applied to Professor Soderberg's calculations. In doing this, it is helpful to generalize somewhat our treatment of the deformation of an isotropic body by simply recognizing that a principal strain will be accompanied by strains in the planes normal to its axis. In the case of plastic deformation where no change in volume occurs, these strains in the normal planes are one half the magnitude of the principal strain. This, however, represents a special case, the more general situation being the sort of change in volume as obtains in elastic deformation, where the strain in the normal planes is usually expressed by Poisson's ratio λ. Suppose the strains in the planes normal to one of the principal stresses are permitted to react upon the other two principal strains. We are then concerned with the algebraic sums of principal strains superimposed upon normal strains

$$\left.\begin{aligned} e_1 &= \epsilon_1 - \lambda(\epsilon_2 + \epsilon_3) \\ e_2 &= \epsilon_2 - \lambda(\epsilon_1 + \epsilon_3) \\ e_3 &= \epsilon_3 - \lambda(\epsilon_1 + \epsilon_2) \end{aligned}\right\} \ldots\ldots\ldots\ldots [30]$$

where e_1, e_2, and e_3 are residual strains, ϵ_1, ϵ_2, and ϵ_3 are principal strains, and λ is Poisson's ratio. The "resultant strain," when related to the tensile test thus becomes

$$e = \sqrt{\epsilon_1^2 + \epsilon_2^2 + \epsilon_3^2 + \frac{2\lambda^2 - 4\lambda^2}{2\lambda^2 + 1}(\epsilon_1\epsilon_2 + \epsilon_1\epsilon_3 + \epsilon_2\epsilon_3)} \ldots [31]$$

For plastic deformation, when $\lambda = 1/2$, this expression becomes identical with the Mises-Hencky resultant. Thus, it appears feasible directly to argue this same result, without introducing the rather obscure concept of a hydrostatic portion of the stress.

The stress intensity S_e, associated with the elastic aspect of the deformation, may thus be assumed to be related to the resultant strain through Young's modulus E

$$S_e = eE = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 + \frac{2\lambda^2 - 4\lambda}{2\lambda^2 + 1}(\sigma_1\sigma_2 + \sigma_1\sigma_3 + \sigma_2\sigma_3)} \ldots [32]$$

The stress intensity S_p, associated with the plastic or creep aspect of the deformation, is related to the resultant strain through the flowability ϕ

$$S_p = \frac{u}{\phi} = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - (\sigma_1\sigma_2 + \sigma_1\sigma_3 + \sigma_2\sigma_3)} \ldots [33]$$

since $\phi = f(Se)$, then by Equation [26] of this discussion

$$u = S_p f(S_e) \ldots\ldots\ldots\ldots [34]$$

Thus we may write

$$S_p = \frac{1}{\phi}\sqrt{u_1^2 + u_2^2 + u_3^2 - (u_1u_2 + u_1u_3 + u_2u_3)} \ldots [35]$$

In order to subject Equation [34] of this discussion to calculations, we must assign a definite function to $f(S_e)$. A function of the type in the writer's Equation [29] has a definite physical significance which has been discussed elsewhere and may be conveniently used. Professor Norton's data are represented within the limits of experimental uncertainty by Equation [29] of this discussion in the form

$$\log_{10}\frac{u}{S_p} = \frac{S_e}{\sigma_0} + K \ldots\ldots\ldots\ldots [36]$$

where σ_0 and K are constants. Since for simple tension $S_p = S_e = \sigma$ by definition, Professor Norton's tensile creep rates may be used to compute σ_0 and K, as given in Table 2.

TABLE 2 VALUES OF CONSTANTS σ_0 AND K

Temp, F	σ	u	$\frac{u}{\sigma} \times 10^3$	σ_0	K
900	13000	0.3	0.0231	16900	—5.405
900	20000	1.2	0.06		
1050	8000	8.5	1.063	5290	—4.486
1050[a]	13930	195.0	14.0		

[a] This test is from Professor Norton's 1939 report, see footnote (17) this discussion.

Using Equation [32] of this discussion to calculate S_e, and introducing the principal stresses for the thin-walled tubes tested by Professor Norton and assigning a value of 0.275 to Poisson's ratio, this expression becomes

$$S_e = p\sqrt{0.837\left(\frac{R}{t}\right)^2 - 0.26\left(\frac{R}{t}\right) - 0.12} \ldots\ldots [37]$$

Using the writer's Equations [37] and [33], with the tube dimensions and pressures of Professor Norton's tests, we find the stress intensities affecting both the flowability and plastic flow as given in Table 3.

TABLE 3 STRESS INTENSITIES AFFECTING FLOWABILITY AND PLASTIC FLOW

Tube wall, in.	$\frac{R}{t}$	Pressure, psi	S_e	S_p
$^1/_8$	15.5	1490	20950	20000
$^3/_8$	4.84	4780	20350	20000
$^1/_8$	15.5	596	8370	8000
$^3/_8$	4.84	1911	8140	8000

The values for S_e, we see, are somewhat larger than the S_p values used by Professor Soderberg. In order to determine the ratios for flowability between these two sets of stress-intensity values, Equation [36] with the constants of Table 2 of this discussion may be used. Such calculations lead to the values in Table 4.

TABLE 4 FLOWABILITY RATIOS BETWEEN S_e AND S_p VALUES

Temp, F	Tube wall, in.	Flowability ϕ For S_p	Flowability ϕ For S_e	$\frac{\phi_e}{\phi_p}$	$\frac{u_1}{u}$	$\frac{u_1\phi_p}{u\phi_e}$
900	$^1/_8$	0.00006	0.000069	1.14	1.33	1.16
	$^3/_8$	0.00006	0.0000625	1.04	0.92	0.88
1050	$^1/_8$	0.001063	0.00125	1.18	1.09	0.92
	$^3/_8$	0.001063	0.00113	1.065	0.93	0.87

Thus, it appears that the margin of increase which the "elastic" stress intensity represents over the "plastic" provides very nearly the correction needed to make the ratio of circumferential creep rate to tensile creep rate approximate the theoretical ideal of $^1/_2\sqrt{3}$.

Reflecting upon Professor Soderberg's observation that the theory should provide a suitable background, against which our experimental results may be projected, makes us mindful that Professor Norton's previous report[17] on tubular creep tests showed where a small but definite longitudinal creep rate was observed. While theory predicts zero longitudinal creep rate, it must be remembered that the theory postulates isotropy, whereas, our metal does not entirely fulfill this ideal. Metal tubes have definite directional properties developed in the working of the material incident to their manufacture. That definite anisotropy in material leads to a considerable longitudinal creep in tubes under internal pressure is borne out by some experiments tried at the writer's laboratory using warm celluloid tubes under air pressure. Celluloid, as is well known, becomes highly birefringent under stress and is thus known to become anisotropic. These tubes showed a longitudinal contraction at a rate of about $^1/_6$ the circumferential expansion. A mathematical analysis of the effects of anisotropy on creep rates appears a difficult but challenging problem. In the light of the foregoing consideration, it would be necessary to deal with the effects of anisotropy, not only from the standpoint of plastic deformation, but also elastic dilation. Fortunately, however, the steels with which we ordinarily are concerned seem to follow satisfactorily the rules deduced from the assumption of isotropy.

R. M. Van Duzer, Jr.,[18] and Arthur McCutchan.[19] The results of Professor Norton's comparative tests on tensile and tubular specimens have been awaited expectantly by all having occasion to apply tensile creep data to the design of pipe and tubing. These tests and those now planned or under way should give a better appreciation of the problems involved in designing for creep conditions.

While Professor Norton has referred the question of correlating results of his tensile and tubular tests to an accompanying report by Professor Soderberg, the statement is made that "the circumferential creep is approximately the same as the creep in the tensile specimen with a stress equivalent to the circumferential stress in the tube." The purpose of the first part of this discussion is to examine the basis for this statement and to consider how the new test results compare with those previously reported.

From Professor Norton's Table 1, it may be observed that, in the case of tests at 900 F, the stress in the tensile specimen is 20,000 psi, as compared with a circumferential stress of 23,000 psi in the tubes. This difference in the stresses being compared arises from the fact that pressures in the tubes were purposely adjusted by application of the Mises-Hencky theory to produce an "intensity of stress" equal to the stress in the tensile specimen. This was deduced as 1.15 times the pressure, which corresponds to the circumferential stress of 20,000 psi. The circumferential creep rate of the tube under this higher pressure was predicted to be 0.867 times the creep rate found for the tensile specimen.

Regardless of the theory applied, it seems essential to the writers that creep rates be compared over the same intervals of time. The rates of creep existing at different intervals during the tests on both the tensile and tubular tests have been plotted in discussing Professor Soderberg's paper. It is apparent from such plotting that only in the case of the tensile test at 8000 psi and 1050 F is it possible to assume that a steady rate of creep has been obtained. Comparing rates of creep of tensile specimens at periods up to 7000 hr with those obtained from tubular tests of 1600 to 2700 hr duration, therefore, would not seem to be admissible.

In order to compare the creep rates of the present tube tests

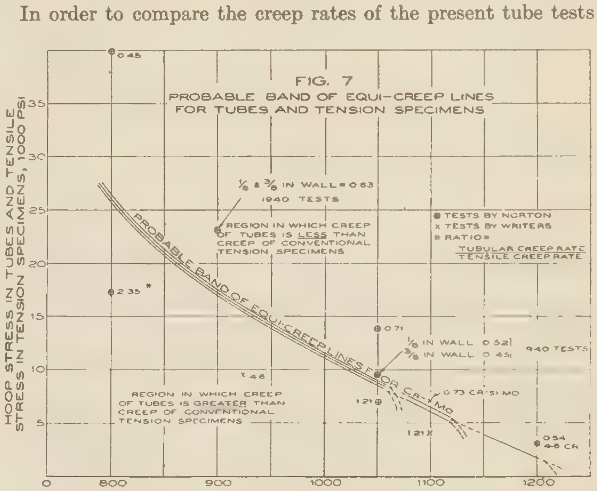

Fig. 7 Probable Band of Equicreep Lines for Tubes and Tension Specimens

with the tensile tests on the same basis as Professor Norton's earlier tests[17] and those reported by the writers,[20] it is necessary to extrapolate on a log-stress versus log-rate-of-creep curve to find the creep rates of the tensile specimens which correspond to the hoop stress in the tubes. By means of this rather debatable procedure, it was found that the rate of creep, measured on the outside of the tubes, was only about 63 per cent of that of the tensile specimen for both the $^1/_8$-in- and $^3/_8$-in-wall tubes at 900 F, and 52 and 45 per cent, respectively, for the $^1/_8$-in- and $^3/_8$-in-wall tubes at 1050 F. These results are plotted in Fig. 7 of this discussion, for comparison with previous data. It will be noted that both these test conditions fall in the region in which creep of tubes has been found to be less than creep of conventional tension specimens, when they are compared on the basis of creep

[17] "Creep in Tubular Pressure Vessels," by F. H. Norton, Trans. A.S.M.E., vol. 61, no. 3, 1939, pp. 239–245.

[18] Engineer, Production Department, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

[19] Engineer, Engineering Division, The Detroit Edison Company, Detroit, Mich. Jun. A.S.M.E.

[20] "High-Temperature-Steam Experience at Detroit," by R. M. Van Duzer, Jr., and Arthur McCutchan, Trans. A.S.M.E., vol. 61, 1939, pp. 383–401.

measurements on the outside of the tubes and simple hoop stress.[21]

It is important to relate actual measured circumferential creep of tubes with simple hoop stress since both the A.S.M.E. Boiler Code and the A.S.A. Piping Code determine pipe-wall thickness in terms of this stress. It seems reasonable that pipe should be designed for this more severe or open-end condition, since both longitudinal thrust and compressive bending stress tend to offset the beneficial effect of longitudinal tensile stress and to accelerate circumferential creep.

The tests for which results are given in Professor Norton's paper are for more rapid rates of creep than are contemplated in usual power-plant practice. The additional tests at lower rates of creep which are planned or under way will do much to confirm whether the circumferential creep of a closed-end tube is greater or less than might be expected from simple tensile creep tests.

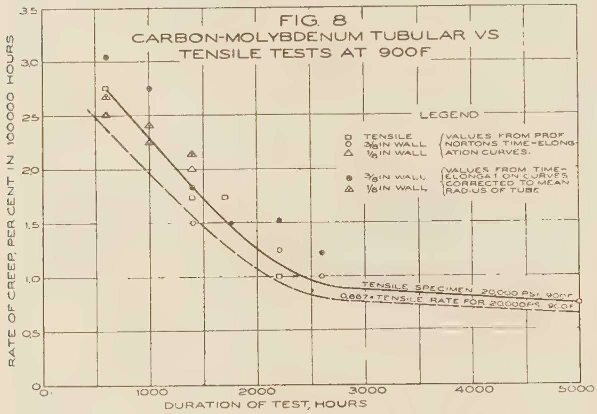

FIG. 8 CARBON-MOLYBDENUM TUBULAR VERSUS TENSILE TESTS AT 900 F

When considered for the same time intervals, Professor Norton's statement that "the circumferential creep is approximately the same as the creep in the tensile specimen with a stress equivalent to the circumferential stress in the tube" should be carefully qualified as to just what creep and what stress are intended. It would be particularly helpful if he could arrange to do so in his closure to the paper.

Turning now to Professor Soderberg's paper, it is apparent he has presented a logical argument that the theory of yielding, based on the Mises-Hencky concept, can be used to interpret creep under three-dimensional stress conditions. His conclusion that the circumferential stress in a closed-end tube can be permitted to be 15 per cent higher than stress in simple tension is in agreement with other authorities on theories of combined stress. Professor Norton's tests were stated to be planned to give direct confirmation of this concept. His conclusion, however, that the resulting circumferential creep rate at this higher circumferential stress will be only 86.7 per cent of the tensile creep rate at the lower tensile stress is not so apparent.

In order to examine more closely whether Professor Soderberg's theories were supported by the test results, the rates of creep existing at different intervals during the tests at 900 F on both the tensile and tubular specimens have been plotted in Fig. 8 of this discussion. These rates were determined from values of elongation read at 400-hr intervals from blueprints of Professor Norton's daily plotting of time-elongation curves. The curves, in Fig. 8, emphasize that it is essential to compare creep rates over the same intervals of time.

The tensile rates are compared with the tubular rates, corrected to the mean radius by Professor Soderberg's Equation [15], in Table 5 of this discussion. His method of correcting the measurements to determine the creep rate at the mean radius of the tube appears to have been an afterthought, since the diameters used in calculating unit elongations were not given in Professor Norton's report. It will be observed that the rates for the 1/8-in- and 3/8-in-wall tubes are reasonably consistent when considered at the same time intervals. Also, that the ratio of circumferential creep to tensile creep appears to be approximately unity or greater.

A similar comparison between tubular and tensile creep rates for the tests at 1050 F, likewise, showed more consistent relations when measured over the same time intervals than when tubular tests of 2600 hr duration were compared with tensile tests of 7000 hr.

Tensile and circumferential creep rates, when considered for the same intervals of time, seem to confirm Professor Soderberg's conclusion that the circumferential stress in a closed-end tube can be made 15 per cent higher than the simple tensile stress. Also, that his proposed means of adjusting measured changes in the outside diameter to give the creep rate at the mean radius is approximately correct. The results, however, do not appear to afford a demonstration that the circumferential creep rate at the mean radius of a tube having 15 per cent higher stress will be 86.7 per cent of the tensile creep rate as predicted by the author's theory. This is illustrated in Fig. 8 of this discussion, by the dashed curve.

It should be emphasized that these results are for fairly high rates of creep. The particular test conditions of 23,000 psi, 900 F and 9200 psi, 1050 F, are in the region in which creep of tubular specimens has been found to be less than tensile specimens when compared on the basis of equal hoop and tensile stresses.[21]

While the present tests seem to indicate that better correspondence exists between creep rates of closed-end tubes and tensile rates, if the circumferential tube stress is 15 per cent higher than the tensile stress, it should be remembered that in pipe design longitudinal thrust may balance the longitudinal force due to internal pressure and give the equivalent of an open-end condition. Bending moments also place one side of the pipe in compression. The greater circumferential creep of the side of a pipe under longitudinal compression due to bend

[21] Discussion of "Creep in Tubular Pressure Vessels," by R. M. Van Duzer, Jr., and Arthur McCutchan, *Mechanical Engineering*, vol. 61, 1939, pp. 757–758.

TABLE 5 COMPARISON OF RATES OF CREEP C-Mo AT 900 F TENSILE, AND TUBULAR TEST RESULTS, PER CENT PER 100,000 HR

Time, hr, mean of 400-hr intervals	Tensile rate, 20,000 psi stress, u	Circumferential rates 23,000 psi stress; values from time-elongation curves: 1/8-In. wall, u'_1	3/8-In. wall, u'_1	Circumferential rates 23,000 psi stress; values from time-elongation curves, corrected to mean radius: 1/8-In. wall, u_1	1/8-In. wall, u_1/u	3/8-In. wall, u_1	3/8-In. wall, u_1/u
600	2.76	2.50	2.50	2.67	0.97	3.05	1.1
1000	2.28	2.24	2.24	2.41	1.06	2.75	1.2
1400	1.83	2.00	1.50	2.14	1.17	1.84	1.0
2200	1.12	..	1.25	..	..	1.52	1.35
2600	0.94	..	1.00	..	..	1.22	1.30
5000	0.75	..	..	..	..	..	..

NOTE:

u = tensile creep rate, per cent in 100,000 hr.

u'_1 = circumferential rate, per cent in 100,000 hr, determined from prints of daily plotting of Professor Norton's time-elongation curves.

u_1 = circumferential rate, per cent in 100,000 hr, corrected to mean radius by Professor Soderberg's Equation [15]

$$u_1 = u'_1 \left(1 + \frac{t}{2R}\right)^2$$

where t = wall thickness, in.

R = mean radius of tube wall, in.

ing has been observed qualitatively on several occasions where pipe has become oval in service.

Thanks are due to Professor Soderberg for his interpretation of Professor Norton's test data and his rationalization of the manner in which measured elongations on the outside of tubes may be corrected to the mean radius. It is hoped that this re-examination of the creep rates, obtaining at comparable intervals throughout the tensile and tubular tests, will be considered as an effort to secure better correlation between the proposed theory and the test results rather than a criticism of the method of comparison used in this paper.

C. O. Rhys.[22] The writer has studied Professor Norton's results, in Table 1 of his paper, with considerable interest. The data from his tensile tests for a creep rate of 1.2 per cent per 100,000 hr, when plotted on semilog paper, Fig. 9 of this discussion, show a remarkable agreement with our design stresses for 4–6 per cent chrome-molybdenum steel. The slopes are identical and Professor Norton's stresses for 1.2 per cent per 100,000 hr are slightly more than ours for 1 per cent per 100,000 hr. Our design-stress curves follow Bailey's creep law and are average values of creep-stress data from a number of sources.

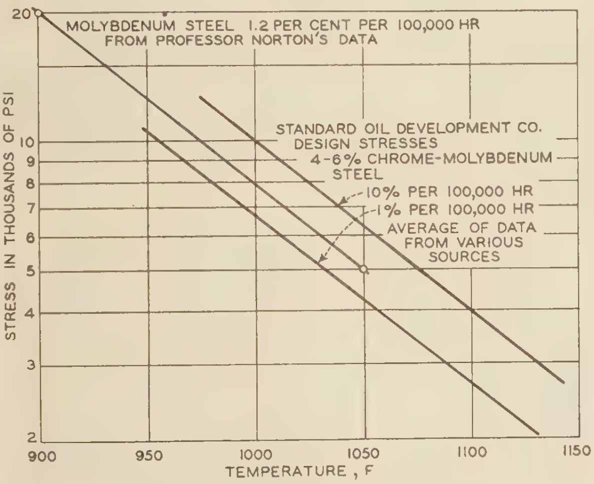

Fig. 9 Comparison of Professor Norton's Creep-Rate Data With Those of Various Others

We have also applied our tube-calculation procedure to Professor Norton's test values for tubular specimens, and give the results in Table 6. Our calculations are based on Bailey's theory, which does not require Professor Soderberg's assumption of a thin tube. We were enabled to determine the creep rate at the outside surface for direct comparison with Professor Norton's results. No correction for creep at the mean radius is necessary. The agreement is rather better than that obtained by the application of the Mises-Hencky theory. It will be noticed that the tests do not all show higher creep rates than those found by Bailey's theory, as they do for the Mises-Hencky theory, the differences being in both directions.

TABLE 6 RESULTS OF APPLYING STANDARD OIL DEVELOPMENT COMPANY'S TUBE-CALCULATION PROCEDURE TO PROF. NORTON'S TEST VALUES FOR TUBULAR SPECIMENS

Outside diam of tube	4 In. in all cases			
Thickness, in.	1/8	3/8	1/8	3/8
Metal temperature, F	900	900	1050	1050
Internal pressure, psi	1490	4780	596	1911
[a] Creep stress corresponding with circumferential creep rate at outside surface, psi	19000	18400	7650	7400
[b] Circumferential creep rate at outside surface, per cent per 100,000 hr	1.02	0.93	7.1	6.2
[c] Experimental values of foregoing creep rate, per cent per 100,000 hr	1.5	0.9	8.7	6.5

[a] Calculated by Bailey's equation. This is the stress in simple tension which will give the circumferential creep rate at the outside tube surface for axial and hoop stresses. Radial stress is zero.
[b] Calculated by Bailey's creep law. Physical constants taken from Table 1 of Professor Norton's paper.[1]
[c] From Table 1 of Professor Norton's paper.

We note Professor Soderberg's criticism of Bailey's theory as being unnecessarily complicated. As far as its application to a thick tube, subjected to internal pressure and heat transfer, is concerned, the reasoning is about as simple as that for the ordinary theory of thick cylinders. From Bailey's final equations we have devised a procedure by which tube calculations can be made easily and quickly. This check on Professor Norton's results is an example of the application of our procedure.

Ernest L. Robinson.[23] When the tests, which are the subject matter of the two papers under consideration, were projected several years ago, the writer pointed out that, if the results were to be used to check the theories for creep under compound stress, the specimens should be manufactured from a forged billet as nearly isotropic as it could be made.

It was, however, deemed more important to secure information on the creep behavior of tubes as usually manufactured, and the test specimens were, therefore, made to simulate actual piping, although they were made extra thick, so tensile-test specimens could be obtained out of the wall thickness. The method of manufacture was, therefore, such as might tend to some fibrous condition in a longitudinal direction. Just how important such effects are in the creep behavior of piping is not known. Indeed it was one of the objects of this test setup to try to evaluate these effects.

Six years ago a paper[24] was presented by L. L. Wyman which gave creep-test results showing that banded specimens of nickel-chrome-molybdenum steel might be only 1/8 or 1/4 as strong in creep as more homogeneous materials. Such nonuniform specimens may creep many times as fast as more uniform material. While the material used by Project 10 is undoubtedly much more uniform than that described by Wyman, yet we must be prepared to accept deviations of behavior from a theory based on isotropic material.

In this connection, it is also well to recall the fact that the Detroit test results, described by Van Duzer and McCutchan,[20,21] for the most part showed definitely more creep than called for by theory, even going so far as to exceed the creep in tension under the same load.

TABLE 7 CREEP RATES SCALED FROM AUTHORS' CURVES FOR VARIOUS TIMES

Temp, F	Nominal stress, psi	Type of test	Creep rates, per cent per 100,000 hr, at indicated time: 2000 hr		4000 hr	6000 hr
900	13000	Tension	0.2		0.2	0.2
900	20000	Tension	1.0		1.0	1.0
900	23000	Thin tube	2.0	2.15[a]	...	...
900	23000	Thick tube	1.6	1.95[a]	...	...
1050	5000	Tension	2.0		1.2	1.1
1050	8000	Tension	8.5		8.5	10.0
1050	9200	Thin tube	9.5	10.1[a]	...	...
1050	9200	Thick tube	8.0	9.75[a]	...	...

[a] Corrected for thickness in same way used by author.

Another important consideration in checking a theory is the legitimate range of interpretation of the test results. Regarded as creep tests, the data presented by Professor Norton appear highly satisfactory but, especially at the lower rates, there appear to be

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[23] Turbine Engineering Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

[24] "The Creep of Steels as Influenced by Microstructure," by L. L. Wyman, *Mechanical Engineering*, vol. 57, 1935, pp. 625–627.

legitimate differences of interpretation possible. Thus, the writer has scaled the resulting creep rates both from the small-scale curves accompanying the paper and from the larger-scale blueprints circulated to the subcommittee and has listed, in Table 7 of this discussion, the resulting creep rates at 2000, 4000, and 6000 hr for comparison with the results given by the authors.

The writer does not wish to substitute these results for those presented by the authors. The important point to note is the magnitude of the differences which may be regarded as legitimate differences of interpretation. Thus, the higher rates, given by the authors for the tensile tests at 900 F, can easily be obtained by scaling slopes at a time somewhat less than 2000 hr. On the other hand, the figure of 6.5 for the thick tube at 1050 F seems to the writer to correspond with a time at which leakage was occurring and, therefore, that a figure of 8 would better represent the test result. Furthermore, the writer scales larger slopes than the authors for each of the tube tests in contrast to smaller or equal slopes for the tension tests. This exaggerates the relative excess creep of the tube tests, as compared with theory.

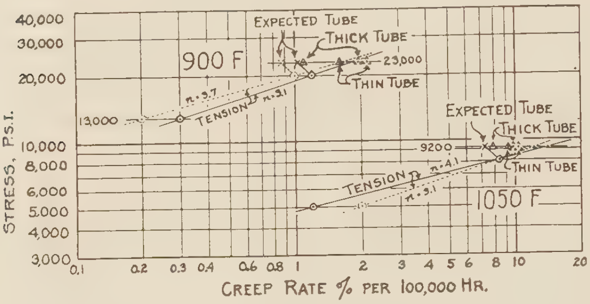

Fig. 10 Comparison of Measured With Expected Creep Rates in Tube Specimens

(The results of the tension tests as interpreted by the authors are shown by the full circles connected by the full lines. The discusser's interpretation of the same results is given by the dotted circles and dotted lines. The crosses at the higher bursting stress used in the tubes represent the creep rate in the tube expected in accordance with the theory. The measured creep rates in the tube, corrected to mid-thickness, are shown by triangles. The full triangles represent the interpretation given by the authors and the broken triangles the interpretation of the discusser.)

In the writer's opinion, when the lower stressed tube tests still on the program have been completed, a more profitable comparison with the theory will be possible. Thus, from the plotted results, it will be possible to compare the relative creep rates which would be caused by equal tensile stress and bursting stress at various levels instead of at a single level. To the ordinary person's mind such comparisons seem more understandable than when both stress and rate are varied.

Professor Soderberg has chosen a bursting stress unequal to the tension-test stress, because his theory shows that, regardless of the slope of Norton's log-log plot of rate versus time, a 15 per cent greater bursting stress will result in a 15 per cent smaller creep rate than that which occurs in the tensile specimen. Even if we did succeed in measuring exactly this set of ratios for these particular stress conditions and, in addition, note zero elongation of the tube as called for by theory, a wider comparison of relations would seem to constitute a more convincing justification of the theory.

Some years ago both Bailey[3] in England and Soderberg[4, 25] in this country published theories of creep under conditions of compound stress. It is to be noted that, when the bursting stress in the thin tube is made equal to the tensile stress in pure tension, the maximum tension stress and the maximum shear stress are the same in both cases. Under such conditions, the two theories referred to yield algebraic relations representing the ratio of the circumferential creep in the tube to the longitudinal creep in the tensile specimen as follows

According to Bailey

$$\frac{3^m\,(2^q + 1)}{2^{n+1}}$$

where $n = 2m + q$.

According to Soderberg

$$\frac{3^{\frac{n+1}{2}}}{2^{n+1}}$$

The more complicated expression given by Bailey is identical with that given by Soderberg in two cases, namely, when $q = 1$ and when $q = 3$.

In Fig. 10 of this discussion, the writer has plotted in full lines results of the various tests as reported by the authors, the creep rates for the tube samples being corrected, as outlined by Professor Soderberg.

The writer has added his own interpretation of the results in dotted lines, again not claiming any greater validity but simply to show the breadth of interpretation possible.

From the two test results in tension at each temperature, it is possible to predict the tensile creep rate under the same loading as used in the tube, and thus to determine the ratio of the creep rate measured in the tube to the predicted rate in tension for comparison with the theory. Such a comparison is shown in Table 8 of this discussion.

TABLE 8 COMPARISON OF CREEP-RATE DATA, AS CALCULATED, WITH THEORETICAL

	Authors' interpretation (Fig. 10, full lines)	Discusser's interpretation (Fig. 10, dotted lines)
(900 F; 23,000 psi)		
Tension creep rate indicated by test	1.9	1.65
Slope of log-log plot, n	3.1	3.7
Theoretical ratio, tube/tensile creep	0.56	0.51
Expected tube creep rate	1.06	0.84
Measured-tube creep rate (corrected for thickness) thin	1.6	2.15
Measured-tube creep rate (corrected for thickness) thick	1.1	1.95
Actual ratio, tube/tensile creep thin	0.84	1.30
Actual ratio, tube/tensile creep thick	0.58	1.18
(1050 F; 9200 psi)		
Tension creep rate indicated by test	15.0	13.0
Slope of log-log plot, n	4.1	3.1
Theoretical ratio, tube/tensile creep	0.49	0.56
Expected tube creep rate	7.3	7.3
Measured-tube creep rate (corrected for thickness) thin	9.3	10.1
Measured-tube creep rate (corrected for thickness) thick	7.9	9.75
Actual ratio, tube/tensile creep thin	0.62	0.78
Actual ratio, tube/tensile creep thick	0.53	0.75

The theoretical ratios in Table 8 are given for the Soderberg theory. These ratios may be varied over a narrow range by assuming various combinations of m and q in the Bailey formulation, but they cannot possibly be made to account for the larger observed values even by making $m = 0$ and $q = n$.

All in all, it seems to the writer that materials being what they are, a too complicated theory of creep behavior is not warranted. We do need to have some notion of the relative strength and behavior of materials under compound stresses as compared with simple tension or simple shear, but we cannot forget that, even with the most careful laboratory technique, successive test bars are likely to differ very noticeably in creep behavior. We must, therefore, not lose sight of the fact that the main object of running these tests and developing a theory is to discover what is going on rather than to set up an exact science of calculation. Unfortunately, one cannot set up a theory in mathematical form without having it evaluated numerically. That is what a theory

[25] "The Interpretation of Creep Tests for Machine Design," by C. R. Soderberg. Trans. A.S.M.E., vol. 58, 1936, pp. 733–743.

is made for. Test results of the sort presented by Professor Norton are urgently needed and in larger volume. At the same time, analytical interpretation such as presented by Professor Soderberg is also necessary, and the fact that rather wide deviations from it are observed under test circumstances does not mean that we cannot make good use of it for understanding behavior and applying test results. For intelligent engineering design, we must have a theory and must also have a notion as to its limitations and be prepared to accept both.

Closure by C. R. Soderberg

Referring to the discussion by Mr. Clinedinst, it should be noted that the "condition of plasticity," as given by Equation [14] of Nádai,[10] is not directly applicable to the case of creep. This condition of plasticity postulates a constant value of the yield strength in tension. For the case of creep it is necessary to relate this yield strength to the creep rate.

An investigation of the influence of the tube thickness will be presented as a part of the concluding report of these tests. The results show that the influence may be even greater than that predicted by Mr. Clinedinst.

On the basis of these results, it is evident that the test pressures should be raised for the thick tubes, in order to obtain a close comparison with the tension tests. This would have the effect of raising the creep rates of the tubes to values about equal to the creep rates of the tension test at $\sqrt{3}/2$ times the tangential tests of the tubes. As shown by Messrs. van Duzer and McCutchan, however, the creep rates must be extrapolated to equal times to justify the application of the theory. The authors are indebted to Mr. Clinedinst for raising this question.

Following the presentation of this paper Dr. Jacobus kindly submitted to the author the data which he and his associates had used in the evaluation of stress limits for boiler tubes, based on their interpretation of the Bailey theory. Applying the analysis referred to in connection with Mr. Clinedinst's discussion, it was found that the limiting heat flows established by Dr. Jacobus involve an extra safety factor of about 1.15, which is due principally to the lower weight of the shearing stress assigned by the Mises-Hencky theory as compared with that of the maximum-shear theory. This is a difference on the safe side, which certainly may be ignored until our experimental knowledge is more precise. However, the author's analysis also shows that the intensity of stress at the inner bore is higher than that anticipated by Dr. Jacobus. The difference is about 20 per cent, and in the author's opinion, it should receive some attention in a future revision of the limiting heat flows, particularly in view of the danger of intercrystalline fractures.

The only aspect of Dr. Kanter's discussion which the author wishes to question is the fundamental one of treating the phenomenon of creep as a viscous flow with variable intrinsic viscosity. As a phenomenological description of the creep phenomenon it may be defended on a purely pragmatic basis, but the author finds it difficult to see any particular significance in the introduction of the flowability factor as defined by Equation [26] of Mr. Clinedinst's discussion. After all, this definition of viscosity has logical justification only for gases, although it may also be extended to liquids. When dealing with solids, however, the introduction of this term seems extremely arbitrary. If a sound phenomenological approach is to be made, it seems necessary to assume that a relation between stress, strain rate, strain, and time be made of the form

$$d\sigma = \phi d\dot{\epsilon}_p + \psi d\epsilon_p - \theta dt \qquad [38]$$

when the function ϕ has a meaning akin to Dr. Kanter's flowability function, but where the additional functions ψ and θ play an even more important role.[26] The present creep data do not afford sufficient information for the evaluation of these three functions, but this is due to the inadequacy of present testing methods.

This difference in outlook, however, does not invalidate Dr. Kanter's conclusions about the possible influence of the hydrostatic pressure. The author is prepared to agree that we are unable at the present time to state with assurance that the hydrostatic component of the stress does not influence the creep phenomena. The only statement which can be made is that the assumption of constant volume for the steady state of plastic flow seems to give results which check tolerably well with experiments. To go further than this requires an exposition of consistent divergencies between experiments and the present theory. The experimental results are far too uncertain to form the basis for such a conclusion at the present time, and no improvement of testing technique is in sight which would offer any immediate hope of such an exposition. If the volume does remain constant, however, and if the hydrostatic component of stress is without effect on the plastic flow, it is necessary that the tangential creep rate is $\sqrt{3}/2$ times the creep rate of the tensile test under the same stress intensity.

The author agrees fully with the statement made by Messrs. van Duzer and McCutchan that the creep rates for the tensile tests and the tubes should be compared at the same time intervals from the beginning of the test. As already mentioned by Professor Norton,[8] it was considered best to leave this to the final report. While a more complete discussion on this point must be deferred until the final report, it seems reasonable to conclude at the present time that the tangential creep rates of the tubes will be somewhat greater than those called for by the present theory. If the theory had demanded that the tangential creep rate of the tube be equal to that of the tensile test at equal stress intensity, all concerned would probably be satisfied that the theory had proved itself to be correct. The fact remains, however, that the theory demands that the creep rate of the tube be $\sqrt{3}/2$ times that of the tensile test, and the cause of the discrepancy must be searched for either in the premises of the theory or in the interpretation of the tests. It is quite possible that the premises of the theory are only approximately fulfilled. It is also possible that the final check of the dimensions of the tubes will require some correction of the final experimental data. It is questionable, however, whether the present tests will give results of sufficient accuracy to settle this point definitely.

In connection with Mr. Rhys's discussion the author would like to point out that the reference to the relative complexity of Bailey's theory[3] referred to the underlying reasoning rather than to the complication of the final formulas. In using the Bailey theory[3] it is necessary to make a decision of the values of m and n to be used. If $n - 2m = 1$ the two theories are identical. The only basis for assigning other values to these exponents would be the results of extensive series of tests of creep rates in two or three dimensions. Such tests are very scarce so far, and those available do not possess the precision required for a reliable estimate of these quantities. As far as the author knows, no convincing case has yet been made for this complication of the analysis.

Mr. Robinson shows that the present test results may be subject to a wide range of interpretations and that they cannot be made into a conclusive proof or disproof of the present theories of creep. This point is well taken, and the author would like to emphasize further the fact that present-day creep tests are altogether too inaccurate to be made a basis for a clinching proof of any theory. It must not be forgotten that the circular tube is a rather unfortunate choice for the investigation of creep in two

[26] "Plastic Flow and Creep in Polycrystalline Materials," by C. R. Soderberg, Proceedings of the Fifth International Congress for Applied Mechanics, Cambridge, Mass., 1938, pp. 238–244.

dimensions. After all, the problem is to establish the influence of the second principal stress when it is superimposed upon the first. In the tube the axial stress is only one half the tangential stress and the effect upon the tangential creep rate of the intensity of stress is reduced from unity in the tension test to $\sqrt{3}/2$ in the tube test. From the point of view of confirmation of a creep theory the thin sphere under internal pressure would be much better, because here the two principal stresses are alike and the result of a certain intensity of stress is reduced to $^1/_2$ as compared with a tension test. Moreover, if the main object of the present series of tests had been to afford a confirmation of the theory they certainly should have been prepared from a forged billet in the manner proposed originally by Mr. Robinson.

The author feels that the tests as made have practical justification but would caution against their being used rigidly to prove or disprove any theory. As long as the results are applied to tubes, it is relatively unimportant whether the results are projected against the intensity of stress, or against the tangential stress. This distinction becomes important, however, if the results are used to predict creep phenomena in other forms of stress application.

Closure by F. H. Norton

The discussion presented by Mr. Kanter brings out the fact, not clearly shown in the progress report, that in some cases the tubular specimen showed a small but definite longitudinal creep. In some cases this creep was an elongation and in other cases a contraction. The author quite agrees with Mr. Kanter that this effect is due to some directional effect left in the metal during the forming of the tube. This creep, however, is very small compared with the circumferential creep, not amounting in any case to more than a few per cent of the latter.

Mr. Van Duzer is quite correct in concluding that the tensile tests and the tubular tests should be compared at equal time intervals, and this will of course be done as nearly as possible in the final report. Some of the specimens which are now running are showing lower rates than they did when the data were presented in the progress report.

The data presented by Mr. Rhys showing the very gratifying agreement between his results and the results presented in the progress report are most interesting, and it is hoped that when the final values are presented this agreement will be equally good.

Mr. Robinson is undoubtedly correct in assuming that there are directional effects left in the tube material, but due to the fact that the tube was machined both inside and out and carefully annealed these should be small. The fact that the longitudinal creep was not always zero, as the theory would demand, would indicate that some of these effects were present, but to a very small degree. In regard to Mr. Robinson's point that various interpretations can be made from the curves, it should be stated that the gage length for the circumferential flow is only 4 in. and that the precision especially at the low rates, cannot be as great as for our usual 10-in. specimens. Therefore, we should not try to read into the low-rate curves too much exactness. Only by a considerable number of very long time tests can we hope to approach a final accurate value. The tests, however, should show the trend and, to a fair degree, should approximate values.

Effect of Grain Size and Structure on Carbon-Molybdenum Steel Pipe

An Investigation of These Properties for Their Bearing on Suitability of Such Pipe for High-Temperature Steam Service

By A. E. WHITE,[1] ANN ARBOR, MICH., and SABIN CROCKER,[2] DETROIT, MICH.

This paper gives the results of an investigation sponsored by The Detroit Edison Company at the University of Michigan for the purpose of relating the grain size and grain structure of carbon-molybdenum steel pipe to its high-temperature creep properties at 925 F. Pipe specimens were obtained from eight different heats of carbon-molybdenum steel, six being made by the open-hearth, and two by the electric-furnace process, and an attempt was made to correlate the respective creep strengths at 925 F with other significant properties obtained through short-time tests. At this stage of the investigation no apparent relationship was found. Further tests were conducted on heat-treated specimens from a single heat of steel which had been made by a preferred melting practice. From these tests, it is evident that, if the carbide structures are of a Widmanstätten type, with grain sizes ranging from 2 to 7, the steels will have high-temperature creep properties superior to those in which the carbides are in other conditions or forms. It is also believed that too great an emphasis has been placed on grain size and not enough on type of carbide structure. It is felt that consideration should be given to both factors, though with greater emphasis on the type of carbide structure.

AT THE present time, the generally accepted material for seamless pipe intended for steam service at 900 to 925 F is a 0.1 to 0.2 per cent carbon, 0.45 to 0.65 per cent molybdenum steel popularly known as carbon-moly. Pipe of this sort is commonly ordered to A.S.T.M. Specification A206.[3] Allowable working stresses (S values) for this material at various temperatures are established in the A.S.M.E. Boiler Construction Code and in the A.S.A. Code for Pressure Piping. These stresses are set with the expectation of a sustained useful life of the material of 100,000 hr or more, with a creep of something under 1 per cent. Fortunately, since corrosion is not a factor of major importance with superheated steam at 900 to 1000 F, design can be based on load-carrying ability rather than on corrosion resistance.

For many years, the very considerable opportunity for saving fuel through using higher operating temperatures has kept steam-power engineers and their metallurgical advisers in constant pursuit of better materials. In this sense a "better material" may be another material having properties superior to those possessed by metals already in use, or it may mean bringing out better properties in ostensibly the same material through improved steel-melting practice, heat-treatment, or the like. In either case, it is logical to strive for the optimum properties obtainable for any given material through properly directed metallurgical control.

Acceptance Criteria Needed

Owing to the comparatively recent application of carbon-molybdenum steel to superheated-steam pipe, much remains to be learned about how to melt and heat-treat steel of this composition in order to get best results in high-temperature service. While in the ultimate analysis all possible variations in the product should be weighed on the basis of performance in service, this takes years of time and more immediate criteria of expected behavior are badly needed. Hence, the interest in finding properties to serve as bases for short-time acceptance tests which can be linked with full-life performance.

Among the various possible short-time performance criteria for this and other materials heretofore considered by various investigators are: (*a*) yield point or proportional limit at operating temperature; (*b*) time-yield stress; (*c*) time versus stress to rupture; (*d*) relation of equicohesive temperature to service temperature; (*e*) austenitic or ferritic grain size; and (*f*) grain structure.

The great number of variables encountered in trying to effect metallurgical control, coupled with uncertainty as to which of these actually have much significance in determining high-temperature properties, have all but baffled rational analysis. To cite a single example of the difficulties encountered, specimens from practically identical analyses of carbon-molybdenum steel pipe with identical heat-treatment have shown creep rates[4] of the order of 10 to 1, refer to chart 1, Figs. 1 to 8. Obviously, chemical composition and heat-treatment alone cannot be the criterion of high-temperature behavior, and some other factor, such as steel-melting practice, with its attendant effect on grain size and structure, must be responsible for these discordant results. One clue to this enigma lies in the supposition that steel-melting practice and heat-treatment tend to establish grain size and structure or, in other words, that they constitute a cause with grain size and structure as effect. In this way the grain size and the structural characteristics of the grains should serve as thumbprint patterns to link the antecedents of a given steel with the behavior to be expected of it in service. Present knowledge of the subject is too meager to permit saying as yet whether grain size and structure are in themselves the cause of variation in high-temperature properties of materials, or whether they are merely symptoms indicative of proper antecedents or of other properties of a hidden

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[1] Consulting Engineer, The Detroit Edison Company, and Director of Engineering Research, University of Michigan. Mem. A.S.M.E.

[2] Senior Engineer, Engineering Division, The Detroit Edison Company. Mem. A.S.M.E.

[3] A.S.T.M. Tentative Specification A206-39-T, "Seamless Carbon-Molybdenum Alloy-Steel Pipe for Service at Temperatures From 750–1000 F," A.S.T.M. Standards for 1939, part 1, Metals, pp. 853–861.

Contributed by the Joint Research Committee on Effect of Temperature on the Properties of Metals and presented at the Annual Meeting, New York, N. Y., December 2–6, 1940, of The American Society of Mechanical Engineers.

[4] Whereas creep rates may be of the order of 10 to 1, corresponding creep strengths for 1 per cent in 100,000 hr probably would be more like 1 to 2, or 2 to 3.

CHART 1 EFFECT OF NORMALIZING AND DRAWING ON GRAIN SIZE AND STRUCTURE, AND CREEP PROPERTIES OF SEVERAL HEATS OF CARBON-MOLYBDENUM STEEL PIPE: HEATED 1 HR AT 1650 F, AIR-COOLED, AND DRAWN 1 HR AT 1200 F

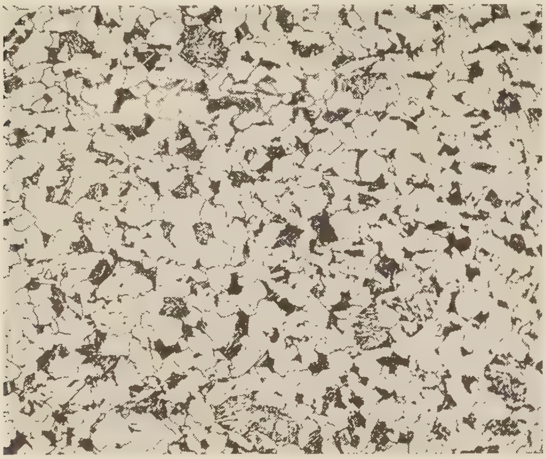


Fig. 1

(a)	4129	(e)	17400
(b)	5 to 7	(f)	1.2
(c)	Widmanstätten	(g)	×100
(d)	0.3		

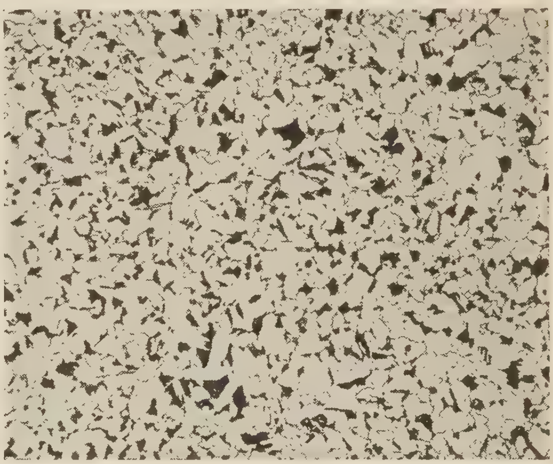


Fig. 2

(a)	6442	(e)	14500
(b)	7 to 8	(f)	1.8
(c)	Sorbitic pearlite	(g)	×100
(d)	1.1		

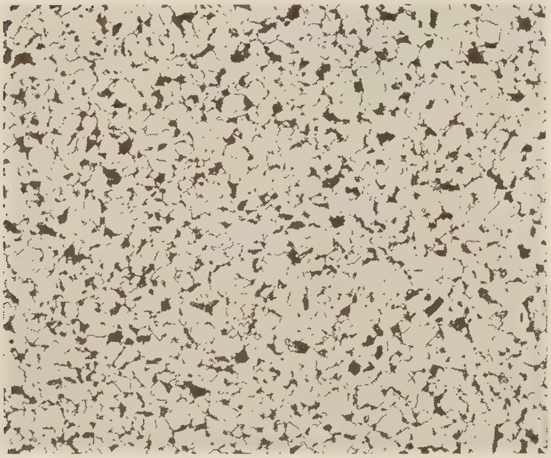


Fig. 3

(a)	11919	(e)	14300
(b)	7 to 8	(f)	2.0
(c)	Sorbitic pearlite	(g)	×100
(d)	1.2		

Fig. 4

(a)	2490	(e)	14100
(b)	7 to 8	(f)	2.25
(c)	Sorbitic pearlite	(g)	×100
(d)	1.4		

Note: (a) Heat; (b) A.S.T.M. grain size; (c) carbide structure; (d) creep rate, 15,000 lb stress at 925 F, per cent per 100,000 hr; (e) creep strength, psi at 925 F, 1 per cent per 100,000 hr; (f) aluminum added, lb per ton; (g) magnification.

but significant nature, as intimated in the 1940 report[5] of the A.S.M.E.-A.S.T.M. Joint Committee on the Effect of Temperature on the Properties of Metals.

In either event, grain size and structure should be a useful guide in appraising the probable behavior of the steels in service, and one which may be of value as an acceptance test in selecting material for high-temperature service. Not only is it important to know the significance of such factors as steel-melting and heat-treatment practice in advance, but from a purchaser's standpoint it is even more important to be able to judge from chemical analysis and metallographic examination whether the material offered will have the hoped-for high-temperature properties.

Evolution of Project

This investigation of carbon-molybdenum steel pipe, which was supported by The Detroit Edison Company, was conceived originally in the fall of 1936 as a means of finding a short-time acceptance test for the material which would be indicative of its probable behavior in service at 925 F. At the outset, pipe specimens were obtained from eight different heats of carbon-molybdenum steel, six being made by the open-hearth and two by the electric-furnace process, and an attempt was made to correlate their respective creep strengths at 925 F with other significant properties. In order to eliminate any variables arising from heat-treatment, the original specimens were all heat-treated according to the following requirement:

"Pipe shall be tested in the normalized and drawn condition. Normalizing shall consist of heating the finished pipe to 1650 F and cooling in still air to 800 F or below. Drawing shall consist of reheating the normalized pipe to 1200 F and allowing to cool slowly."

[5] Refer to p. 28 of report.

CHART 1 (*Continued*) EFFECT OF NORMALIZING AND DRAWING ON GRAIN SIZE AND STRUCTURE, AND CREEP PROPERTIES OF SEVERAL HEATS OF CARBON-MOLYBDENUM STEEL PIPE; HEATED 1 HR AT 1650 F, AIR-COOLED, AND DRAWN 1 HR AT 1200 F

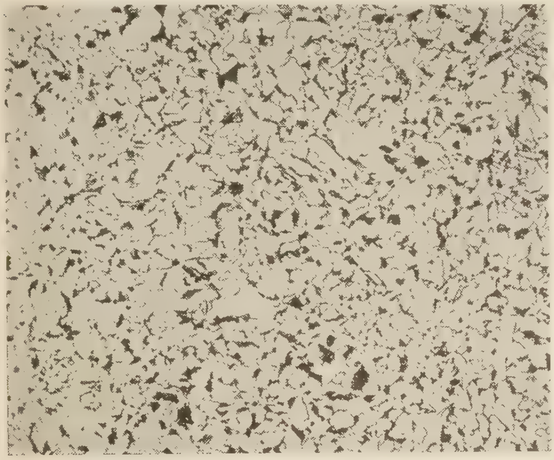


Fig. 5

(*a*) 11067
(*b*) 6 to 8
(*c*) Sorbitic pearlite
(*d*) 1.6
(*e*) 13900
(*f*) 2.0
(*g*) ×100

Fig. 6

(*a*) 43674
(*b*) 7 to 8
(*c*) Sorbitic pearlite
(*d*) 2.1
(*e*) 12800
(*f*) 1.0
(*g*) ×100

Fig. 7

(*a*) W.Y.F.
(*b*) 7 to 8
(*c*) Sorbitic pearlite
(*d*) 2.3
(*e*) 12500
(*f*) 2.25
(*g*) ×100

Fig. 8

(*a*) 4493
(*b*) 7 to 8
(*c*) Sorbitic pearlite
(*d*) 2.9
(*e*) 12000
(*f*) 2.25
(*g*) ×100

(Refer to note on opposite page for letter key.)

Although numerous tests were made to determine various room-temperature properties and short-time properties at 925 F, no significant relation was found at the time between these properties and the respective creep strengths of the different specimens. Neither was any particular superiority observed in the open-hearth process over the electric-furnace process, or vice versa. Among the possibilities studied was time-to-rupture under a stress calculated to produce failure within a day or two. Unfortunately, the required loading produced strain-hardening of the material at 925 F and the results were discordant. At this stage of the investigation, the possibility of finding a satisfactory short-time acceptance test was not promising.

Shortly thereafter, however, the authors' attention was called to an investigation[6] of the effect of grain size on creep strength being carried on in the laboratories of the General Electric Company. Arrangements were made to secure specimens from pipe (heat 7011 and T.C.) which had been given a heat-treatment conducive to producing the large grains which were thought to be associated with superior high-temperature properties. In order to produce the desired grain size of 3 to 6, rather drastic heat-treatment was found necessary, requiring holding the temperature for 2 to 3 hr at 1900 to 1975 F. At that time, it was held by some of the proponents of large grain size that the material should be full-annealed rather than normalized.

However, the creep strengths of the annealed specimens from heat 7011 and T.C., despite their large grain size, were decidedly inferior to the creep strengths of the other heats which were in a normalized and drawn condition.

The results confirmed the belief, already held by many, that grain size in itself was not the main contributing factor to the attainment of the desired high-temperature properties in a

[6] "Actual Grain Size Related to Creep Strength of Steels at Elevated Temperatures," by S. H. Weaver, Proceedings American Society for Testing Materials, vol. 38, 1938, part 2, pp. 176–181.

TABLE 1 CHEMICAL COMPOSITION OF DESIGNATED HEATS OF CARBON-MOLYBDENUM STEEL PIPE

Heat no.	Type of steel	Chemical composition, per cent: C	Mn	P	S	Si	Mo	Cr	Ni	Al	Al_2O_3	Al added, lb per ton
4299	OH[a]	0.14	0.43	0.010	0.016	0.110	0.48	..	....	...	...	0.50
4129	OH	0.15	0.41	0.014	0.014	0.19	0.58	0.07	0.14	0.011	0.008	1.20
4493	OH	0.16	0.45	0.009	0.021	0.14	0.58	0.06	Absent	0.043	0.010	2.25
6442	OH	0.17	0.52	0.011	0.018	0.16	0.54	0.05	Absent	0.045	0.010	1.80
11919	OH	0.15	0.47	0.013	0.017	0.13	0.55	0.02	Absent	0.020	0.010	2.00
2490	EF[b]	0.15	0.59	0.009	0.020	0.19	0.52	0.08	0.06	0.043	0.010	2.25
11067	EF	0.15	0.51	0.010	0.011	0.28	0.58	0.08	0.14	0.022	0.010	2.00
43674	OH	0.21	0.48	0.019	0.028	0.31	0.53	0.05	0.11	0.014	0.006	1.00
W.Y.F.	OH	0.13	0.50	...	...	0.13	0.52	0.06	Trace	0.042	0.008	2.25
7011	OH	0.16	0.47	0.014	0.020	0.14	0.45	..	....	...	...	1.50
TC	OH	0.17	..	...	...	0.15	0.56	..	....	...	...	..

[a] OH = open-hearth.
[b] EF = electric-furnace.

TABLE 2 PHYSICAL PROPERTIES AT ROOM TEMPERATURE OF DESIGNATED HEATS OF CARBON-MOLYBDENUM STEEL PIPE

Heat no.	Heat-treatment or condition	Tensile strength, psi	Yield stress, psi, 0.2 per cent set	Elongation, per cent in 2 in.	Reduction of area, per cent
4299	As-rolled	59250	33750	35.5	65.4
4299	A	55250	27875	36.7	61.5
4299	B	54700	27750	36.7	63.0
4299	C	65225	38125	33.5	67.6
4129	D	64900	40000	36.8	71.7
4493	E	59500	30000	36.3	67.9
4493	D	58500	35000	41.8	76.0
6442	D	60800	36250	40.0	74.1
11919	D	58900	35000	40.8	74.2
2490	D	61700	40000	38.5	76.8
11067	D	64400	40000	38.5	76.7
43674	D	71700	43250	35.5	69.3
W.Y.F.	D	58100	35000	39.8	76.2
7011	F	57250	25600	39.5	60.1
TC	G	54000	28000	41.5	67.0

(A) Heated to 1700 F and held for 2 hr to coarsen grain; furnace-cooled at 50 F per hr to below 1000 F.
(B) Water-quenched from 1800 F to simulate fabrication; heated to 1700 F and held for 2 hr to coarsen grain; furnace-cooled at 50 F per hr to below 1000 F.
(C) Water-quenched from 1800 F to simulate fabrication; heated to 1700 F and held for 2 hr to coarsen grain; air-cooled.
(D) Heated to 1650 F and held for 1 hr followed by air cooling; reheated to 1200 F and held for 1 hr, followed by air cooling.
(E) Hot-rolled followed by 1300 F process-anneal.
(F) Heated in range 1925 to 1975 F for 3 hr to coarsen grain; furnace-cooled.
(G) Heated in range 1900 to 1925 F for 2 hr to coarsen grain; furnace-cooled.

TABLE 3 PHYSICAL PROPERTIES AT 925 F OF DESIGNATED HEATS OF CARBON-MOLYBDENUM STEEL PIPE

Heat no.	Heat-treatment or condition	Tensile strength, psi	Yield stress, psi, 0.2 per cent set	Elongation, per cent in 2 in.	Reduction of area, per cent
4299	As-rolled	50000	26250	27.5	69.0
4129	D	56900	26250	28.8	73.4
4493	E	50750	25250	29.8	73.6
4493	D	51100	22000	34.3	82.8
6442	D	52100	23250	33.0	82.2
11919	D	52100	23000	33.5	83.2
2490	D	50900	23500	34.3	83.4
11067	D	49900	25500	33.8	79.2
43674	D	57000	28000	32.3	77.5
W.Y.F.	D	50700	21300	33.0	82.3
7011	F	51250	27000	31.0	78.3
TC	G	50800	23000	32.0	80.3

(D) Heated to 1650 F and held for 1 hr, followed by air cooling; reheated to 1200 F and held for 1 hr followed by air cooling.
(E) Hot-rolled, followed by 1300 F process-anneal.
(F) Heated in range 1925 to 1975 F for 3 hr to coarsen grain; furnace-cooled.
(G) Heated in range 1900 to 1925 F for 2 hr to coarsen grain; furnace-cooled.

given type of steel. In fact, the General Electric engineers, who advanced the theory on grain size, recognized at the time they advanced it that other factors were involved.

There is, however, a tie-in between grain size and steel-melting practice. An improperly killed steel on heating to temperatures around 1700 F will remain fine-grained, whereas, a properly killed steel will show a grain size of 3 to 7 if annealed or normalized from this temperature. The American Society for Testing Materials, recognizing this condition, called for a 3 to 6 grain size in its A206 specification[8] for pipe. The determination could be made either on pipe when in the as-rolled condition (on the assumption that the pipe would be worked down to about 1700 F) or on pipe when annealed or normalized from 1700 F.

This was a step forward, but by no means a final solution to the problem, for the results of a number of investigations have shown that superior high-temperature creep properties were obtained from steel in the "as-rolled" or normalized conditions rather than in the annealed condition.

A series of tests was made, therefore, on specimens from a single heat of steel, produced according to the preferred melting practice, embracing the various types of heat-treatment which it was felt would be of interest, viz. (*a*) as-rolled, (*b*) annealed, (*c*) annealed, preceded by a quench to simulate one of the possible conditions in fabrication practice, and (*d*) normalized, preceded by a quench to simulate one of the other possible conditions in fabrication practice.

The results of this work, coupled with the findings of earlier work by the authors bearing on the subject of grain size, grain structure, and high-temperature properties, as well as a high-temperature test on an all-weld section, are given in the succeeding sections of this paper.

Discussion of Results

As previously mentioned, the initial investigation was made on eight heats of carbon-molybdenum steel. To these were added two heats received through the courtesy of the General Electric Company, and later a heat through the courtesy of the National Tube Company which had been produced according to what is now believed to be the preferred melting practice.

Chemical Composition. The chemical compositions of these various heats are given in Table 1. These compositions, for the most part, are from drillings taken from actual sections of the pipe. Because of the general interest in aluminum additions in steel for high-temperature purposes, the amount of aluminum added in pounds per ton is also given in this table, as well as the amount of residual aluminum and aluminum oxide which was found in the original eight heats. All of the heats were of killed steel. It will be noted that the amount of aluminum added, expressed in pounds per ton, ranged from $^1/_2$ lb to 2.25 lb. Further reference to this matter will be made in a later section of the paper.

Physical Properties.[7] The physical properties of these steels at room temperature, with two heats showing different conditions of heat-treatment, are given in Table 2. The findings are without much significance. Based on the physical-property findings at room temperature, they show all the heats to be of an acceptable character.

Short-time tests at 925 F, Table 3, were also made on these heats. Heat 4493 was in two conditions of heat-treatment. These findings, again, are of but slight significance beyond showing that, on

[7] Though impact properties should be given consideration in the selection of materials for high-temperature properties, the subject matter was not included as the authors desired to confine the paper mainly to a treatment of the relation of grain size and structure to creep properties. It is hoped, however, that at the time of the presentation of the paper, the authors will have some information to contribute on this subject.

CHART 2 EFFECT OF HEAT-TREATMENT ON GRAIN SIZE AND STRUCTURE, AND CREEP STRENGTH ON TWO HEATS OF CARBON-MOLYBDENUM STEEL PIPE

Fig. 9

(a) 4299
(b) 5 to 7
(c) Widmanstätten
(d) 0.8
(e) 15250
(f) 0.50
(g) ×100

Heat-treatment
As-rolled

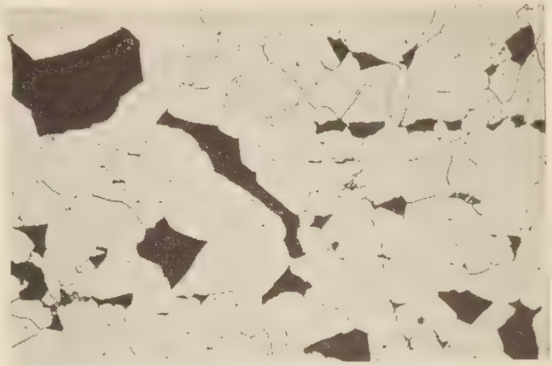


Fig. 10

(a) 4299
(b) 3 to 6
(c) Pearlitic
(d) 2.7
(e) 12100
(f) 0.50
(g) ×100

Heat-treatment
Heated 2 hr 1700 F
Cooled at rate of 50 F per hr to 1000 F
Air-cooled below 1000 F

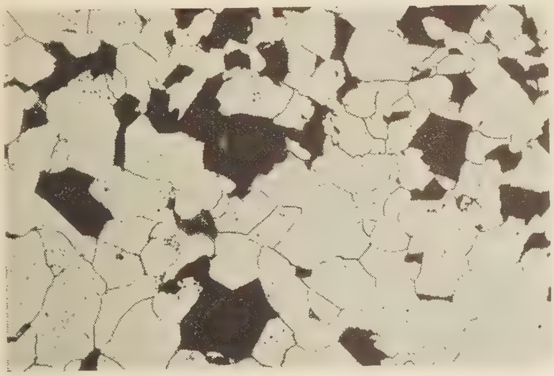


Fig. 11

(a) 4299
(b) 3 to 6
(c) Pearlitic
(d) 3.8
(e) 11250
(f) 0.50
(g) ×100

Heat-treatment
Heated 1 hr 1800 F
Water-quenched
Heated 2 hr 1700 F
Cooled at rate of 50 F per hr to 1000 F
Air-cooled below 1000 F

Fig. 12

(a) 4299
(b) 2 to 6
(c) Widmanstätten
(d) 0.6
(e) 16000
(f) 0.50
(g) ×100

Heat-treatment
Heated 1 hr 1800 F
Water-quenched
Heated 2 hr 1700 F
Air-cooled

Fig. 13

(a) 4493
(b) 4 to 6
(c) Widmanstätten
(d) 0.4
(e) 16700
(f) 2.25
(g) ×100

Heat-treatment
Hot-rolled
Process-anneal
1300 F

Fig 14

(a) 4493
(b) 7 to 8
(c) Sorbitic Pearlite
(d) 2.9
(e) 12000
(f) 2.25
(g) ×100

Heat-treatment
Heated 1 hr 1650 F
Air-cooled
Heated 1 hr 1200 F
Air-cooled

Note: (a) Heat; (b) A.S.T.M. grain size; (c) carbide structure; (d) creep rate, 15,000 lb stress at 925 F, per cent per 100,000 hr; (e) creep strength, psi at 925 F, 1 per cent per 100,000 hr; (f) aluminum added, lb per ton; (g) magnification.

CHART 3 RELATION BETWEEN GRAIN SIZE AND STRUCTURE AND CREEP STRENGTH OF SEVERAL HEATS OF CARBON-MOLYBDENUM STEEL PIPE WITH VARYING HEAT-TREATMENTS

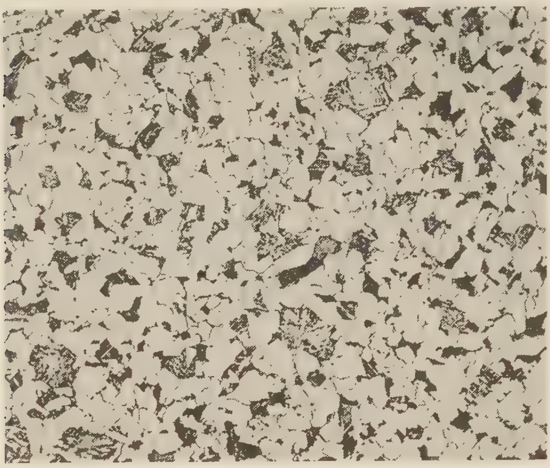


Fig. 15

		Heat-treatment
(a)	4129	Heated 1 hr 1650 F
(b)	5 to 7	Air-cooled
(c)	Widmanstätten	Heated 1 hr 1200 F
(d)	0.3	Air-cooled
(e)	17400	
(f)	1 20	
(g)	×100	

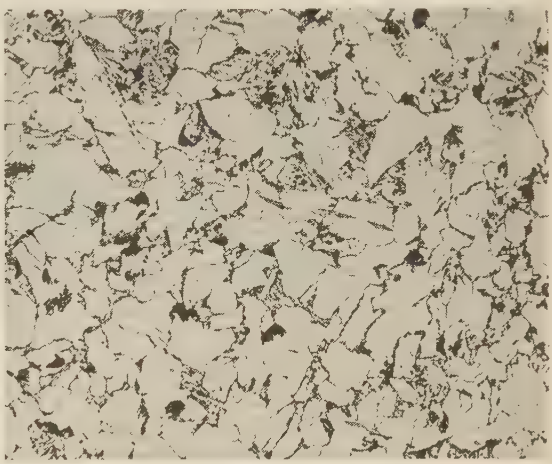


Fig. 16

		Heat-treatment
(a)	4493	Hot-rolled
(b)	4 to 6	Process-anneal
(c)	Widmanstätten	1300 F
(d)	0.4	
(e)	16700	
(f)	2.25	
( )	×100	

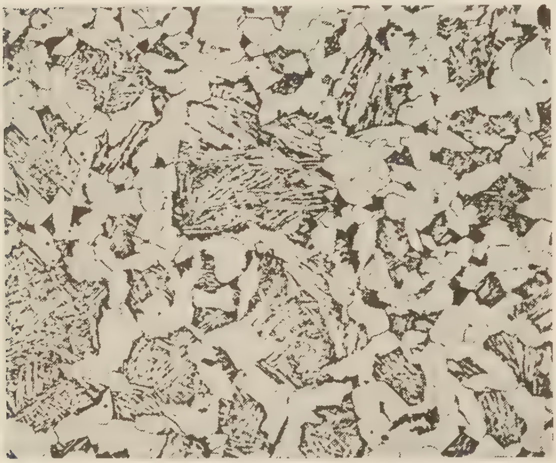


Fig. 17

		Heat-treatment
(a)	4299	Heated 1 hr 1800 F
(b)	2 to 6	Water-quenched
(c)	Widmanstätten	Heated 2 hr 1700 F
(d)	0.6	Air-cooled
(e)	16000	
(f)	0.50	
(g)	×100	

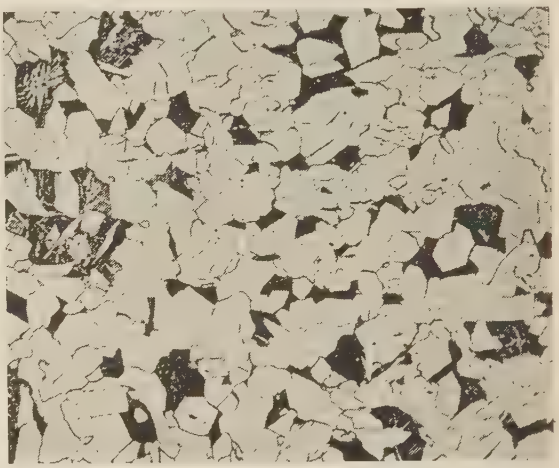


Fig. 18

		Heat-treatment
(a)	4299	Hot-rolled
(b)	5 to 7	
(c)	Widmanstätten	
(d)	0.8	
(e)	15250	
(f)	0.50	
(g)	×100	

Note: (a) Heat; (b) A.S.T.M. grain size; (c) carbide structure; (d) creep rate, 15,000 lb stress at 925 F, per cent per 100,000 hr; (e) creep strength, psi at 925 F, 1 per cent per 100,000 hr; (f) aluminum added, lb per ton; (g) magnification.

the basis of this test, the heats all appear to be of an acceptable character.

Grain Size, Grain Structure, and Creep Tests. The important findings of the investigation are given in Charts 1, 2, and 3, showing (a) the effect of normalizing and drawing on grain size and structure and creep properties of several heats of carbon-molybdenum steel pipe; (b) the effect of heat-treatment on grain size and structure and creep strength on two heats of carbon-molybdenum steel pipe; and (c) a summary chart showing the relation between grain size and structure and creep strength of carbon-molybdenum steel pipe with varying heat-treatments.

The A.S.T.M. grain-size numbers listed on the charts refer to the size of ferrite grains, pearlite patches, or Widmanstätten areas seen at a manification of 100 diameters in the materials as used.

Normalized and Drawn. In the planning of the initial work, it appeared desirable to have the steel from the various heats all given the same heat-treatment. Therefore, these steels were heated for 1 hr at 1650 F, air-cooled, and drawn 1 hr at 1200 F. This is a normalizing-and-drawing operation. This type of treatment was selected because, as a result of previous work, it was felt that it would give the most satisfactory high-temperature properties. The results are shown in Chart 1. It presents

CHART 3 (*Continued*) RELATION BETWEEN GRAIN SIZE AND STRUCTURE AND CREEP STRENGTH OF SEVERAL HEATS OF CARBON-MOLYBDENUM PIPE WITH VARYING HEAT-TREATMENTS

FIG. 19

(a) 6442
(b) 7 to 8
(c) Sorbitic pearlite
(d) 1.1
(e) 14500
(f) 1.80
(g) ×100

Heat-treatment
Heated 1 hr 1650 F
Air-cooled
Heated 1 hr 1200 F
Air-cooled

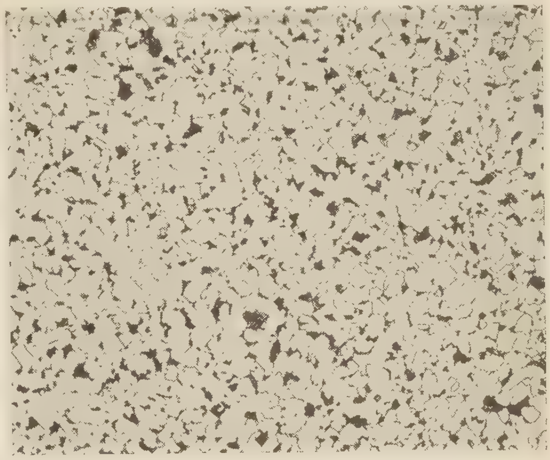


FIG. 20

(a) 4493
(b) 7 to 8
(c) Sorbitic pearlite
(d) 2.9
(e) 12000
(f) 2.25
(g) ×100

Heat-treatment
Heated 1 hr 1650 F
Air-cooled
Heated 1 hr 1200 F
Air-cooled

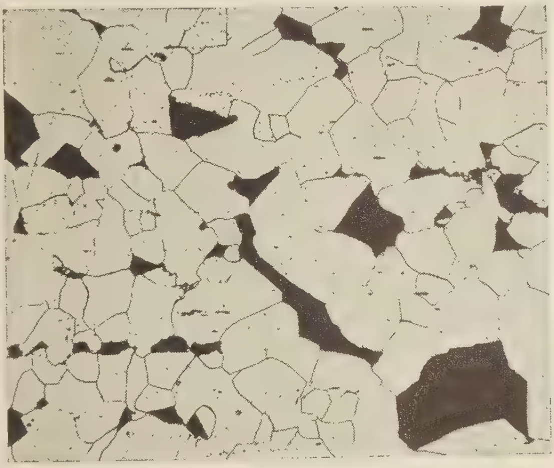


FIG. 21

(a) 4299
(b) 3 to 6
(c) Pearlitic
(d) 2.7
(e) 12100
(f) 0.50
(g) ×100

Heat-treatment
Heated 2 hr 1700 F
Cooled at rate of 50 F per hr to 1000 F
Air-cooled below 1000 F

FIG. 22

(a) 7011
(b) 3 to 6
(c) Pearlitic
(d) 7.3
(e) 9800
(f) 1.50
(g) ×100

Heat-treatment
Heated 3 hr 1925 to 1975 F
Furnace-cooled

(Refer to note on opposite page for letter key.)

not only the resultant structures, but also data as to ferritic grain size, carbide structure, creep rate, creep strength, and the amount of aluminum added expressed in pounds per ton. In only one instance was a true Widmanstätten type of structure for the carbide found. That is shown in Fig. 1, taken from heat 4129. This specimen showed not only the best creep properties for all those given in this chart, but also for all of those in the investigation as a whole. Carbide structures of all of the other heats were of the sorbitic-pearlitic type, with only a suggestion of a Widmanstätten type of carbide structure in one, namely, heat 43674, shown in Fig. 6. This specimen, however, did not have as good creep properties as some of those which were completely free from even a suggestion of a Widmanstätten type of structure.

Because of a suggestion which has been advanced that the amount of aluminum is a controlling factor in the attainment of superior high-temperature properties (it being intimated that, if the amount of aluminum added is under 2 lb per ton, a heat of steel will have good high-temperature properties, whereas, the reverse would be expected if it was in excess of that amount), the amount of the aluminum added is given in the chart. At first glance, the hypothesis with respect to the aluminum does not appear to be substantiated. Heat 43674, Fig. 6, was made with but 1 lb per ton of aluminum added; yet it is inferior, from the standpoint of high-temperature properties to other heats in which

the amount of aluminum added was greater than 1 lb per ton and in one case as much as 2.25 lb per ton. Although it is possible that the amount of aluminum added per ton may be an influencing factor, these results do not appear to bear out this hypothesis, at least within the limits of the amounts of aluminum found in these heats.

Only one of the heats investigated was so made as to develop a 5 to 7 grain size when normalized from 1650 F. This is heat 4129. Its carbide structure is Widmanstätten and it shows the best high-temperature properties of all the heats given in this chart. It gives an indication, which is supported in the succeeding charts, that, with a grain size ranging from 3 to 6 and with a steel so made that when normalized from 1650 to 1700 F a Widmanstätten structure results, good high-temperature creep properties may be expected.

Varying Heat-Treatments on Two Heats. For the purpose of throwing further light on the influence of grain size and structure resulting from heat-treatment on high-temperature properties, the results of an investigation on two heats of steel with varying conditions of heat-treatment are presented in Chart 2. The results given in Figs. 9, 10, 11, and 12 are all from heat 4299, which was a heat produced by the National Tube Company under conditions which they believed would result in the best possible high-temperature properties. It will be noted in looking at the chart that this heat of steel, when in the as-rolled or normalized conditions, Figs. 9 and 12, shows grain sizes ranging from 2 to 7 with the carbide structures of the Widmanstätten type. The high-temperature creep properties are good, with creep rates under the given conditions of 0.8 and 0.6 per cent per 100,000 hr, respectively. The creep strengths are equally good, with values of 15,250 and 16,000 psi, respectively.

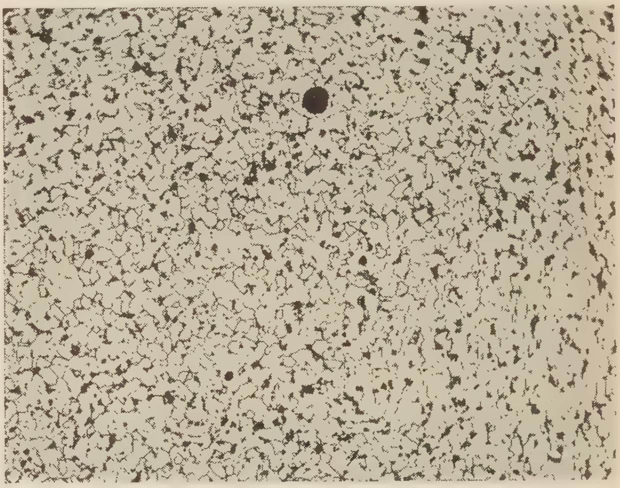
On the other hand, when specimens from this same heat of steel were annealed, in one case without a preliminary quench (Fig. 10) and in the other case following a quench (Fig. 11), the high-temperature creep properties of the specimens were adversely affected. That is, instead of the creep rates being 0.8 and 0.6 per cent, they were 2.7 and 3.8 per cent, and, whereas, the creep strengths, as stated in the preceding paragraph, were 15,250 and 16,000 psi, they dropped to 12,000 and 11,250 psi.

These differences cannot be laid to grain size, as the grain size for all of the specimens, with the possible exception of that shown in Fig. 9, was of the same order. Nor can the differences be laid to steel-melting practice, as these various specimens were all taken from the same heat. The one outstanding variation is that of carbide structure. In the first instance the carbide structures are of the Widmanstätten type, and in the other case they are of the pearlitic type.

Two different structures and the high-temperature properties resulting therefrom are shown for heat 4493 in Figs. 13 and 14 in Chart 2. Fig. 13 shows the specimen in an as-rolled condition followed by a 1300 F process anneal and Fig. 14 shows the specimen after it was normalized from 1650 F followed by a draw at 1200 F. The structure in the process-annealed condition is of the Widmanstätten type, with a grain size ranging from 4 to 6. Good high-temperature creep properties were obtained. In the case of the specimens normalized from 1650 F followed by a draw at 1200 F, the structure is of the sorbitic-pearlitic type, with a small grain size. Its resulting high-temperature creep properties are inferior to those found in the process-annealed specimen. The fact, however, that good creep properties were obtained in the pipe in the process-annealed condition without an abnormally large grain size leads the authors to advance the hypothesis that, if the normalizing temperature had been enough higher to give the preferred type of grain size and carbide structure, the creep properties of the pipe thus normalized would have more nearly approached those found in the pipe when in the process-annealed condition.

The amount of the aluminum addition throws no light on the findings, for, in spite of the fact that more than 2 lb per ton of aluminum were added, it did not appear adversely to affect the attainment of good high-temperature creep properties for the process-annealed specimen.

In reviewing the findings from the data given in this chart, all those specimens which showed superior high-temperature creep properties had a Widmanstätten type of carbide structure. They also showed moderately large grain size. Inferior creep properties were found in all those specimens in which the carbide structure was of the sorbitic-pearlitic or pearlitic type, irrespective of grain size.



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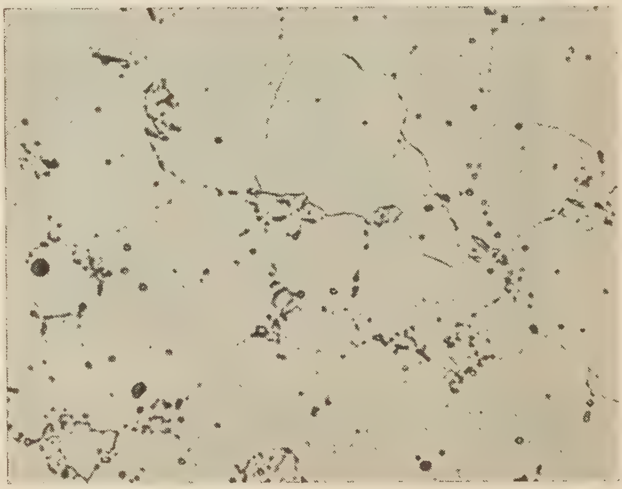

×1000

Fig. 23 Microstructures of Welded Section of 0.50 Carbon-Molybdenum Steel Pipe

(Heat 4299; weld deposited metal section; stress-relieved at 1200 F for 2 hr.)

Summary Data. A general summary of the conditions showing the relation between grain size and structure and creep strength on several heats of carbon-molybdenum steel pipe with varying heat-treatments is given in Chart 3. The high-temperature creep properties of the specimens shown in Figs. 15 to 18, inclusive, are all of a most acceptable character. The structures of the specimens are all of the Widmanstätten type, with grain sizes ranging from 2 to 7.

On the other hand, the high-temperature creep properties of the specimens shown in Figs. 19 to 22, inclusive, are inferior to those just mentioned. In the case of the specimen shown in Fig. 22, the high-temperature creep properties are decidedly poor, although the grain size is large. The grain size is equally large in the case of the specimen shown in Fig. 21, although its high-temperature creep properties are superior to those found in the specimen shown in Fig. 22. These two specimens are from different heats. It is quite possible that the difference in the high-temperature creep properties, even though the grain size and the carbide structures are substantially the same, is due to the steelmaking practice. Heats 6442 and 4493, shown in Figs. 19 and 20, were of the sorbitic-pearlitic type, with grain size ranging from 7 to 8. They have both been normalized from 1650 F. The grain size and structures did not respond under this treatment to give a 3 to 6 grain size or a Widmanstätten type of structure. Failure to do so is probably due to the type of steelmaking practice used, with the result that superior high-temperature creep properties were not obtained.

It would appear again, on the basis of the data presented in this chart, coupled with the findings shown in Charts 1 and 2, that whenever the steelmaking practice is of such a type, accompanied by suitable heat-treatment, as to cause the carbides to assume a Widmanstätten pattern, with a grain size ranging from 3 to 7, superior high-temperature creep properties may be expected. Especially is this believed to be the case if the steelmaking practice is such that these grain sizes and types of carbide structures develop when the heats of steel under examination are normalized from a temperature of approximately 1700 F.

All-Weld Metal Test

Recognizing that a very considerable quantity of the carbon-molybdenum pipe used for high temperatures is assembled into pipe lines by welding, it appeared desirable to determine the high-temperature properties of an all-weld section. Two segments of pipe, therefore, from heat 4299, were welded together longitudinally so as to afford sufficient weld material for an all-weld specimen. Welding was done by the electric-arc process, using a carbon-moly rod. Preceding welding, the pipe was preheated at 450 to 600 F. After welding, it was stress-relieved for 2 hr at 1200 F. Specimens were taken from all-weld metal sections and tested at room temperature and a creep test was made at 925 F under a stress of 15,000 psi.

The results of the room-temperature tensile tests were as follows:

Tensile strength, psi	66650
Yield strength, psi, 0.2 per cent set	43000
Elongation, per cent in 2 in	32.5
Reduction of area, per cent	65

These results show that the material possessed acceptable physical properties at room temperature. It was stronger, though slightly less ductile, than the parent metal.

The creep-test result was surprisingly good, when it is realized that all of the stock in the specimen was made up entirely of all-weld metal. It showed the following values under a stress of 15,000 psi at 925 F:

Creep rate, per cent per 1000 hr	0.019
Creep rate, per cent per 100,000 hr	1.9
Estimated stress for 1 per cent per 100,000 hr	13,250 psi

The grain size and structure of the weld metal at 100 and 1000 diam is given in Fig. 23. It shows the metal to be fine-grained, with marked evidence of spheroidization of the carbides.

These results are both interesting and gratifying. Yet it must be recognized that these are the results of but one test of a weld made under almost ideal conditions. It would be unwise to attach too great significance to the findings until numerous check tests have been made with a variety of welding procedures.

Conclusions

So far as carbon-molybdenum steel pipe is concerned, it is evident, on the basis of this investigation, that, if the carbide structures are of a Widmanstätten type with grain sizes ranging from 2 to 7, the pipe will have high-temperature creep properties superior to those in which the carbides exist in other conditions or forms.

The steel must be properly killed to obtain the desired structure. A method to determine this is to normalize from around 1700 F. If the carbides assume a Widmanstätten type of structure with grain sizes ranging from 2 to 7, then it may be assumed the steel has been properly killed.

It is also believed that too great emphasis has been placed on grain size and not enough on type of carbide structure. It is felt that consideration should be given to both factors, though with the greater emphasis on the type of carbide structure.

The results of the high-temperature tests on the all-weld specimen are both interesting and encouraging, for, in spite of the fact that the specimen under test was made up entirely of deposited metal, it possessed satisfactory high-temperature properties.

Acknowledgment

The authors wish to acknowledge their indebtedness to Mr. Waldo M. McKee of the M. W. Kellogg Company, to Messrs. T. S. Fuller, E. L. Robinson, and S. H. Weaver of the General Electric Company, to Mr. E. C. Wright of the National Tube Company, and to Mr. W. G. Hildorf of The Timken Roller Bearing Company, for contributing valuable suggestions in arranging the tests, or for furnishing specimens of carbon-molybdenum pipe from various heats made by different melting practices. Thanks also are due to Dr. J. W. Freeman of the University of Michigan staff and to Dr. C. L. Clark, while a member of the staff, for their painstaking work in conducting the tests and recording the results.

Discussion

R. W. Emerson[8] and G. Sinding-Larsen.[9] The authors demonstrate quite conclusively that, in regard to high-temperature properties, the structure of carbide grain is equally important to that of structural grain size. This fact should be beneficial to piping fabricators and others who do not have facilities for creep testing but do have metallographic equipment for checking the final grain size and carbide structure and who may, by this method, pass judgment on the probable creep characteristics of the steel by comparison of the grain structure with those shown by the authors.

It seems that no definite conclusions have been reached with respect to the direct effect of actual aluminum additions on the creep characteristics of carbon-molybdenum steel, A.S.T.M. Specification A-206.

It cannot be overemphasized that the two most important facts brought out by the authors, namely, structural grain size and carbide structure, upon which good creep properties in carbon-molybdenum steel appear to be dependent, are in turn dependent upon maximum heat-treating temperature and rate of cooling from that temperature. However, the maximum temperature and cooling rate, to produce a given grain size and

[8] Metallurgist, Pittsburgh Piping & Equipment Company, Pittsburgh, Pa.

[9] Chief Engineer, Pittsburgh Piping & Equipment Company, Pittsburgh, Pa.

carbide structure, are to a great extent dependent upon the amount of aluminum added to the steel. The carbide structure formed by cooling, at a given rate from the maximum heat-treating temperature, is dependent upon the austenite grain size produced by heating to that temperature.

For example, upon heating a carbon-molybdenum steel, containing $^1/_2$ lb of aluminum per ton, to 1700–1750 F, an austenite grain size should be established which will normally produce upon air-cooling a Widmanstätten carbide-ferrite grain size of

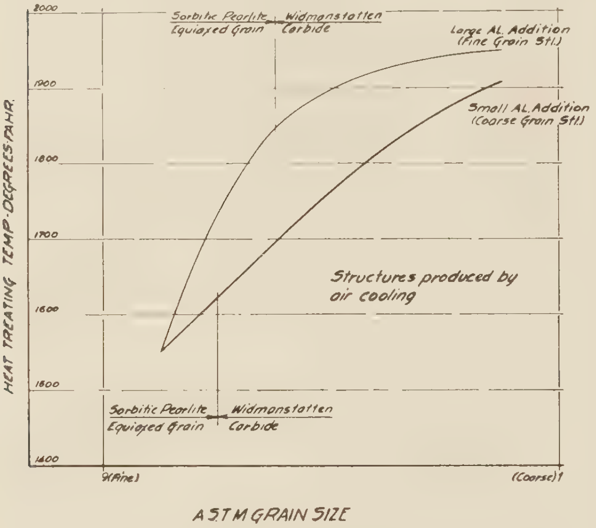

FIG. 24 VARIATION IN GRAIN SIZE AND CARBIDE STRUCTURE AS A FUNCTION OF THE HEAT-TREATMENT TEMPERATURE
(Coarse- and fine-grain carbon-molybdenum steel, A.S.T.M. Specification A-206.)

the desired A.S.T.M. 3 to 6. Air cooling is sufficiently rapid so that complete rejection of the ferrite to the austenite grain boundaries is impossible while cooling through the transformation range of the steel, resulting in somewhat of an acicular carbide grain structure[10] (Widmanstätten), as is shown in the large grains of Fig. 12 of the paper.

In contrast to this, a carbon-molybdenum steel, containing 2.25 lb of aluminum or more per ton, when heated to 1700–1750 F would establish a fine grain size (A.S.T.M. 7 to 8). Since the austenite grain size is small, the distance from center to edge of the austenite grain is also small and, for the same cooling rate as mentioned, sufficient time elapses for essentially complete rejection of the proeutectoid ferrite, with the result of a sorbitic pearlite and an equiaxed grain structure. Thus, aluminum is believed by the writers to have an indirect effect on the creep characteristics of carbon-molybdenum steel by affecting the grain size and carbide structure of the material for a given heat-treating cycle. This is substantiated by Figs. 1 to 8 of the paper, in which Figs. 1 and 6 show evidence of a Widmanstätten structure though Figs. 2, 3, 4, 5, 7, and 8 show no traces of such. The effect of aluminum on grain size, carbide structure, and heat-treating temperature is shown schematically in Fig. 24 of this discussion.

Fig. 24 shows that a Widmanstätten carbide can be obtained at a slightly smaller grain size in a steel with a small quantity of aluminum added than with one containing a larger quantity of aluminum. It also shows that carbon-molybdenum steel, containing a large quantity of aluminum, can be made to coarsen to the desired 3- to 6-grain size and produce a Widmanstätten-carbide structure by the use of excessively high heat-treating temperatures (1850–1950 F). From the customer's standpoint, however, it is desirable to get the required grain structure at the lowest possible heat-treating temperature, since oxidation (scaling) may possibly encroach on the minimum wall thickness of the pipe when excessive heat-treating temperatures are used.

P. A. SALMON.[11] This interesting paper makes no reference to uniformity of grain size which in the past has been considered as an important factor in determining the acceptability of carbon-molybdenum steel pipe for high-temperature service. Some of the photomicrographs illustrated in the paper show a pronounced duplex type of structure. Some comments by the authors on this phase of the subject would be of interest.

It is hoped that information concerning the impact value of steel showing the Widmanstätten structure will be given in the authors' closure.

S. H. WEAVER.[12] In their present contribution on grain size and structure in carbon-molybdenum steels, the writer agrees with the general principles and conclusions of the authors' but would like to express a somewhat different viewpoint on some of the details presented.

E. R. Parker,[13] in a recent paper, states that increased creep strength in steels at elevated temperatures is obtained by two general methods: (*a*) By the use of solid-solution alloys, as small percentages of molybdenum and tungsten, where a crystal-boundary effect necessitates adjusting the grain size to an optimum value; (*b*) by the use of precipitation-hardening alloys where size, shape, and dispersion of the precipitated phase are much more important than the grain size.

The writer, in a current article,[14] presented a series of long-time tests in support of Parker's first type of creep strength and outlined the grain-size creep-strength temperature characteristic for carbon-molybdenum steel. In those various special tests, the object was to hold the structure constant and isolate the boundary or grain-size effect. At no time has any one in our creep-test organization overlooked the importance of the structure. The early creep tests at 750 and 840 F were always run with three treatments, oil quench, normalize, and anneal. These and later tests indicate that in the low-alloyed molybdenum steels at 750 F, the oil-quench treatment is the strongest in creep; at about 850 F, the oil-quench and normalized structures are of equal creep strength; for increasingly higher temperatures, the normalized treatment leads, until near 1000 F the normalized and annealed structures become of equal creep strength for long-time, low-creep-rate tests.

Recognizing the greater creep strength at 925 F for the normalized or Widmanstätten structure over the annealed or sorbitic-pearlite structure, another factor enters the problem in the decrease in creep strength with time at temperature due to carbide spheroidization. The normalized carbon-molybdenum pipe steel becomes spheroidized in one third of the time required for the annealed steel to attain a similar condition, when both are without stress. There is evidence that the time period is de-

[10] "The Grain Size of Steel," by J. R. Vilella, *Mechanical Engineering*, vol. 62, April, 1940, p. 302; par. 7 gives a detailed description of these phenomena.

[11] Assistant Engineer, Public Service Electric and Gas Company, Newark, N. J. Mem. A.S.M.E.

[12] Turbine Engineering Department, General Electric Company, Schenectady, N. Y.

[13] "The Development of Alloys for Use at Temperatures Above 1000 F," by E. R. Parker, Trans. American Society of Metals, vol. 28, Dec., 1940, p. 797.

[14] "Relation of Grain Size to Creep Strength of Carbon-Molybdenum Steel," by S. H. Weaver, *General Electric Review*, vol. 43, no. 9, Sept., 1940, pp. 357–364.

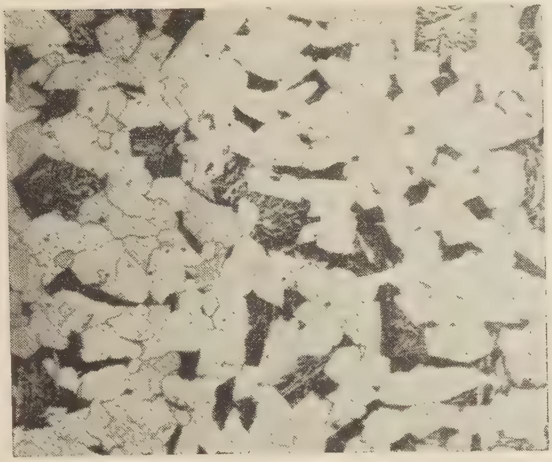

Fig. 25 Grain Size and Structure as Rolled
(Air-cooled from rolls, then drawn at 1200 F, 1 hr, furnace-cooled. Creep stress 18,700 psi at 932 F, for creep rate of 1 per cent per 100,000 hr; ×100.)

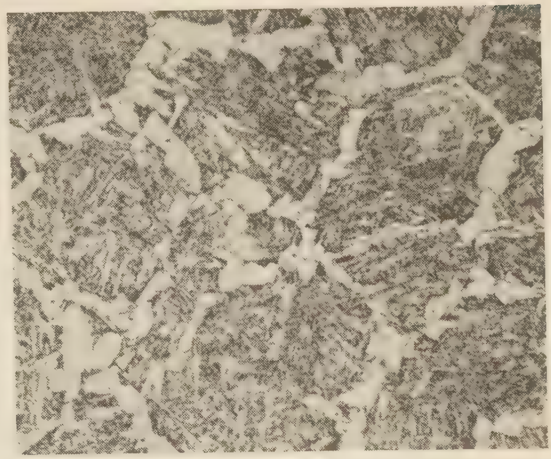

Fig. 26 Same Material as Fig. 25 After Additional Heating
(After 1800 F, 1 hr, cooled by water spray and *not drawn*. Creep stress 23,000 psi at 932 F for creep rate of 1 per cent per 100,000 hr; ×100.)

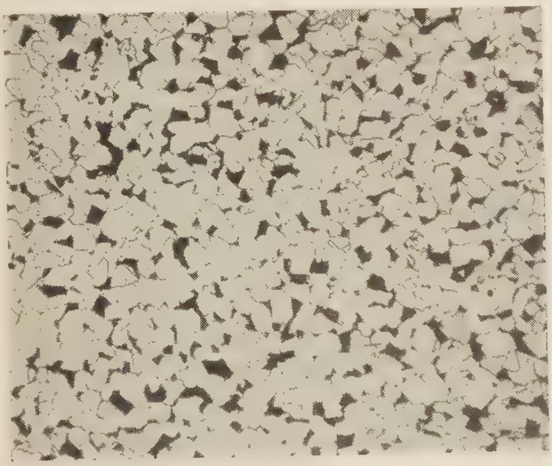

Fig. 27 Same Material as Fig. 25, Air-Cooled Twice Through Critical Temperature Then Given Additional Heating
(Heated at 1920 F, 15 min, air-cooled, and 1200 F, 1 hr, furnace-cooled. Creep stress 11,000 psi at 932 F for creep rate of 1 per cent per 100,000 hr; ×100.)

Fig. 28 Grain Size and Structure of Two Samples Resulting From Similar Pretreatment as Fig. 27, but With a Different Annealing and Cooling Cycle
(One sample annealed at 1650 F, 8 hr, furnace-cooled, and other at 1950 F, 4 hr, furnace-cooled. Creep stress 8200, and 8000 psi, respectively, at 932 F for creep rate of 1 per cent per 100,000 hr; ×100.)

creased by stress. Some authorities contend that the greater initial strength of the normalized structure compensates for its more rapid deterioration. Until more data are obtained, it is believed conservative practice to design for the annealed, more stable steel for 925 F and higher service temperatures. Prof. White was consulted on several occasions upon the spheroidization question and the problem is mentioned here only to indicate why, at the present time, an annealed treatment for the higher service temperatures is preferred.

Tests on material from the same pipe length demonstrate the effects of structure, grain size, mixed grains, and aluminum killing of the steel. A $6^5/_8$-in-outer-diam, 0.56-in-wall carbon-molybdenum pipe, manufactured before the present grain-size requirement was placed in A.S.T.M. Specification A-206, was selected from stock for special tests. The chemical analysis was carbon 0.14, manganese 0.44, silicon 0.16, and molybdenum 0.46; open-hearth steel killed with 2.25 lb per ton of aluminum. Fig. 25 of this discussion represents grain size and structure as rolled. Fig. 26 is the same structure after heating to 1800 F for 1 hr, cooled by water spray, and not drawn. Creep strengths at 932 F, 7 deg higher than the authors' tests, are 18,700 and 23,000 psi, respectively. Fig. 27 began with the structure in Fig. 25, heated and cooled twice through the critical temperature and then heated to 1920 F for 15 min, air-cooled, and followed by 1200 F, 1 hr, furnace cool; creep strength 11,000 psi at 932 F. Fig. 28 had a similar pretreatment as Fig. 27, then annealed at 1650 F for 8 hr. A second sample also represented by Fig. 28 has the same cycle except annealed at 1950 F for 4 hr. The creep strengths at 932 F were 8200 and 8000 psi, respectively. Figs. 25, 27, and 28 show the combined effect of grain size and structure upon the creep strength, for there is only a trace of the Widmanstätten formation in Fig. 27. Fig. 26, due to the absence of the draw, gave a decreasing creep strength for a creep rate of 1 per cent per 100,000 hr, the longer the test was continued.

The separation of the effects of grain size and structure requires the reproduction of the grain size in Fig. 25 of this discussion, in an annealed structure in the same steel. This was impractical as shown by curve I for grain growth in Fig. 29 of this discussion. The uniform grain size in Figs. 27 and 28 held constant for 4 hr;

then for longer periods, greatly enlarged grains slowly appeared in isolated regions, producing a mixed grain size; that is, a mixture of large and small grains. At least 12 hr at temperature is required before an enlarged, uniform grain size is obtained. Other holding temperatures would produce different curves and size of coarsened grains.

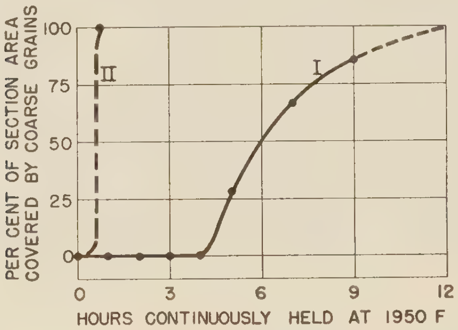

Fig. 29 Extent of Grain Growth With Time at Temperature

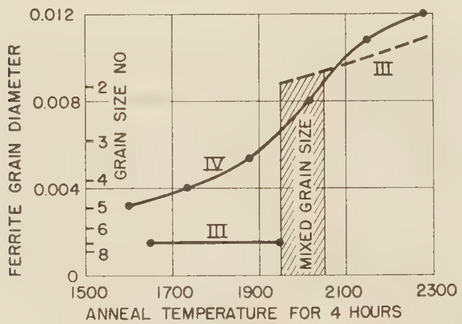

Fig. 30 Grain Size and Heat-Treating Temperature in Steel

Curve III in Fig. 30 plots uniform ferrite grain size against annealing temperature held for 4 hr. The aluminum-killed pipe steel has a uniform No. 7 grain size up to 1950 F, then enters the shaded area which is a "grain-coarsening temperature range" of about 100 to 150 F. This shaded area shifts temperature location which must be found by trial test of each steel melt. Treatment in this temperature range produces a mixed grain size unless held at temperature for very long periods, which finally results in a uniform, very coarse grain size, indicated by curve I, in Fig. 29, and the dashed top of curve III, Fig. 30. There is no choice in uniform grain size; only very fine or very coarse grains of uniform size can be obtained. When a mixed grain size is once established in a steel it is difficult to eliminate by the usual subsequent heat-treatments.

For comparison, another steel was tested with that in curve I. This steel was from a 4-in-diam pipe, of the same specification as that mentioned, where the billet manufacturer stated that the steel was open-hearth, not killed with aluminum. It coarsened from an initial No. 6 to No. 2 uniform ferrite grain size within 1 hr, as in curve II, Fig. 29. While a grain-size-temperature curve was not made on this steel, Fig. 30, curve IV is representative of curves on electric-furnace and open-hearth low-alloy steels, fully deoxidized and without a grain-growth inhibitor. Data for similar curves upon differently alloyed steels are given in a previous article.[14] Such steels acquire a definite uniform grain size in a short time for each increased treating temperature. There are many curves, intermediate between curves III and IV, depending upon the deoxidization details in steel manufacture.

The grain size in Fig. 25, with a pearlite structure, could not be obtained by isothermal transformations or by different pretreatment cycles, or the treatment proposed by Dorn and Harder.[15] The resulting grains were always small or of mixed size and creep tests were omitted.

The authors have not sufficiently emphasized the ill effects of mixed grain size upon the creep strength. A definite value cannot be assigned to the loss in creep strength, as the effect probably varies with the range from larger to smaller grain size and the proportion of section area each size occupies in a given steel. The author's Figs. 9 to 12 have mixed grain size while the writer's Figs. 25 to 28 are each of uniform grain size. The validity of the test results themselves is not in question because the writer's laboratory had previously exchanged bars with the authors and obtained identical creep rates for the same loadings. Fig. 9 of the paper, with the mixed grain size, has a creep strength of 15,250 psi at 925 F; Fig. 25, with a uniform structure and the same size of ferrite grain, tests 18,700 psi at 932 F. Fig. 12 with an extremely mixed grain size had nearly the same treatment as the uniform structure in Fig. 26. Fig. 12 tested 16,000 psi at 925 F, while Fig. 26 gave 23,000 psi at 932 F. A correct comparison of the annealed structures is difficult. Figs. 10 and 11 of the paper both show mixed sizes with an oversize grain and a smaller grain which is probably the optimum grain size for maximum creep strength; tests 12,100 and 11,250 psi at 925 F. Fig. 28 has undersize but uniform grains; tests 8200 and 8000 psi at 932 F. A possible explanation of these comparisons of creep strength gives greater weight to the effect of grain size; the weakening effect of mixed grain sizes, when all sizes are large as in Figs. 10 and 11, is less than the weakening effect of undersized but uniform grains as in Fig. 28 of this discussion. Considering the harmful effects of nonuniform grain size upon the creep strength of steels, the data given do not justify the conclusion that structure is more important than grain size in creep strength.

The characteristics of grain size in carbon steel have been extensively studied in technical literature, particularly in connection with the carburizing steels and with heat-treatment of steel for use at lower temperatures where the finer-grained, aluminum-killed steels are desirable. The making of alloy steels to a larger grain size for service at higher temperatures has not been much discussed. While open-hearth practice is usually limited by aluminum deoxidization to either a fine or coarse grain size similar to curve III, Fig. 30 of this discussion, the high-temperature user prefers a steel with the characteristics of curve IV so that, by a change in heat-treating temperature, any desired grain size could be obtained.

The mixed sizes of grains are related to the aluminum killing of the steel which introduces a peculiar inhibitor to grain growth. The grain-size restrainer cannot be judged by the amount of aluminum added to the molten steel, the total aluminum remaining in the steel, or the aluminum in solution, but is closely related to the aluminum oxide in the steel. The effectiveness of the latter is regulated principally by the degree of oxidization and temperature of the molten steel when the aluminum is introduced. The characteristics produced by a strong grain-growth inhibitor seem to apply to the eight steels analyzed in the authors' Table 1 in Figs. 1 to 8 inclusive, and to the writer's tested steel in Figs. 25 to 28, inclusive.

H. Montgomery.[16] Professor White and Mr. Crocker are to be congratulated on this fine piece of work which shows the ef-

[15] "Relation of Pretreatment of Steels to Austenitic Grain Growth," by J. E. Dorn and O. E. Harder, Trans. American Society of Metals, vol. 26, 1938, p. 106.

[16] The Lunkenheimer Co., Cincinnati, Ohio.

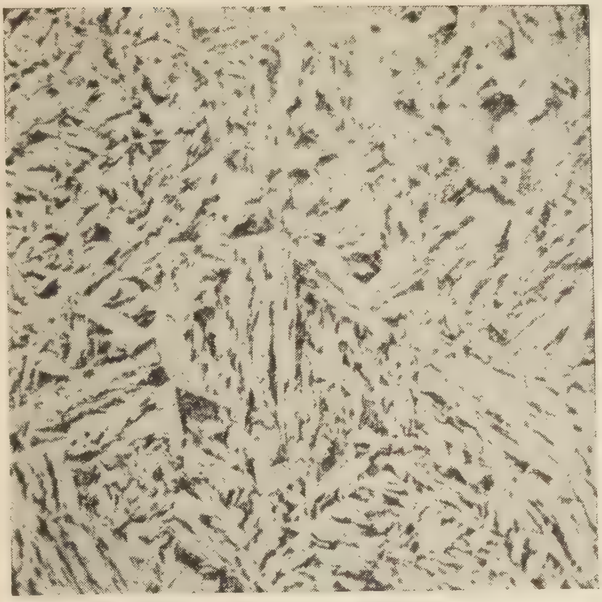

Fig. 31 (Magnification 500)

Fig. 32 (Magnification 500)

fects of some variables on the high-temperature creep properties of wrought carbon-molybdenum steel pipe.

From work done on cast steels of the WC-4 (A.S.T.M. A-217-39T) type, the writer has also come to the conclusion that the Widmanstätten type of structure imparts superior creep strength. Photomicrographs, Figs. 31 and 32, show two different types of structure as illustrative of this. Fig. 31, a specimen in which the Widmanstätten type of structure has been developed, shows a creep rate only one tenth as great as that of Fig. 32 with a structure similar to the sorbitic pearlite of the authors. Tests were made at 950 F and 15,000 psi.

From the apparent agreement of the writer's findings with the authors on the effect of structure the question is raised whether or not other steels of similar type may behave likewise.

Authors' Closure

The authors agree with the statement of Messrs. Emerson and Sinding-Larsen that "no definite conclusions have been reached with respect to the direct effect of actual aluminum additions on the creep characteristics of carbon-molybdenum steel, A.S.T.M. A206." The results given in the paper do tend to show, however, that when more than about $1^1/_2$ lb of aluminum are added per ton, the steel is sluggish in responding to grain-coarsening treatment and that very little if any of the Widmanstätten type of structure is present even in normalized material. The effect on creep strength of deoxidizing steel with aluminum or silicon-aluminum, as compared with silicon alone, is under investigation by the Joint Research Committee on Effect of Temperature on the Properties of Metals,[17] although no information seems to be available as yet on carbon-molybdenum pipe steel. It is to be hoped that the scope of future reports will be extended to furnish data on this important material.

Mr. Salmon touches upon the effect of duplex structure. Probably no steel can be said to be completely free from a duplex structure. It is a question of degree. The authors do not feel that most of the steels reported in their paper could be classified as duplex in the normally accepted interpretation of the term. Work of other investigators has shown that a pronounced duplex structure does appear to affect adversely the high-temperature properties of the steel.

The authors are in general agreement with Mr. Weaver with respect to the effect of oil-quenching, normalizing, and drawing on creep strength at the temperatures listed or implied. However, he apparently prefers an annealed structure to a Widmanstätten structure for 925 F service, basing his objections to the latter on an assumed higher rate of spheroidization, with resultant loss in creep strength. While Mr. Weaver concurs in recognizing the greater initial creep strength at 925 F for the normalized or Widmanstätten structure over the annealed or sorbitic-pearlite structure, he expresses concern over the possibility that the creep strength of normalized material would decrease sooner with time at temperature owing to faster carbide spheroidization. Reasoning from the latter premise he concludes that the annealed structure should be the more stable over extended service periods and that it should be preferred for that reason. This hypothesis seems to be based on an expressed belief that "the normalized carbon-molybdenum pipe steel becomes spheroidized in one third of the time required for the annealed steel to attain a similar condition."

In the authors' opinion the data available on spheroidization are too meager at the time of writing to justify any definite conclusion on behavior at 925 F. One of the authors did some work in his laboratory on a 0.13 per cent plain carbon steel which at 1200 F showed the rate of spheroidization in the normalized state to be about 3 times that for the same steel when in an annealed condition. This probably is the test which Mr. Weaver had in mind in his discussion. By extrapolation, the time for complete spheroidization at 925 F would be about 80,000 hr for the same plain carbon steel in an annealed condition. As stated, the authors question the wisdom of drawing conclusions from such meager data, especially when observations made at 1200 F have to be extended to 925 F, when conclusions are drawn from findings on a plain carbon steel rather than on a low-alloyed steel, with the recognition that a low-alloyed steel takes longer to spheroidize than a plain carbon steel, and, further, when conclusions are drawn from a test made on only one heat of steel.

[17] "Study of Effect of Variables on the Creep Resistance of Steels," by H. C. Cross and J. G. Lowther, Report of A.S.M.E.-A.S.T.M. Joint Research Committee on Effect of Temperature on Properties of Metals, Proceedings, American Society for Testing Materials, vol. 40, 1940, pp. 125–153.

CHART 4 STRUCTURES OF A CARBON-MOLYBDENUM PIPE STEEL (HEAT 5085) AFTER VARIOUS HEAT-TREATMENTS (REFER TO TABLE 4); LONGITUDINAL SECTIONS; MAGNIFICATION 100

Fig. 33 Specimen 1, "As Received"

Fig. 34 Specimen 2, Drawn at 1200 F

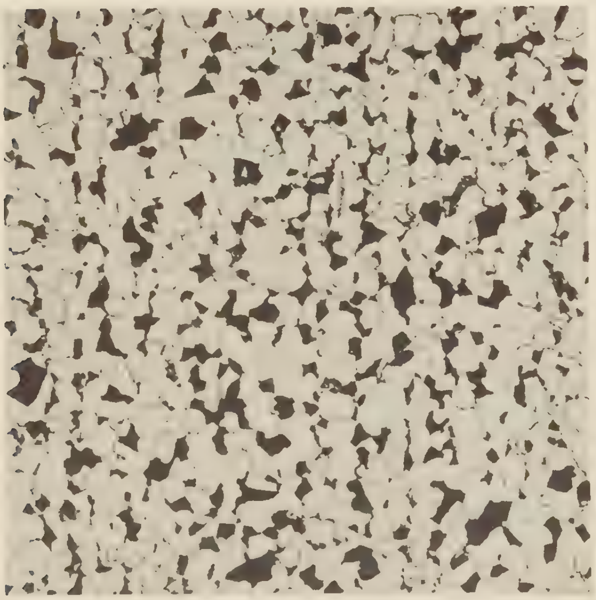

Fig. 35 Specimen 3, Annealed at 1650 F

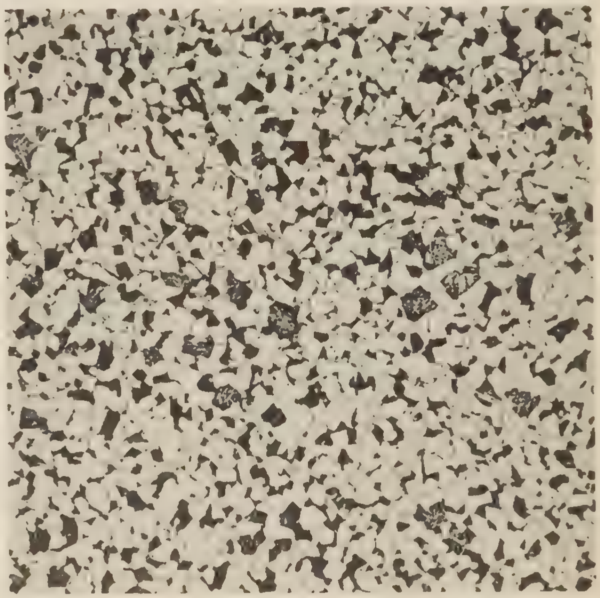

Fig. 36 Specimen 4, Normalized at 1600 F

In fact, Mr. Weaver observed in the preceding paragraph of his discussion that "for increasingly higher temperatures (above 800 F), the normalizing treatment leads, until near 1000 F the normalized and annealed structures become of equal creep strength for long-time, low-creep-rate tests." In this connection the authors wish to point out that their own conclusions are based on creep tests carried on at a loading of 15,000 psi for a full 1000 hr or more at temperature as prescribed by the Joint Research Committee on the Effect of Temperature on the Properties of Metals. Until conclusive data based on long-time tests are offered in proof of serious spheroidization of normalized carbon-molybdenum pipe steel at 925 F, the authors are disposed to dismiss this fear as not substantiated.

Furthermore, the most deleterious effect of spheroidization which the authors can foresee for normalized material would be a gradual decline in physical properties. The authors see no reason why any mere change of a small order in the rate of creep need be distressing in the case of a carbon-molybdenum pipe line. In fact the lesser over-all, or cumulative, creep of the normalized material would seem preferable to the greater, though more uniformly progressing, creep rate of the annealed material.

The high-temperature data given by Mr. Weaver and his dis-

CHART 4 (*Continued*) STRUCTURES OF A CARBON-MOLYBDENUM PIPE STEEL (HEAT 5085) AFTER VARIOUS HEAT-TREATMENTS (REFER TO TABLE 4); LONGITUDINAL SECTIONS; MAGNIFICATION 100

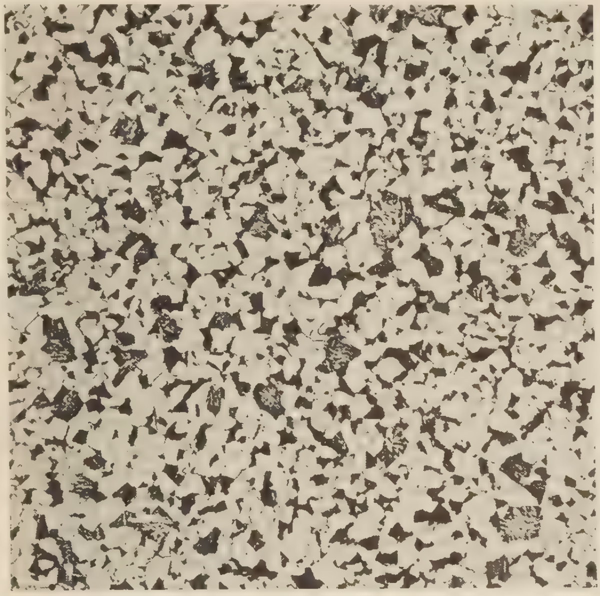

Fig. 37 Specimen 5, Normalized at 1650 F

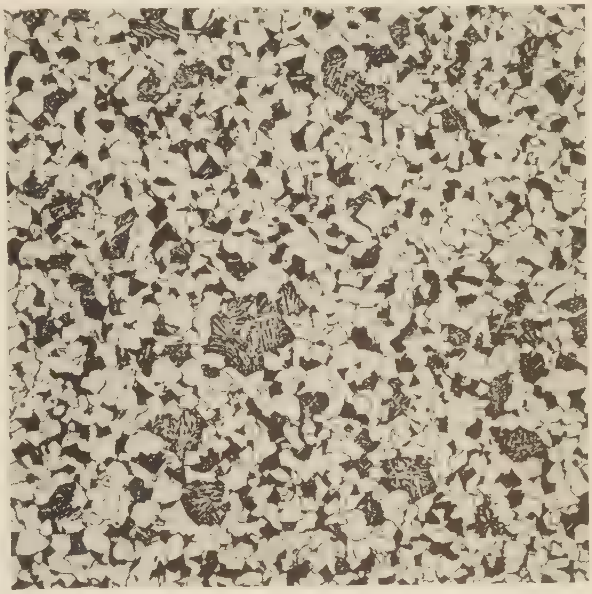

Fig. 38 Specimen 6, Normalized at 1700 F

Fig. 39 Specimen 7, Normalized at 1800 F

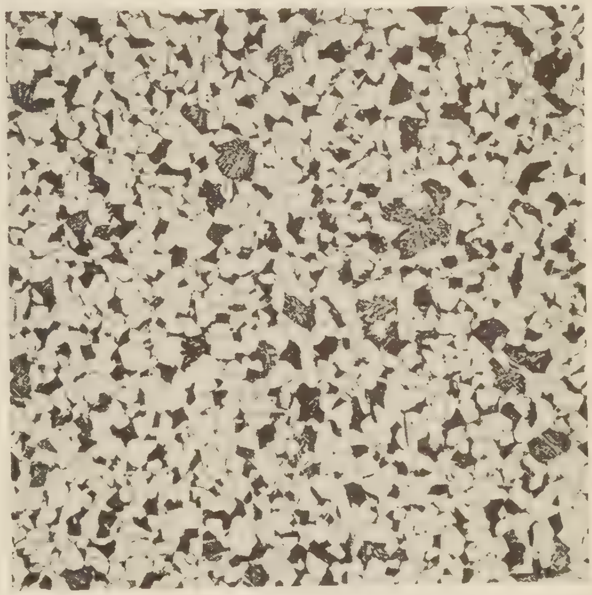

Fig. 40 Specimen 8, Normalized at 1650 F and Drawn at 1200 F

cussion of them are appreciated by the authors. Unfortunately, the type of steel from which these data were obtained is no longer considered acceptable for high-temperature purposes, that is, it would not show grain-coarsening characteristics if annealed or normalized from 1700 F. This is shown by the fineness of the grain in Mr. Weaver's photomicrograph, Fig. 28. If the steel were of the grain-coarsening type, the grain size would have been materially coarser than was given by the heat-treating cycle which ended with an anneal at 1650 F for 8 hr.

The authors fail to see how conclusions can be drawn from Mr. Weaver's data as to the relative effect of grain size and structure for, in the data given, no one variable appears to be held constant.

Attention is directed to the fact that treatment given the authors' sample, shown in Fig. 12 of the paper, was different from that given the sample shown in Mr. Weaver's Fig. 26. The authors' sample was quenched and normalized, whereas, Mr. Weaver's was quenched only.

The authors appreciate the material contributed by Mr. Weaver but regret that they cannot agree with all of his conclusions.

Mr. Montgomery's findings to the effect that the Widman-

stätten type of structure in cast steels of the WC-4 type improves high-temperature properties are interesting and timely.

The authors concede, of course, that there are other criteria besides tensile properties and creep strength on which the behavior of carbon-molybdenum pipe steel should be judged. The relative susceptibilities to stress-temperature embrittlement of annealed versus normalized carbon-molybdenum pipe steel have been investigated on a basis of time-to-rupture tests by E. L. Robinson.[18] While the results so far reported are somewhat limited, they would seem to indicate that, for temperatures of 900 to 1000 F, the normalized material continues to exhibit superior load-carrying ability over long periods despite a somewhat lesser elongation at the point of failure. It is to be hoped that this line of investigation will be continued by Mr. Robinson and others in an effort to obtain more conclusive data on this interesting phase of high-temperature behavior.

At the time the paper was presented, assurances were given that impact values for several conditions of heat-treatment would be furnished in the authors' closure. These tests were conducted with a view to determining:

(a) Whether the impact strength of "as-received" pipe was adequate without further heat-treatment.

(b) Whether normalizing or annealing gave superior impact strength.

(c) Whether a draw at 1200 F was beneficial to the impact strength of normalized pipe.

In order to make the results truly comparable, all impact-test coupons were machined from a single specimen of pipe. It is of interest to note that the impact strength for this particular specimen of pipe was somewhat below par in the "as-received" condition so that the effect of favorable heat-treatment was distinctly marked. For the sake of accuracy, three longitudinal and three transverse impact coupons were tested for each condition of heat-treatment. The results are given in Table 4.

The microstructures resulting from the various heat-treatments are shown in Figs. 33 to 40, inclusive, of this discussion. Photomicrographs were taken of both longitudinal and transverse specimens but without showing any significant difference, hence, only one set is reproduced here to serve for both.

The results of the impact tests serve to confirm the general conclusions stated by the authors in their original presentation of the paper. They still prefer to have the carbides exhibit a

[18] "High-Temperature Rupture and Creep Tests," by E. L. Robinson, Proceedings of the American Society for Testing Materials, vol. 40, 1940, pp. 811–817.

TABLE 4 RESULTS OF IMPACT TESTS, CARBON-MOLYBDENUM PIPE 268, HEAT 5085

No. of specimen	Condition of specimens	Impact strength, ft-lb[b] Longitudinal	Transverse
1	As received[a]	10	12
2	Drawn at 1200 F	14	13
3	Annealed at 1650 F	14	14
4	Normalized at 1600 F	87	62
5	Normalized at 1650 F	83	60
6	Normalized at 1700 F	68	52
7	Normalized at 1800 F	40	29
8	Normalized at 1650 F and drawn at 1200 F	79	63

[a] According to Section 4(a) of A 206–40 T.
[b] Average of three tests; standard A.S.T.M. Charpy V-notch specimens

considerable amount of Widmanstätten structure with a grain size ranging from 2 to 7 when normalized from a temperature around 1700 F. This view is supported by the decided improvement in impact strength afforded by normalizing, as compared with the inferior strength exhibited by the annealed specimens.

The results of the impact tests point to the possible desirability, moreover, of normalizing the "as-received" pipe rather than using any of it (viz., as straight lengths) in the "as-received" condition. On the other hand they raise a question of whether a draw at 1200 F following the normalize is sufficiently beneficial to be worth the extra expense. Owing to temperature variations throughout a commercial heat-treating furnace, it may be expedient to state a desired temperature range for the pipe material in the charge, say from 1650 to 1725 F as outside limits where the steel is made by the preferred melting practice.

In conclusion the authors wish to emphasize that heretofore not all molybdenum steel has been made by a melting practice conducive to producing large grains (say 3 to 6) when given a grain-coarsening treatment at 1700 F, or thereabouts. Steel which is not responsive to grain-coarsening treatment at this temperature does not tend to develop the desired Widmanstätten structure on normalizing. Sluggish steels of this sort are illustrated in the photomicrographs of Figs. 2, 3, 4, 5, 7, and 8 of the paper, whereas, somewhat more responsive steels are shown in Figs. 1 and 6, and the preferred type in Fig. 12 of the paper and Fig. 38 of this closure. None of the steels investigated, however, showed characteristics sufficiently poor to warrant outright rejection for the working stresses contemplated. Where the superior characteristics possessed by Heats 4129 and 4299 (Figs. 1 and 12, respectively) can be counted on, there would seem to be some justification for allowing correspondingly higher working stresses, provided further investigation does not show an excessive tendency to spheroidization or to stress-temperature embrittlement after prolonged service at 925 F.

Index to Mechanical Engineering

Volume 63, January—December, 1941

(A) denotes Abstract; (BR) Book Review; (C) Correspondence; (D) Discussion; (Ed) Editorial; (P) Photograph

Issue	Page Numbers
January	1– 90
February	91–172
March	173–256
April	257–334
May	335–420
June	421–502
July	503–570
August	571–632
September	633–692
October	693–768
November	769–854
December	855–944

B

C

D

E

G

H

N

O

P

Q

R

S

T

U

V

W

X

Y

Z

Index to A.S.M.E. Transactions

Volume 63, 1941

The A.S.M.E. Transactions for 1941 was issued monthly. Four of the twelve issues are the *Journal of Applied Mechanics*, the page numbers of which are preceded by the letter A.

The Society Records for the year 1941 appeared as supplements, one in February, 1941, the Memorial Biographies in October, 1941, and the index section in January, 1942. The page numbers for these supplements are designated by the symbol RI, and the February supplement contains its own index.

(BR) denotes book review; (D) discussion of a paper.

Issue	Page Numbers
January	1– 80
February	81–176
April	177–272
May	273–368
July	369–464
August	465–560
October	561–668
November	669–764
Journal of Applied Mechanics	
March	A- 48
June	A- 49–A- 96
September	A- 97–A-144
December	A-145–A-192

E

F

Q

R

S

T

U

V

W

Y

Z

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